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Effect of refrigerant thermophysical properties on flow reversal in microchannel evaporators

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ABSTRACT

The amount and frequency of reverse flow in microchannel evaporators largely depends on the thermophysical properties of the working fluid. In this study, four refrigerants (R134a, R1234yf, R245fa and R32) are examined in the same air conditioning system operated under flash gas bypass mode with venting of reversed vapor (to provide better control of the flow). The flow rate and frequency of reversed vapor are measured for each refrigerant. It has been found that fluids with lower heat of vaporization and higher specific volume difference between vapor and liquid phase) tend to generate more reversed vapor flow. Experimental results validate qualitatively and in most cases quantitatively the mechanistic model presented in Li and Hrnjak (2017a). The validated model is used to demonstrate the effects of thermophysical properties on flow reversal for nine widely used refrigerants.

1. Introduction

Microchannel heat exchangers are widely used in air conditioning and refrigeration systems because of their compactness and enhancement of heat transfer performance. However microchannel heat exchangers suffer from the problem of refrigerant maldistribution, especially for evaporators [13–15]. For condensers, refrigerant distribution has less effect on performance, on the other hand, better separation of flow might be desired instead of better mixing [16–19]. The non-uniform distribution of two phase refrigerant in evaporators can significantly deteriorate the evaporator performance as well as system efficiency. Boiling instabilities and flow reversal is one cause for refrigerant maldistribution. According to Tuo and Hrnjak [22]), flow reversal in microchannel evaporators deteriorates refrigerant distribution, increase pressure drop, although may increase local heat transfer coefficient. The focus of previous research on boiling instability in microchannels has been on heat sinks for small scale cooling applications, such as electronic cooling [1,4,25,26,20,8,9,5,7,10,13] and etc. The heat transfer and fluid flow conditions in heat sinks for electronic cooling are significantly different from those in air conditioning systems. In heat sinks, the incoming flow is normally subcooled and the exit quality is very low, while in microchannel evaporators of A/C systems, the fluid at the inlet is either two-phase or saturated liquid and the outlet is normally superheated. In addition, the heat flux in electronic cooling applications is much higher than that in air conditioning applications. Last but not least, water is the most commonly used fluid in the reviewed studies, and its thermophysical properties (vapor density, heat of vaporization, etc.) are significantly different with those of hydrofluorocarbons (R134a, R410A, etc.). To the best of the authors' knowledge, flow reversal in real air conditioning systems was first observed at CTS (Creative Thermal Solutions) in 2006 and later published by Bowers et al. [3]. The only studies of flow reversal and boiling instabilities in realistic air conditioning systems in open literature were carried out by Tuo and Hrnjak. They [22] invented a new system configuration to vent the reversed vapor in the evaporator. They found that the reverse vapor accounted for 2–8% of the total supplied liquid into the evaporator. By venting the reverse vapor, 5 % of capacity and 3 % of COP improvement could be achieved compared with the flash gas removal AC system baseline. The Flash Gas Bypass system was first used for systems with microchannel evaporators by Beaver, Hrnjak [2], later Elbel and Hrnjak [6]. Li and Hrnjak [16,18] developed a mechanistic model of bubble dynamics in a single microchannel. This model is able to predict the occurrence of flow reversal and was validated against the visualization results in Tuo and Hrnjak [24]. The same authors Li and Hrnjak [17,19] visualized the flow regimes including flow reversal in an air heating aluminum channel as part of an automotive evaporator. This

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Nomenclature

A	area (m^2)
D	hydraulic diameter (m)
F	force (N)
g	acceleration of gravity ($m{\cdot}s^{-2}$)
h	enthalpy ($J{\cdot}kg^{-1}$)
L	length (m)
m	mass (kg)
$\dot{m}$	mass flow rate ($kg{\cdot}s^{-1}$)
P	pressure (kPa)
$\dot{q}$	heat flux ($kW{\cdot}m^{-2}$)
R	ratio
V	velocity ($m{\cdot}s^{-1}$)

Greek

ρ	density ($kg{\cdot}m^{-3}$)

Subscripts

d	downstream
$evap$	evaporated
f and l	liquid
$fric$	friction
g and v	vapor
i	index
in	inlet
out	outlet
u	upstream

visualization is believed to be very representative of the flow regimes in realistic automotive evaporators.

