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Identification of nucleate boiling as the dominant heat transfer mechanism during confined two-phase jet impingement

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A B S T R A C T

Thermal management of high-power electronics requires cooling strategies capable of dissipating high heat fluxes while maintaining the device at low operating temperatures. Two-phase jet impingement offers a compact cooling technology capable of meeting these requirements at a low pressure drop. Generally, confined impingement geometries are used in electronics cooling applications, where the flow is constrained between the hot surface and orifice plate. Understanding the primary heat transfer mechanisms occurring as boiling takes place on the surface during jet impingement is important, specifically under such confined conditions. In this study, heat transfer from a copper surface is experimentally characterized in both confined jet impingement and pool boiling configurations. The dielectric liquid HFE-7100 is used as the working fluid. For the jet impingement configuration, the jet issues through a single 2 mm-diameter orifice, at jet exit velocities of 1, 3, 6, and 9 m/s, into a confinement gap with a spacing of 3 jet diameters between the orifice and heat source. Additional orifice-to-target spacings of 0.5, 1, and 10 jet diameters are tested at the lowest (V_j = 1 m/s) and highest (V_j = 9 m/s) jet velocities. By incrementing the heat flux applied to the surface and observing the steady-state response at each flux, the single-phase and two-phase heat transfer performance is characterized; all experiments were carried through to critical heat flux conditions. The jet impingement data in the fully boiling regime either directly overlap the pool boiling data, or coincide with an extension of the trend in pool boiling data beyond the pool boiling critical heat flux limit. This result confirms that nucleate boiling is the dominant heat transfer mechanism in the fully boiling regime in confined jet impingement; the convective effects of the jet play a negligible role over the wide range of parameters considered here. While the presence of the jet does not enhance the boiling heat transfer coefficient, the jet does greatly increase single-phase heat transfer performance and extends the critical heat flux limit. Critical heat flux displays a linear dependence on jet velocity while remaining insensitive to changes in the orifice-to-target spacing.

1. Introduction

Two-phase jet impingement heat transfer is a viable method for thermal management of high-density electronics systems. Due to the large number of adjustable geometric parameters and operating conditions, jet impingement systems can be designed for low-pressure-drop operation while maintaining high levels of heat dissipation [1]. Confined outlet conditions, where the jet issues into a gap bounded by the heated surface and an orifice plate, are typical in electronics cooling applications due to compact packaging constraints.

The heat transfer regimes observed in two-phase jet impingement as the surface heat flux increases include: a single-phase regime before incipience, a partial boiling regime, and a fully boiling regime, followed by critical heat flux (CHF). During the partial boiling regime, boiling initiates at the periphery of the heated surface and traverses inward towards the jet axis with increasing heat flux. Consequently, both single-phase and nucleate boiling heat transfer exist simultaneously on the heated surface during this regime [2–6]. Once the boiling front reaches the jet axis, boiling occurs over the entire surface in the fully boiling regime. The fully boiling regime is a desirable operating condition, because substantial increases in heat flux are accompanied by only modest surface temperature increases in this regime. A better understanding of the dominant heat transfer mechanisms that govern performance in the fully boiling regime can help direct predictive modeling efforts. It is also important to understand the dependence of the CHF limit on the geometric parameters and operating conditions of the impinging jet, such as orifice-to-target spacing and jet velocity.

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Nomenclature

A	heat source area
Bl^*	non-dimensional heat input $qA/\dot{m}C_p(T_{sat} - T_j)$
CHF	critical heat flux
C_p	specific heat
d	orifice diameter
H	orifice-to-target spacing
$\bar{h}$	area-averaged heat transfer coefficient
h_{fg}	latent heat of vaporization
l	orifice plate thickness
$\dot{m}$	mass flow rate
p_{op}	operating pressure
q	heat flux
Ra	average surface roughness
Re	Reynolds number $(\rho v_j d/\mu)$
T	temperature
$\bar{T}$	area-averaged temperature
ΔT_{sub}	degree of subcooling $(T_{sat} - T_j)$
$\dot{V}$	volumetric flow rate
V_j	jet exit velocity

Greek symbols

μ	liquid dynamic viscosity
ρ	liquid density

Subscript

CHF	critical heat flux limit
j	jet exit condition
s	surface condition
sat	saturated condition

