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Consolidated methodology to predicting flow boiling critical heat flux for inclined channels in Earth gravity and for microgravity

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ABSTRACT

The transition from single-phase to two-phase thermal systems in future space vehicles demands a thorough understanding of flow boiling critical heat flux (CHF) in reduced gravity, including microgravity. This study is a comprehensive, consolidated investigation of the complex trends of flow boiling CHF in a rectangular channel in both microgravity and for different orientations in Earth gravity. It is shown that the *Interfacial Lift-off Model* provides good predictions of CHF data for both gravitational environments and both single-sided and double-sided heating. CHF mechanism in Earth gravity is shown to be highly sensitive to flow orientation at very low velocities, but is consistent with the wavy vapor layer depiction of the *Interfacial Lift-off Model* at high velocities. The model predicts a stable vapor-liquid interface for downflow with a downward-facing heated wall at lower velocities, and wavy interface with a critical wavelength that decreases with increasing velocity at higher velocities. Predicted CHF values for microgravity fall about midway between the maxima and minima for Earth gravity. Overall, predicted values of CHF and key interfacial parameters for all orientations in Earth gravity and for microgravity converge above ~1.5 m/s, which points to a velocity threshold above which inertia begins to effectively negate gravity effects.

1. Introduction

1.1. Two-phase thermal management

Single-phase thermal management systems have been widely used in many industrial applications. But increasing heat densities in many modern technologies are making single-phase thermal management increasingly difficult to implement, and have shifted interest to two-phase thermal management [1]. Technologies demanding intense heat removal include high performance computers, hybrid vehicle power electronics, avionics, and laser and microwave directed energy weapon systems. All these applications share a common trend of increasing rate of heat removal from small surface areas. The effectiveness of two-phase thermal management schemes for these applications stems from their ability to capitalize upon latent heat of the coolant rather than sensible heat alone, providing orders of magnitude enhancement in heat transfer coefficient compared to single-phase schemes.

Another important attribute of two-phase thermal management is flexibility in selecting a flow configuration that is compatible with the geometrical and packaging needs of the heat dissipating device or system. This includes pool boiling thermosyphons, channel flow boiling, jet-impingement and spray [1], with emphasis placed on very high flux cooling schemes [2–4]. Channel flow boiling consists of mounting heat dissipating devices in a linear fashion along the walls of a flow channel. This configuration is both very versatile and compatible with packaging practices in many applications. More recently, researchers determined that the cooling performance in channel flow boiling can be greatly ameliorated by reducing the hydraulic diameter of the flow channel, i.e., by using mini/micro-channel flow boiling [1,5,6].

1.2. Critical heat flux (CHF) limit

The afore-mentioned ability of two-phase cooling schemes is realized within the nucleate boiling regime, which capitalizes on high frequency formation, growth, and departure of vapor bubbles from the heat-dissipating wall, while also requiring continued replenishment of the surface with bulk liquid to compensate for the liquid that is consumed at the wall by evaporation. Critical heat flux (CHF) is arguably the most important limit for two-phase cooling schemes, and is closely associated with cessation of bulk liquid access to the surface. With the nucleate boiling process