This paper first presents the development of a numerical model of bubble dynamics in single microchannel based on Li and Hrnjak [16,18]. Using this model, bubble dynamics simulations are carried out for nine widely used refrigerants in the same microchannel and under the same operation conditions. Simulation results demonstrate that the amount and frequency of reverse flow largely depend on the specific volume difference and heat of vaporization of the working fluid. Four refrigerants with different thermophysical properties are selected and experimentally examined under the conditions that are very close to the simulated. In reality, each refrigerant should have its own optimized channel geometry thus different heat flux, but the purpose of this study is to single out the effect of refrigerant thermophysical properties on flow reversal, so the heat flux and channel geometry are kept the same for all refrigerants. Experimental results are used to validate the numerical model.

2. Model description

This model is based on the study of Li and Hrnjak [16,18]. In Li and Hrnjak [16,18], the flow regime inside of one microchannel is treated as alternating liquid and vapor slugs (annular flow is treated as elongated vapor slug). Their model treats each vapor and liquid slug as finite volume as shown in Fig. 1 and applies mass, momentum and energy conservation for each of them. For each liquid slug, the momentum, mass and energy equations are formulated as Eqs. (1)–(3). $L_{l,i}$ is the length of the corresponding liquid slug, and $L_{v,i-1}$ and $L_{v,i}$ are the lengths of the upstream and downstream vapor slugs. Since all heat input is used to evaporate liquid refrigerant, mass conservation equation and energy conservation equation can be coupled together.

$$P_{i-1,d}A - P_{i,u}A + F_{fric,l,i} + m_{l,i}g = m_{l,i}\frac{dV_i}{dt} \tag{1}$$

$$\frac{dm_{l,i}}{dt} = \dot{m}_{evap,i,u} + \dot{m}_{evap,i-1,d} \tag{2}$$

$$\frac{dm_{l,i}}{dt}h_{lv} = (\dot{m}_{evap,i,u} + \dot{m}_{evap,i-1,d})h_{lv} = \dot{q}\pi D\left(L_{l,i} + \frac{L_{v,i-1}}{2} + \frac{L_{v,i}}{2}\right) \tag{3}$$

Following the same logic, the momentum, mass and energy equation for each vapor slug is formulated as Eqs. (4)–(6). It is assumed that half of the vapor slug has the velocity of the upstream liquid slug and the other half has the velocity of the

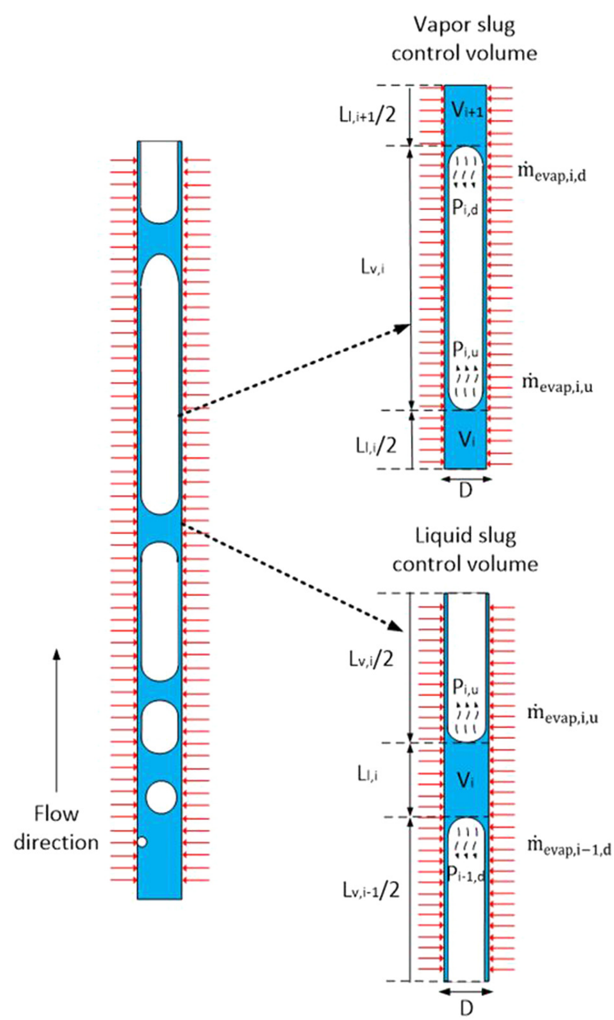

Fig. 1. Control volumes for liquid and vapor slugs.

downstream liquid slug. $L_{v,i}$ is the length of the corresponding vapor slug, and $L_{l,i}$ and $L_{l,i+1}$ are the lengths of the upstream and downstream liquid slugs.

$$P_{i,u}A - P_{i,d}A + F_{fric,v,i} = m_{v,i}\frac{d(V_i/2 + V_{i+1}/2)}{dt} \tag{4}$$