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Condensation flow patterns and heat transfer in horizontal microchannels

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ABSTRACT

An experimental investigation was carried out to study the effect of refrigerant mass flux, local vapour quality, coolant flow rate and inlet coolant temperature on the local condensation heat transfer coefficient. Flow visualization was also conducted to capture flow patterns during flow condensation using a high-speed camera integrated with a microscope. HFE-7100, a dielectric and eco-friendly refrigerant was used in rectangular multi-microchannels with a hydraulic diameter of 0.57 mm. Experiments were performed at a saturation temperature of 60 °C, mass flux range 48–126 kg/(m^2 s), coolant flow rate range 0.5–1.1 L/min and inlet coolant temperature range 20–40 °C. The results showed that the local condensation heat transfer coefficient increases with increasing mass flux and decreases with decreasing local vapour quality. A negligible effect of the coolant side conditions, saturation-to-wall temperature difference, on the local condensation heat transfer coefficient was found. The main flow regime was annular flow, while slug and bubbly flow were found at some operating conditions. The experimental results were compared with the existing correlations for heat transfer rates. Also, two existing flow pattern maps, for conventional and mini/microchannels, were used to compare the current flow pattern results.

1. Introduction

The dissipation of high heat fluxes from small areas with uniform surface temperature is one of the greatest challenges in the thermal design of semiconductor electronic devices. Two phase flow boiling in microchannels is one of the promising cooling techniques that can dissipate high heat fluxes at nearly uniform surface temperature (slightly higher than the saturation temperature). This is due to a number of advantages: high heat transfer rate, small size, light weight and small fluid charge (small amount enters the atmosphere if leakage occurs). Accordingly, a large number of researchers such as [1–3] focused on studying flow boiling characteristics in microchannels. This was motivated by the lack of understanding several fundamental issues in flow boiling such as the dominant heat transfer mechanism(s), flow instability, boiling incipience, mechanisms of dryout and the prediction of heat transfer rates and pressure drop. With the assumption that these fundamental issues are resolved and the possibility of designing and fabricating multi-microchannel evaporators, the design of a small scale pumped loop cooling system is then the next challenge. This difficulty arises from the basic fundamental knowledge in designing appropriate small/micro condensers, whose size might be significant depending on the final heat sink (ambient air or water for example) compared to the size of a small scale pumped loop cooling system. There are very limited numbers of studies in the literature focused on studying condensation in multi-microchannel configurations. Therefore, the objective of the present study is to investigate experimentally the thermal performance of a multi-microchannel condenser. It is commonly agreed that flow boiling characteristics in microchannels are different compared to conventional large diameter channels. Similarly, condensation in microchannels may also differ compared to large diameter channels. For example, the dominant forces that might affect the condensation mechanism in microchannels may be different compared to that in larger channels. Del Col et al. [4] reported that the condensation mechanism depends on the relative importance of the surface tension, gravity and shear forces, which, in turn, depend on several parameters, such as vapour quality, mass flux, fluid properties and channel geometry. In larger tubes, the gravity and shear forces are more dominant, while in mini/microchannels, the surface tension force can also become important. Del Col et al. [5] mentioned that in non-circular mini/microchannels, the surface tension force can pull the liquid phase towards the channel corners leading to a thin liquid film and low thermal resistance at the flat sides. This means that the heat transfer coefficient in non-circular channels is expected to be higher compared to circular channels. Kim and Mudawar [6] reported that, when the mass flux increases,

