Advances in
HEAT TRANSFER

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PREFACE

The serial publication *Advances in Heat Transfer* is designed to fill the information gap between the regularly scheduled journals and university-level textbooks. The general purpose of this publication is to present review articles or monographs on special topics of current interest. Each chapter starts from widely understood principles and brings the reader up to the forefront of the topic in a logical fashion. The favorable response by the international scientific and engineering community to the volumes published to date is an indication of how successful our authors have been in fulfilling this purpose.

The Editors are pleased to announce the publication of Volume 23 and wish to express their appreciation to the current authors, who have so effectively maintained the spirit of this serial publication.

CONTENTS

Jet Impingement Boiling

D. H. Wolf, Frank P. Incropera, and Raymond Viskanta

Radiative Heat Transfer in Porous Media

Massoud Kaviany and B. P. Singh

Fluid Flow, Heat, and Mass Transfer in Non-Newtonian Fluids: Multiphase Systems

R. P. CHHABRA

Advances in Heat Flux Measurements

THOMAS E. DILLER

One- and Two-Equation Models for Transient Diffusion Processes in Two-Phase Systems

MICHEL QUINTARD AND STEPHEN WHITAKER

Jet Impingement Boiling

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I. Introduction

Increasing needs for high-heat-flux convective cooling of solids have directed considerable effort toward the development of effective cooling schemes. In some industries experiencing rapid technological growth, such as high-speed computing and data processing, thermal engineering could foreseeably become the factor which limits further growth. This statement is certainly true for the field of microelectronics, where the seemingly inexorable trend of achieving ever larger scales of circuit integration is straining the capabilities of existing high-flux cooling technologies (Incropera [1, 2]). In such cases, cooling requirements are exacerbated by restrictions on available space, choice of coolant, local environmental conditions, and maximum allowable surface temperatures. Likewise, the production of steel, aluminum, and other metals having desired mechanical and metallurgical properties requires accurate temperature control during processing (Viskanta and Incropera [3]). The surface temperature and heat flux are typically very large and acceptable cooling times are relatively short.

One means of achieving very high rates of heat transfer is through the use of impinging liquid jets. Heat transfer coefficients for systems of this type typically exceed 10,000 W/m^2-°C for single-phase convection and are much larger in the presence of boiling. Impinging liquid jets have found usage in many industrial applications, in both submerged (liquid-into-liquid) and free-surface (liquid-into-gas) arrangements.

Because of the attractiveness of jet impingement cooling for high-heat-flux applications, numerous studies have been performed for both single- and

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two-phase conditions. This statement is particularly true for jet impingement boiling, which is distinguished by its ability to dissipate heat fluxes at the high end of the cooling spectrum. However, although the related literature is extensive, ambiguities and contradictions do exist, and there is need for a comprehensive review to assess the state of current knowledge. Such a review has been performed in order to identify strengths and weaknesses in the existing knowledge base and to identify areas requiring additional research. In so doing, every attempt has been made to retrieve and review all of the archival literature on the topic of jet impingement boiling, regardless of source.

II. Background

A. Jet Impingement Hydrodynamics

This review addresses liquid jets with continuous cross sections, thereby excluding spray and droplet impingement studies. Throughout this review, jet configurations will be delineated into the five categories of free-surface jets, plunging jets, submerged jets, confined jets, and wall jets. These configurations are shown schematically in Fig. 1. The free-surface jet is injected into an immiscible atmosphere (liquid into gas), and the liquid travels relatively unimpeded to the impingement surface. The plunging jet differs only in that it impinges into a pool of liquid covering the surface, where the depth of the pool is less than the nozzle-to-surface spacing. The submerged jet is injected directly into a miscible atmospshere (liquid into liquid), and the confined jet is injected into a region bounded by the impingement surface and nozzle-plate. The wall jet flows parallel to the surface and occurs in both free-surface and submerged configurations.

