



Heat transfer coefficient calculated using a linear pressure gradient assumption and measurement for flow boiling in microchannels



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ABSTRACT

To obtain local flow boiling heat transfer coefficients, some studies used a linear pressure gradient method. In this study, this method is compared with a pressure gradient measurement method. The measurement was carried out for flow boiling of de-ionized water in single horizontal microchannels. Three single microchannels with widths of 0.5 mm, 1 mm and 1.71 mm and a depth of 0.39 mm, giving hydraulic diameters of 0.438 mm, 0.561 mm and 0.635 mm, were tested. The channel length was the same, 62 mm. The nominal mass flux used was $800 \text{ kg/m}^2 \text{ s}$ and heat fluxes ranging from 222 to 685 kW/m^2 were applied. An inlet fluid temperature of $98 \text{ }^\circ\text{C}$ and an inlet pressure of 125 kPa (abs) corresponding to an inlet subcooling of 7 K, were maintained at the channel entrance. The trends of heat transfer coefficient decrease with quality at a fixed heat flux and mass flux along the channel. However, the heat transfer mechanism was dominated by nucleate boiling for the 0.561 mm and 0.635 mm channels indicated by the increased local heat transfer coefficient with heat flux, while for the 0.438 mm channel, it was dominated by convective boiling because there was no effect of heat flux on the local heat transfer coefficient. Local heat transfer coefficients determined using the two methods were found to be significantly different, especially at high heat fluxes. When the Φ is less than $9 \text{ W/m}^2 \text{ Pa}$ for the 0.438 mm channel, less than $11 \text{ W/m}^2 \text{ Pa}$ for the 0.561 mm channel and less than $16 \text{ W/m}^2 \text{ Pa}$ for the 0.635 mm channel, the linear pressure distribution may be used with caution.

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1. Introduction

Flow boiling phenomena in microchannels are still challenging the research community due to their complicated nature and the difficulties involving parameter measurements such as dimensions, surface roughness, pressure and temperatures. Therefore, contradictory experimental results have been noticed in the microchannel studies and have led to different interpretations of heat transfer trends and mechanisms. Some researchers, e.g. [1–3], concluded that nucleate boiling was the dominant mechanism, whereas others, e.g. [4–6], reported that convective boiling dominated the heat transfer mechanism in microchannels.

In the literature, many heat transfer trends and mechanisms are revealed. When the trend of heat transfer coefficients increase with heat flux at a constant mass flux and there is no effect of vapor quality detected, the heat transfer mechanism is classified as nucleate boiling, e.g. [1–3,7]. On the other hand, when the heat transfer coefficient increase with mass flux and vapor quality, but does not depend on the heat flux, then the heat transfer mechanism is categorized as convective boiling, see a review paper

[7]. The effect of mass flux on heat transfer coefficient is usually found to increase the heat transfer coefficient, while the effect of quality on heat transfer coefficient may increase or decrease the heat transfer coefficient. Some results indicate a decreased heat transfer coefficient with quality, e.g. Qu and Mudawar [5], Wang and Sefiane [8]. Although the effect of quality on the heat transfer coefficient was found to decrease the heat transfer coefficient, they concluded that the heat transfer mechanism was convective boiling. This was due to no effect of heat flux on the heat transfer coefficient. In contrast, Sobierska et al. [9,10] and Mirmanto et al. [11,12] found that the heat transfer coefficient decreased with mass flux and quality but increased with heat flux and they elucidated that the dominant heat transfer mechanisms were nucleate boiling. Regardless nucleate or convective boiling, when the heat transfer coefficient starts to decrease with quality, authors conventionally explain that the decreased heat transfer coefficient is due to dryout, e.g. Kew and Cornwell [13], Wang et al. [14], Mahmoud et al. [15] and Wang and Safiane [8], or because of the sharply decreased pressure drop, see [10–13] and Lin et al. [16]. On the other hand, when the heat transfer coefficient increases with quality, authors elucidate that the increased heat transfer coefficient is due to the liquid film evaporation, e.g. Sumith et al. [17],