Two heat transfer modes are present during two-phase jet impingement: forced convection and nucleate boiling. To better understand how the impinging jet affects performance, a direct comparison with pool boiling heat transfer, where forced convection effects are absent, is useful as a benchmark. Though significant experimental work has been performed on two-phase jet impingement heat transfer, few experimental studies have directly compared jet impingement heat transfer with pool boiling. Furthermore, none have considered confined jets, which more closely resemble the conditions of flow boiling through confined channels, in which the degree of confinement is known to affect the boiling performance [7]. Katto and Kunihiro [8], Bergles and Ma [9,10], Struble and Witte [11], and Zhou and Ma [12] reported experimental comparisons of pool boiling and submerged, unconfined jet impingement. In each of these studies, the jet impingement data in the fully boiling regime coincided with the pool boiling curve. An exception found by Bergles and Ma [10] was attributed to their extensive extrapolation of the pool boiling data to draw a comparison at heat fluxes above their measured pool boiling data.

In recent work by Mira-Hernández et al. [5], a semi-empirical model for predicting area-averaged two-phase heat transfer from confined impinging jets was developed. The model, which treats single-phase and boiling regions on the surface separately, used nucleate pool boiling correlations to predict the local heat transfer coefficient in regions of the surface undergoing boiling under the jet. The model accurately predicted jet impingement experimental data in the fully boiling regime [5], without including any convection effects. This result calls for an experimental investigation that directly compares confined jet impingement boiling behavior with nucleate pool boiling heat transfer.

The present study performs experiments that characterize heat transfer from a surface under both confined jet impingement and pool boiling conditions. In the jet impingement configuration, a single 2 mm-diameter orifice is used; the jet velocity (V_j = 1 m/s, 3 m/s, 6 m/s, and 9 m/s) and orifice-to-target spacing (H/d = 0.5, 1, 3, and 10) are varied. Comparison of jet impingement heat transfer in the fully boiling regime to pool boiling provides insight into the effect of impingement on boiling heat transfer and reveals that even at high jet velocities, convective effects do not have an influence on the heat transfer coefficient. The effect of the impinging jet on critical heat flux is also discussed; trends in critical heat flux with respect to jet velocity are identified.

2. Experimental methods

Experiments are performed using a two-phase flow loop charged with the dielectric fluid HFE-7100 [13]. The test section is reconfigurable to allow characterization of both jet impingement and pool boiling heat transfer. Details of the experimental facility, procedures, and data reduction are described in this section.

2.1. Flow loop

The flow loop used to perform the experiments is described in detail in Ref. [14] and is shown schematically in Fig. 1(c). A magnetically coupled gear pump circulates fluid through the flow loop. The flow rate is coarsely set by adjusting the rotational speed of the pump and finely tuned by metering the flow through a bypass loop and the test section. Mass flow rate is measured by a Coriolis flow meter (CMFS015, Emerson) with ±0.1% accuracy. Any plasticizers or organic contaminants are removed from the working fluid using an activated carbon filter while particulates are removed using 0.5 µm particulate filters. The liquid subcooling is controlled by adjusting the voltage supplied to the 1.2 kW inline preheater. Fluid exits the test section and is returned to the fluid reservoir; for degassing purposes, the reservoir is equipped with a 1 kW immersion heater and two Graham reflux condensers. A copper-finned liquid-to-air heat exchanger equipped with a voltage-regulated fan is used to cool the fluid before the pump inlet; this prevents cavitation in the pump and provides greater control over the test section inlet temperature.

2.2. Test section

The test section, modified from the original heater assembly used in Ref. [4], is shown schematically in Fig. 1(a). The side walls are made of polyether ether ketone (PEEK) for thermal insulation, while the front and back walls are made of polycarbonate for optical access, allowing high-speed visualizations through the side of the test section. Fluid enters the top of the circular cross-section plenum and passes through two screens and a honeycomb to straighten the flow. The test section inlet pressure and temperature are measured by a pressure tap and a T-type thermocouple placed close to the orifice plate inside the plenum. The circular plenum is sealed to the top of the test section by an O-ring, allowing the plenum to translate vertically so that its position can be adjusted.