The first four configurations induce flow fields on the impingement surface which are qualitatively similar, and Fig. 2 depicts representative conditions for a planar, free-surface jet. The inviscid pressure and streamwise velocity distributions for a uniform jet velocity profile (Milne-Thomson [4]) are also shown. The pressure is a maximum at the stagnation point due to the dynamic contribution of the impinging jet. With increasing streamwise distance, the pressure declines monotonically to the ambient value. Conversely, the streamwise velocity is zero at the stagnation point and increases to the velocity of the jet with increasing distance along the surface. To clarify the discussion of boiling at various locations on the impingement surface, the flow has been demarcated into stagnation, acceleration, and parallel-flow regions. The stagnation region coincides with that of the impinging jet, in both size and location ($x/w_j < 0.5$), and contains a nearly linear increase in

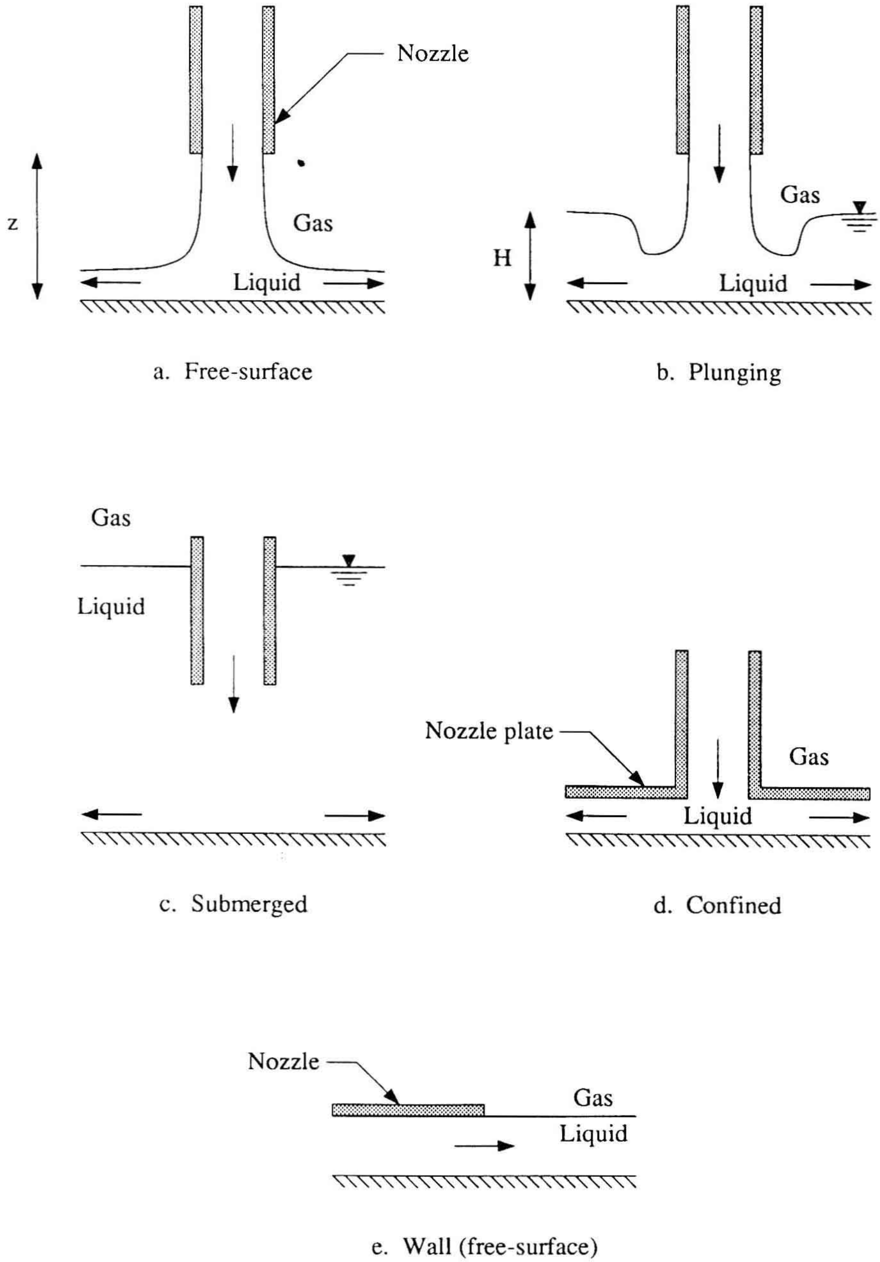

FIG. 1. Schematic of the various jet configurations.

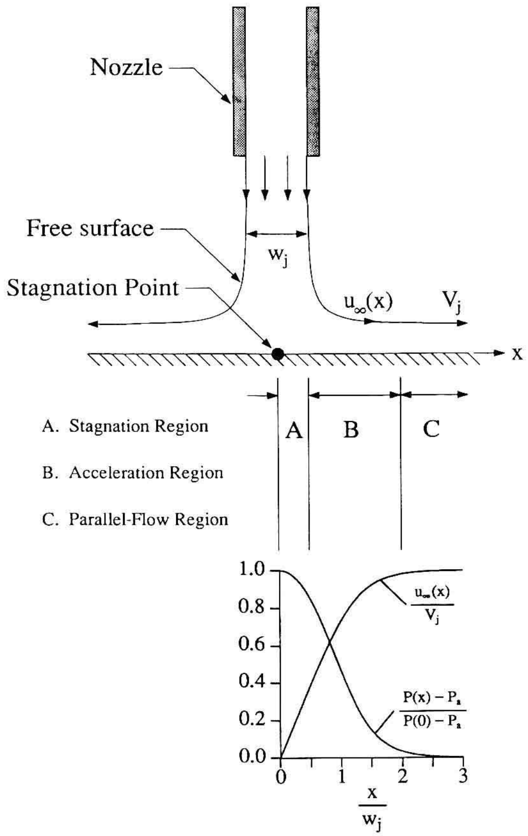

FIG. 2. Inviscid pressure and velocity distributions for a planar, free-surface jet with uniform velocity profile, along with the respective flow regions.

the streamwise velocity. Within the acceleration region ($0.5 \leq x/w_j \lesssim 2$), the fluid continues to accelerate and approaches the jet velocity to within a few percent. For $x/w_j \gtrsim 2$ (parallel-flow region), the streamwise velocity is essentially that of the jet and the hydrodynamic effects of impingement are no longer realized within the flow.

Exceptions to this scenario can occur for plunging and submerged configurations when the exchange of momentum between the jet and miscible fluid is large, causing the flow to decelerate and expand laterally prior to impingement. Such cases typically occur for low jet velocities and/or large nozzle-to-surface spacings (or large pool heights in the case of plunging jets), resulting in lower stagnation pressures and a spatially broader velocity and pressure distribution than that shown in Fig. 2 (see, for example, Gardon and

Akfirat [5]). Exceptions can also occur for confined arrangements, where the nozzle-to-surface spacing is so small that the fluid accelerates further due to a decrease in flow cross-sectional area (see, for example, Miyazaki and Silberman [6]).