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Nomenclature

A_{ht}	heat transfer area, m ²	L^*	dimensionless length
c_p	specific heat, J/kg K	\dot{m}	mass flow rate, kg/s
C	constant for Eq. (4)	P	power $P = VI$
D_h	hydraulic diameter, m	p	pressure, Pa
f_{app}	fanning friction factor, dimensionless	q''	heat flux, W/m ²
f_{FD}	fanning friction factor for fully developed flow	q_{Loss}	thermal heat loss, W
G	mass flux, kg/m ² s	Ra	average surface roughness, μm
h	heat transfer coefficient, W/m ² K	Re	Reynolds number = GD_h/μ , dimensionless
I	current, A	T	temperature, K
i	enthalpy, J/kg	V	voltage, V
$K(\infty)$	incremental pressure drop	x	thermodynamic vapor quality, dimensionless
k	thermal conductivity, W/m K	z	axial position, m
L	length, m	sp	single-phase
		tp	two-phase
		tc	thermocouple
		w	wall
		Φ	Phi
Subscripts		Greek symbols	
cu	copper	α	heat transfer coefficient, W/m ² K
g	gas/vapour	Δ	Chang
i	inlet	μ	dynamic viscosity, Pa s
ip	inlet plenum	ρ	density, kg/m ³
l	liquid		
lg	liquid gas/vapour		
o	outlet		
op	outlet plenum		
sat	saturation		
sub	sub-cooled		

Lee and Lee [18]. Table 1 shows findings regarding heat transfer coefficient trends obtained by previous studies.

The trend of local heat transfer coefficients basically depends on the local saturation pressure (or saturation temperature) inside the channel and the wall temperature. Meanwhile, there is no accurate correlation to predict the local saturation pressures (temperatures) along the channel. Therefore, some studies use an assumption that the saturation pressure distribution decreases linearly from the point where the saturated flow boiling starts to occur to the outlet of the channel, e.g. Qu and Mudawar [5], Mirmanto et al. [11,12], Mahmoud et al. [15]. The local saturation temperature distribution mainly depends on the predicted local saturation pressures. The values of the local saturation temperature is greatly influenced by the assumption of the local pressure distribution. In contrast, Harrichian and Garimella [23] assumed that the saturation temperature decreased linearly along the channel. These two different assumptions may give dissimilar trends/values of heat transfer coefficient.

Flow boiling pressure drops have been studied extensively and intensively, even, many correlations for flow boiling pressure drop have been developed, see Lee and Garimella [6], Qu and Mudawar [21], Lockhart–Martinelli [24], Collier and Thome [25], Lee and Lee [26], Zhang et al. [27]. Applying the correlations to assert the local pressure distribution along the channel is not simple and even becoming more complicated when all components of the pressure drop such as accelerational and frictional pressure drops are included. However, as the correlation is a function of vapor quality powered 2 or more, it can be inferred that the pressure distribution along the channel is not linear. In general, experimental flow boiling pressure drops reported in the literatures were obtained from the inlet and outlet pressure measurements, while the local pressure distribution usually was not reported.

In this study, the trends of heat transfer coefficient obtained using the two methods (linear pressure gradient assumption and measurement methods) are evaluated and the heat transfer

mechanisms are examined using dominant parameters, i.e. heat flux and vapor quality.

2. Experimental apparatus

A schematic diagram of the flow loop is shown in Fig. 1. The working fluid, de-ionized water, was degassed prior to performing flow boiling experiments at a temperature of 105 °C controlled using a PID West 6100. Water was circulated using a gear pump (model GA-T22, PFSB) adjusted using a programmable variable speed drive (model Ismatec Reglo ZS-Digital). Two filters, a 1 mm mesh and 1 μm , were installed in the test rig to remove any particles in the fluid. To increase the fluid temperature to the desired value at the entrance of the test section, electric pre-heaters with PID controllers were fitted in the upstream of the test section. A water-glycol solution supplied from an external chiller unit was used to cool the water in the subcooler and vapor in the reflux and cooler-condensers.

Fig. 2 shows an exploded view of the main parts of the test section assembly. The overall dimension of the microchannel test section, which was made of oxygen-free copper block, was 12 mm wide \times 25 mm high \times 72 mm long. A single rectangular microchannel with a width of 0.5 mm, depth of 0.38 mm and length of 62 mm was cut in the top surface of the copper block using a Kern HSPC 2215 high-speed micro-milling machine. Other two test sections were also fabricated with the same ways, depth and length but different widths (1 mm and 1.71 mm). The microchannel width was measured using a scanned electron microscope (Zeiss Supra 34 PV) with an accuracy of $\pm 1 \mu\text{m}$, and the channel depth was measured using a microscope (TSER V-190) with an accuracy of $\pm 1 \mu\text{m}$. The channel length was measured using a digital vernier calliper with a resolution of 10 μm . The average surface roughness on the channel base was measured using a Zygo New View 5000 surface profiler with an accuracy of $\pm 1 \text{ nm}$ and found to be around 1.11 μm for all test sections. The measured surface roughness in

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