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Nomenclature

A	area [m^2]
b	length of channel side [m]
Bo	bond number [–] $Bo = (\rho_l - \rho_g)*g*D_h^2*\sigma^{-1}$
Bo_{cr}	critical Bond number [–] $Bo_{cr} = 1/[\rho_l*(\rho_l - \rho_g)^{-1} - \pi/4]$
C_p	specific heat at constant pressure [J/kg K]
D_h	hydraulic diameter [m]
f	fanning friction factor [–]
G	mass flux [kg/m^2 s]
g	gravitational acceleration [m/s^2]
H	height [m]
h, HTC	heat transfer coefficient [W/m^2 K]
h_{LT}	heat transfer coefficient with total mass flowing as liquid [W/m^2 K]
h_{NU}	heat transfer coefficient, the Nusselt equation [W/m^2 K]
$\bar{h}$	average heat transfer coefficient [W/m^2 K]
i_{lg}	latent heat of vaporization [J/kg]
J_G	dimensionless vapour velocity [–] $J_G = G*x/(g*D_h*\rho_g*\Delta\rho)^{0.5}$
k	thermal conductivity [W/m K]
K_{90}	loss coefficient for the 90° turns [–]
L	length [m]
m	fin parameter, $\sqrt{2h/k_{cu}*W_{fin}}$
$\dot{m}$	mass flow rate [kg/s]
MAE	mean absolute error
N	number of channels [–]
$\overline{Nu}$	average Nusselt number [–] $\overline{Nu} = \bar{h}*D_h*k_f^{-1}$
P	pressure [Pa]
Pr	Prandtl number [–] $Pr_l = cp_l*\mu_l*k_f^{-1}$
P_R	reduced pressure [–] $P_R = P_l/P_{cr}$
Q	heat rate [W]
q''	heat flux [W/m^2]
$\overline{q''}$	average heat flux [W/m^2]
Re	Reynolds number [–] $Re = G*D_h*\mu_l^{-1}$
Re_{ls}	superficial liquid Reynolds number [–] $Re_{ls} = G(1-x)*D_h*\mu_l^{-1}$
Re_{gs}	superficial vapour Reynolds number [–] $Re_{gs} = G*x*D_h*\mu_g^{-1}$
Su_{go}	vapour only Suratman number [–] $Su_{go} = \rho_g*\sigma*D_h*\mu_g^{-2}$
T	temperature [K]
v	specific volume [m^3/kg]
W	width [m]
W_c	centre to centre distance of microchannel widths [m]
We	Weber number [–] $We = G^2*D_h*(\rho*\sigma)^{-1}$
We^*	modified Weber number [–] $We^* = 2.45(Re_{gs}^{0.64}/Su_{go}^{0.3}(1 + 1.09X_{tt}^{0.039})^{0.4})$
x	vapour quality [–]
X	Lockhart–Martinelli parameter [–] $X = [(dP/dz)_l/(dP/dz)_g]^{0.5}$
$y1$	vertical distance between thermocouple and channel bottom [m]
$y2$	vertical distance between thermocouples [m]
Z	Shah's correlating parameter [–] $Z = (1/x-1)^{0.8}*P_R^{0.4}$
z	axial distance [m]

Greek symbols

α	area ratio [–] $\alpha = A_{sec,min}/A_{sec,max}$
β	aspect ratio [–] $\beta = W_{ch}/H_{ch}$
ΔP	pressure drop [Pa]
η	fin efficiency [–]
Θ	percentage predicted within ± 30% of data
μ	viscosity [Pa s]
ρ	density [kg/m^3]
σ	surface tension [N/m]
Ø	two-phase pressure drop multiplier [–]

Subscripts

3	based on three-sided heat transfer in rectangular channel
4	based on four-sided heat transfer in rectangular channel
b	base
B	bottom
ch	channel
cr	critical
cu	copper
eq	equivalent
exp	experiment
f	fluid
fin	channel fin
fi	fluid in
fo	fluid out
g	gas or vapour
ht	heat transfer
i	inlet
ip	inlet plenum
l	liquid
meas	measured
o	outlet
op	outlet plenum
p	plenum
pred	predicted
sat	saturation
sc	sudden contraction
se	sudden expansion
sec	section
sp	single phase
T	top
tp	two phase
tt	turbulent liquid-turbulent vapour
wi	internal wall surface
wti	water in
wto	water out
z	axial local

interfacial shear stress increases. This leads to a thinner liquid film and higher heat transfer coefficient. As mentioned above, in larger tubes, gravity force could also affect the heat transfer coefficient by draining the liquid condensed to the channel bottom leading to a thicker liquid film. Moreover, Zhang and Li [7] studied numerically the effect of gravity force (g = 0 m/s^2, g = 9.81 m/s^2 and g = 19.6 m/s^2) on flow condensation in horizontal single tubes with diameter 0.25–4 mm. They reported that when the gravity force increases, the liquid film becomes thinner at the upper section of the tube leading to low thermal resistance and thus high transfer coefficient. Their simulation results indicated that nearly two-thirds of the tube perimeter was lubricated with a thin liquid film (small thermal resistance and thus high heat transfer coefficient). In other words, the contribution of the upper section of the tube to the heat transfer rate is more pronounced compared to the lower section of the tube. In the following sub-sections, the flow patterns, heat transfer coefficient, pressure drop and heat transfer correlations are reviewed and discussed.