In addition to governing the hydrodynamics, the pressure distribution controls the local saturation conditions along the surface. For a saturated water jet with an ambient pressure of $P_a = 1.013$ bar and an impingement velocity of $V_j = 10$ m/s, the resulting saturation temperature at the stagnation point would be 111°C ($P = P_s = 1.492$ bar), compared to 100°C several jet dimensions downstream. Consequently, variations in T_{sat} cause attendant variations in the degree of subcooling ΔT_{sub} and wall superheat ΔT_{sat}. Mudawar and Wadsworth [7] have addressed this issue for a confined jet, where an additional decrease in pressure can be realized for flow between the impingement surface and confining wall. Since the pressure distribution along the surface for a confined jet is a function of the velocity and nozzle-to-surface spacing, the local subcooling will exhibit a similar dependence. For small nozzle-to-surface spacings and large velocities, Mudawar and Wadsworth have shown the streamwise variation in ΔT_{sub} to affect the critical heat flux. Most of the other publications cited herein have based saturation conditions on the ambient pressure. However, for nonconfined arrangements where the heater size greatly exceeds that of the nozzle, ambient conditions exist over nearly the entire surface (excluding the region within several jet dimensions of the stagnation point). In such cases, the use of ambient saturation conditions is justifiable. For the purpose of this review, quantities such as the wall superheat and subcooling will be based on the saturation temperature corresponding to the ambient pressure (P_a); special mention is made of the few cases in which this does not apply.

In each of the jet configurations, the velocity can vary between the nozzle exit (V_n) and impingement surface (V_j). For free-surface jets, gravity accelerates the flow for downward impingement and decelerates it for upward impingement. The nozzle and impingement velocities are related, to a good approximation, by the expression $V_j = (V_n^2 \pm 2gz)^{1/2}$, where differences in V_n and V_j become negligible for large V_n or small z. Indirectly, gravity also causes the jet dimension to vary in order to satisfy continuity. For plunging, submerged, and confined arrangements with large nozzle-to-surface spacings (or large pool heights), momentum exchange, initially occurring at the perimeter of the jet, will ultimately move inward and retard the velocity at the jet's centerline with increasing distance from the nozzle. The axial distance over which the centerline velocity remains equal to that of the nozzle exit (the so-called potential core) typically ranges from 5 to 8 nozzle dimensions. Hence, spacings outside this range will cause the impingement velocity to be lower than the corresponding nozzle velocity. The majority of the impingement boiling literature has made no distinction between the two values and has used the nozzle velocity (V_n) to compare and correlate data. Several investigators, reporting results for free-surface jets, have accounted for gravity and presented their data in terms of the velocity (V_j) and jet dimension at the point of impingement. In this review, the term *jet velocity* will generally refer to conditions at the nozzle exit; special mention will be made of the few exceptions for which the impingement velocity V_j is intended.

B. JET IMPINGEMENT BOILING

The most descriptive representation of boiling data is obtained by plotting the surface heat flux, q'', as a function of the difference between the wall and saturation temperatures (the wall superheat, ΔT_{sat}), yielding the *boiling curve* shown schematically in Fig. 3 for a saturated liquid.

