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Periodic bubble emission and appearance of an ordered bubble sequence (train) during condensation in a single microchannel

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Abstract

Condensation of steam in a single microchannel, silicon test section was investigated visually at low flow rates. The microchannel was rectangular in cross-section with a depth of 30 μm, a width of 800 μm and a length of 5.0 mm, covered with a Pyrex glass to allow for visualization of the bubble formation process. By varying the cooling rate during condensation of the saturated water vapor, it was possible to control the shape, size and frequency of the bubbles formed. At low cooling rates using only natural air convection from the ambient environment, the flow pattern in the microchannel consisted of a nearly stable elongated bubble attached upstream (near the inlet) that pinched off into a train of elliptical bubbles downstream of the elongated bubble. It was observed that these elliptical bubbles were emitted periodically from the tip of the elongated bubble at a high frequency, with smaller size than the channel width. The shape of the emitted bubbles underwent modifications shortly after their generation until finally becoming a stable vertical ellipse, maintaining its shape and size as it flowed downstream at a constant speed. These periodically emitted elliptical bubbles thus formed an ordered bubble sequence (train). At higher cooling rates using chilled water in a copper heat sink attached to the test section, the bubble formation frequency increased significantly while the bubble size decreased, all the while forming a perfect bubble train flowing downstream of the microchannel. The emitted bubbles in this case immediately formed into a circular shape without any further modification after their separation from the elongated bubble upstream. The present study suggests that a method for controlling the size and generation frequency of microbubbles could be so developed, which may be of interest for microfluidic applications. The breakup of the elongated bubble is caused by the large Weber number at the tip of the elongated bubble induced by the maximum vapor velocity at the centerline of the microchannel inside the elongated bubble and the smaller surface tension force of water at the tip of the elongated bubble.

Keywords: Condensation; Microchannel; Microfluidics; Bubble train

1. Introduction

Condensation in microchannels could have many applications in miniature refrigeration systems. Nearly all theoretical and experimental work available in the public domain focus on channel sizes greater than 1 mm. Garimella et al. [1,2] reported the development of an experimentally validated model for pressure drop during intermittent flow for refrigerant R134a condensing in horizontal minichannels with hydraulic diameters ranging from 0.5 to 4.91 mm. A simple correlation for a non-dimensional unit-cell length based upon slug Reynolds numbers was used to compute the total pressure drop. Baird et al. [3] developed a new apparatus capable of controlling local heat fluxes through single small passages in thermoelectric coolers. They reported local heat transfer coefficients for condensation of R123 and R11 for a wide range of mass

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Nomenclature

$\frac{dp}{dx}$	the axial pressure gradient of the vapor inside the elongated bubble (Pa/m)
D	the hydraulic diameter of the microchannel (m or μm)
D_x	horizontal axis length of elliptical bubble (μm)
D_y	vertical axis length of elliptical bubble (μm)
f	bubble emission frequency
G	mass velocity (kg/m^2 s or g/cm^2 s)
H_{in}	total enthalpy at the microchannel entrance (W)
H_{out}	total enthalpy at the microchannel exit (W)
h_f	saturated liquid enthalpy (J/kg)
h_v	saturated vapor enthalpy (J/kg)
Ja	Jakob number
L	the central microchannel length (m)
M	total mass flow rate (kg/s)
m_v	vapor mass flow rate near the exit of the central microchannel (kg/s)
Nu	Nusselt number
Pe	Peclet number
p_{in}	inlet pressure of the saturated water vapor (Pa or kPa)
R^*	reduced equivalent bubble radius
R_o	initial equivalent bubble radius after release from the elongated bubble (μm)
$R_{o,cir}$	optically measured initial biscuit-shaped bubble radius (μm)
R_{out}	equivalent biscuit-shaped bubble radius near the exit of the central microchannel (μm)
t	time (s or ms)
t_r	bubble residence time in the microchannel after its generation (s or ms)
T_{in}, T_{out}	inlet and outlet fluid temperature of the micro-condenser (°C)
T_{w1}, T_{w2}	temperatures measured by thermocouples in the two holes of the copper heat sink (°C)
u_f	the axial velocity of the condensed water (m/s)
u_v	the axial velocity of the vapor inside the elongated bubble (m/s)
W	the width of the microchannel (m or μm)
We	Weber number
x	horizontal coordinate in the microchannel along the flow direction (μm)
$\bar{x}$	vapor mass quality near the exit of the central microchannel
y	vertical coordinate in the microchannel perpendicular to the flow direction (μm)
δ	microchannel depth (μm)
μ_v	the vapor viscosity (Pa s)
ρ_v	the vapor density (kg/m^3)
σ_f	the surface tension of condensed water (N/m)

velocities (70–600 kg/m^2 s) and heat fluxes (15–110 kW/m^2) in circular channels with internal diameters of 0.92 and 1.95 mm. They observed an enhancement in the condensation heat transfer coefficient for which the tube size had little or no influence. In another study of note, a theoretical model was proposed to estimate the film condensation heat transfer coefficient in square cross-section horizontal minichannels by Wang et al. [4]. Their model took into account the effects of surface tension, vapor shear stress and gravity. Their simulations were performed with R134a as the working fluid in a channel of 1.0 mm hydraulic diameter.

Médéric et al. [5] performed a flow visualization study on capillary condensation of *n*-pentane in a 0.56 mm glass channel at flow rates of 0–18 kg/m^2 s. They observed the oscillation of an elongated bubble formed at the entrance of their channel and also periodic break off of bubbles that flowed downstream. They deduced void fractions from image processing of their videos of the condensation process inside their capillary glass tube.

Cavallini et al. [6] studied the pressure drop characteristics of a 1.4 mm hydraulic diameter, multiport, minichannel tube during adiabatic two-phase flow of HFC refrigerants. The tube consisted of eleven parallel rectangular cross-section channels. Bandhauer et al. [7] developed a model to predict condensation heat transfer in circular microchannels and compared their predictions with their measured values. Recently, theoretical/numerical work on condensation heat transfer in microchannels has been presented by Du and Zhao [8] and Wang and Rose [9,10].

Up to now, very little information has been available on condensation in microchannels with a hydraulic diameter smaller than 100 μm. A recent study of condensation in microchannels was reported by Wu and Cheng [11] for channels having a hydraulic diameter of 82.8 μm where various flow patterns, such as fully droplet flow, droplet/annular/injection/slug-bubbly flow, annular/injection/slug-droplet flow, and fully slug-bubbly flow, were observed. They found that pressure and temperature oscillations correlated with these flow patterns.

In summary, most studies on condensation in microchannels have focused on: (a) channels with diameters about 1.0 mm in size or larger; (b) their two-phase pressure drops; (c) their condensation heat transfer coefficients and (d) the types of flow patterns formed.

With respect to the present study, there is ever increasing interest in controlling bubble and droplet size and frequency for microfluidic applications. Typical microfluidic channel sizes (height and width) are in the range of 10–100 μm, which are one to two orders of magnitude smaller than those often used for heat transfer engineering, e.g. in miniature refrigeration systems. The flow rates are in the range of 10–1000 nl/s. These parameters lead to nanoliter