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Nomenclature

A	channel flow area
A_k	flow area for phase k
b	ratio of wetting front length to wavelength
$C_{f,i}$	interfacial friction factor
CHF	critical heat flux, q''_m
c_p	specific heat at constant pressure
c_r	real component of wave speed
D	hydraulic diameter of flow channel
D_k	hydraulic diameter for phase k
G	mass velocity
g	gravity
g_e	Earth's gravity
g_n	component of gravity normal to heater wall
H	height of flow channel's cross-section
H_a	heated wall a
H_b	heated wall b
h_{fg}	latent heat of vaporization
k_c	critical wave number
L_d	development length of flow channel
L_e	exit length of flow channel
L_h	heated length of flow channel
$\dot{m}$	mass flow rate
MAE	mean absolute error
p	pressure
P_i	interfacial perimeter
p_{in}	pressure at inlet to heated portion of channel
P_w	wall friction perimeter
q''_m	critical heat flux (CHF)
q''_w	wall heat flux
Re	Reynolds number
T	temperature
T_{in}	temperature at inlet to heated portion of channel
T_{out}	temperature at outlet from heated portion of channel
T_{sat}	saturation temperature
$\Delta T_{sub,in}$	inlet subcooling, $T_{sat} - T_{in}$
$\Delta T_{sub,out}$	outlet subcooling, $T_{sat} - T_{out}$
U	mean inlet liquid velocity
W	width of flow channel and heated walls
x	quality
x_e	thermodynamic equilibrium quality
y	coordinate perpendicular to heated wall
z	axial coordinate
z_0	axial coordinate where $U_g = U_f$
z^*	axial location for determining vapor layer thickness and critical wavelength in *Interfacial Lift-off Model*

Greek Symbols

α	vapor void fraction
δ	mean thickness of vapor layer
δ_a	mean thickness of vapor layer generated along heated wall H_a
δ_b	mean thickness of vapor layer generated along heated wall H_b
ε	heat utility ratio
η	interfacial perturbation
θ	flow orientation angle
λ_c	critical wavelength
ρ	density
ρ''	modified density
σ	surface tension
τ_i	interfacial shear stress
τ_w	wall shear stress

Subscripts

a	vapor layer generated along heated wall H_a
b	vapor layer generated along heated wall H_b
c	critical
e	Earth
exp	experimental
f	saturated liquid; bulk liquid; frictional
g	saturated vapor
ga	saturated vapor generated along heated wall H_a
gb	saturated vapor generated along heated wall H_b
i	interfacial
in	inlet to heated portion of channel
k	phase k, $k = f$ or ga or gb
out	outlet from heated portion of channel
$pred$	predicted
sat	saturation
sub	subcooling
w	heated wall (H_a or H_b)

interrupted, CHF for heat-flux-controlled surfaces is a catastrophic event, resulting in most of the heat trapped within the wall rather than rejected to the coolant, which is manifest by a rapid, unsteady rise in the wall temperature. Without a means to cut-off the power dissipation, CHF can lead to physical damage of the device being cooled by overheating or burnout. These serious consequences point to the need to both measure and accurately predict CHF.

1.3. Predictive flow boiling CHF models

Like most two-phase phenomena, researchers rely heavily on empirical correlations to predict flow boiling CHF. However, correlations are valid for specific fluids and limited ranges of operating and flow parameters, and there is great uncertainty when attempting to determine CHF for other fluids or beyond the validity range of individual parameters [7–10].

Very few theoretically based, mechanistic models have been constructed for flow boiling CHF, and these models are intended mostly for vertical upflow. As discussed in a recent review article by Konishi et al. [11] and depicted schematically in Fig. 1(a), these models are based on four competing mechanisms: *Boundary Layer Separation*, *Bubble Crowding*, *Sublayer Dryout* and *Interfacial Lift-off*. Postulated by Kutateladze and Leont've [12], the *Boundary Layer Separation Model* is based on analogy between vapor production and gas injection from a permeable wall into a turbulent boundary layer. In the same manner a turbulent boundary layer is separated when the injection velocity exceeds a threshold value, CHF is postulated to occur when the rate of vapor production perpendicular to the wall is increased to a level that greatly decreases the bulk liquid velocity near the wall, causing liquid stagnation at the wall and preventing adequate liquid replenishment of the wall. Proposed by Weisman and Pei [13], the *Bubble Crowding Model* is described by formation of a dense bubbly layer close to the wall at CHF, which renders turbulent fluctuations in the bulk liquid flow, which they postulated as the main source of liquid replenishment, too weak to penetrate the bubbly layer and reach the wall. Lee and Mudawar [14] proposed the *Sublayer Dryout Model*, which states that CHF will occur when the enthalpy of bulk liquid supplied to liquid sublayers that are trapped beneath large vapor blankets at the wall falls short of dissipating the heat supplied at the