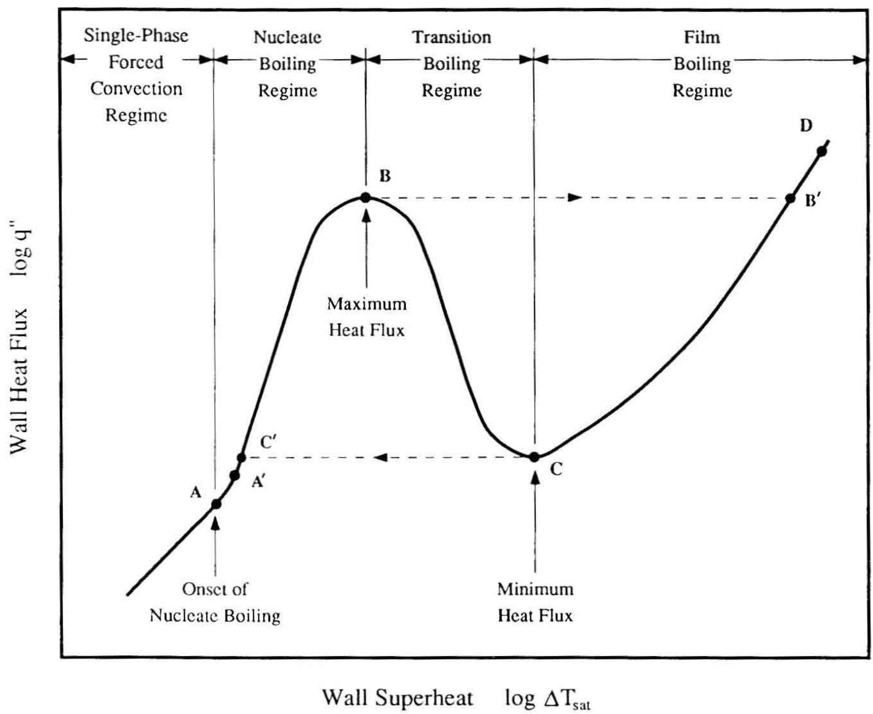

FIG. 3. Schematic of the boiling curve for a saturated liquid.

1. *Single-Phase Forced Convection*

The single-phase forced-convection regime represents heat transfer in the absence of boiling, and the relationship between the heat flux and wall superheat is governed by Newton's law of cooling, $q'' = h(\Delta T_{sat} + \Delta T_{sub})$. For jet impingement, the convection coefficient (h) varies over the surface due to hydrodynamic variations in the streamwise direction. In addition to effects caused by the inviscid flow field discussed in Section II.A, factors such as boundary layer development and transition also affect the distribution of the convection coefficient. As a result, either the wall temperature (heat flux constant) or heat flux (wall temperature constant) will also vary. Single-phase jet impingement heat transfer is extensively discussed in existing surveys of experimental and numerical investigations (Martin [8]; Downs and James [9]; Polat *et al.* [10]; Viskanta and Incropera [3]).

2. *Nucleate Boiling*

The forced-convection regime extends to wall temperatures that exceed that of saturation, and the nucleate boiling regime exists in the temperature range between points A and B. Although the formation of vapor within surface cavities commences at T_{sat}, temperatures above this value are required for the vapor to emerge and form a thermally stable bubble. Point A marks the onset of nucleate boiling (ONB), where discrete bubbles begin to detach from the surface and enhance the local fluid motion, causing the convection coefficient to increase. With increasing heat flux or wall temperature, the generation of vapor progresses from a few relatively small bubbles at point A to many larger bubbles coalescing near point B. The points A′ and B form the extremes of what this review refers to as the *fully developed nucleate boiling region*. Point A′ has been chosen because it marks the beginning of the linear nucleate boiling region in log–log coordinates ($q'' \sim \Delta T_{sat}^{n}$) and the end of the transition from single-phase convection. This definition has been chosen as a matter of convenience in order to convey the reported results in a clear, well-defined manner. Although a universal definition of fully developed nucleate boiling (FNB) does not appear to exist, the term is commonly associated with behavior that is insensitive to conditions in the bulk liquid, such as velocity or subcooling (Collier [11]). Most of the data to be discussed embody this definition, but exceptions are shown to exist.

The attractive feature of nucleate boiling is the large increase in heat transfer that accompanies only moderate changes in the surface temperature. Consequently, it is the desired region of operation for many high-heat-flux cooling applications. However, controlled cooling depends on accurate knowledge of the location of point B, commonly referred to as the maximum or critical heat flux (CHF). The term *maximum heat flux* will be used for boiling curves obtained through a quench. The large degree of bubble coalescence ultimately prevents liquid from reaching the surface, and the vapor forms an insulative barrier to heat transfer. Depending on whether the surface boundary condition is heat flux-controlled or temperature-controlled, a large increase in ΔT_{sat} (B to B′) or decrease in q'' (B to C) will result, respectively.

Due to the wall temperature (heat flux constant) or heat flux (wall temperature constant) distribution on the impingement plane, both single-phase forced convection and nucleate boiling can occur simultaneously at different locations on the surface. Vader *et al.* [12], for example, have shown through local temperature measurements and high-speed photography that finite regions of nucleate boiling can develop amidst surrounding regions of single-phase convection for a free-surface, planar jet. They showed boiling to initiate near the transition from a laminar to a turbulent boundary layer (a local maximum in temperature for a constant heat flux surface) and subsequently propagate upstream and downstream to envelop the entire surface with increased heating. Cho and Wu [13] similarly reported single-phase convection at the center of the heater with nucleate boiling around the perimeter for a free-surface, circular jet. With increased heating, the nucleate boiling region propagated inward toward the stagnation point.

Observations of the heating surface near the critical heat flux (point B) have consistently reported blanketing to initiate at the perimeter of the heated section (Katto and Kunihiro [14]; Katto and Ishii [15]; Monde and Katto [16]; Monde [17]; Ma and Bergles [18]; Cho and Wu [13]). Blanketing of the inner surface area was generally reported to occur either immediately thereafter, without any additional heating, or upon a marginal increase in the heat flux. In either case, however, the vapor blanket at the heater's edge causes a substantial increase in the local surface temperature, which eventually propagates inward toward the stagnation point, inducing additional blanketing (heat flux-controlled boundary condition).

3. *Transition Boiling*

The transition boiling regime represents conditions where unstable vapor blankets form and collapse accompanied by intermittent wetting of the surface. The regime is demarcated by point B, the maximum heat flux, and point C, the minimum heat flux and temperature (q''_{min}, T_{min}). The q''-ΔT_{sat} relationship within the regime depends on the surface boundary condition. The temperature-controlled condition follows the solid curve (B to C), and the heat flux-controlled condition follows the dashed lines (B to B′ or C to C′), depending on whether the heat flux is increasing or decreasing. With one possible exception (Miyasaka *et al.* [19]), operation in the transition boiling regime for impinging jets has been limited to temperature-controlled conditions obtained through transient quenches (here the term temperature-*controlled* is used loosely). The primary focus of these investigations has been the measurement and prediction of q''_{min}, T_{min}, and T_{wet} (the wetting temperature). In the quench of a specimen, the temperature and heat flux will decline from point D to point C. However, the initiation of liquid–surface contact (wetting) will occur at a temperature (T_{wet}) that is somewhat larger than T_{min}, thereby inducing the local minimum. While q''_{min} and T_{min} result directly from measurement of the boiling curve, T_{wet} can be obtained from measurement of the surface temperature and simultaneous measurement or observation of the liquid–surface contact. Use of the variables T_{min} and T_{wet} is intended to differentiate between these experimental approaches.

4. *Film Boiling*

The film boiling regime (C to beyond point D) represents heat transfer from the surface to the liquid across a vapor film. The mode of heat transfer is primarily forced convection of the vapor, with radiation becoming dominant at higher surface temperatures. For impinging jets, film boiling can often accompany other regimes of boiling on the same surface. Observations of a transient quench with an impinging jet reveal that, at low subcoolings and high plate temperatures, the jet is isolated from the surface by the vapor layer. As the plate temperature declines, the jet penetrates the vapor and wets the surface surrounding the stagnation point while film boiling persists at locations farther downstream (Kokado *et al.* [20]).

5. *System-Specific Effects*

Heat transfer associated with each of the modes of boiling is sensitive, in varying degrees, to the experimental conditions used in the measurement. Factors such as surface finish (Rohsenow [21]), surface contamination (Joudi and James [22]), noncondensible gases (Fisenko *et al.* [23]), heater thickness (Guglielmini and Nannei [24]), heater material (Klimenko and Snytin [25]), method of heating (ac or dc powered) (Houchin and Lienhard [26]), and the type of experiment conducted (steady state or transient) (Bergles and Thompson [27]) have all been shown to affect one or more of the modes of boiling. However, the foregoing results pertain mainly to pool boiling, and there are, in fact, few data for forced-convection boiling. Hence, no attempt has been made in the following review to interpret results in terms of such system-specific effects. Nevertheless, details of each experimental investigation have been provided in tabular form.