Int. J. Heat Mass Transfer. Vol. 29, No. 8, pp. 1095–1101, 1986
Printed in Great Britain

0017-9310/86 $3.00+0.00
Pergamon Journals Ltd.

Jet impingement nucleate boiling

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Abstract—The characterististics of nucleate boiling with jet impingement were investigated. Included are the effects of velocity, subcooling, flow direction and surface condition on fully developed boiling and on the correspondence of the extrapolation of pool boiling with developed jet boiling. Incipient boiling and partial boiling are in accordance with the Bergles and Rohsenow correlations. In general, much of the experience and procedures for forced convection boiling inside tubes can be applied to jet impingement boiling.

1. INTRODUCTION

FORCED convection boiling under subcooled and low quality conditions is a highly effective means of cooling that has applications ranging from pressurized water nuclear reactors to large scale digital computers. High heat fluxes can be accommodated with moderate temperature differences because of efficient nucleate boiling, particularly under high velocity and high subcooling.

This mode of heat transfer has been extensively studied for flow inside tubes and there is reasonable agreement regarding the mechanisms and design procedures for the principal phenomena: incipient boiling, partial boiling, fully developed boiling (including effects of surface conditions, dissolved gas, subcooling, velocity and fluid properties), relation of forced convection boiling to pool boiling, critical heat flux and flow stability [1–3].

Jet impingement cooling has attracted much interest in recent years for both gas and liquid jets. Very high heat fluxes can be accommodated with subcooled or saturated jets. Single jets are effective for small heated surfaces, while multiple jets can be arranged to cool large surfaces.

A main concern is that the critical heat flux (CHF) is not exceeded in systems having imposed heat fluxes. The recent survey by Katto [4] describes CHF mechanisms with single jets and summarizes the rather extensive data and numerous correlations.

Less attention has been given to the jet boiling heat transfer characteristics prior to CHF. The prediction of the complete nucleate boiling curve is particularly important for microelectronic cooling applications because component performance is very sensitive to temperature level.

It was the intent of this study to clarify the nucleate boiling heat transfer characteristics of subcooled and saturated jets. An inert dielectric liquid was used and the jet was submerged as in direct immersion cooling of microelectronic components. Preliminary results have been reported in ref [5].

2. APPARATUS AND PROCEDURE

Apparatus

The overall apparatus is shown in Fig. 1. The working fluid chosen was R-113 because it is inexpensive and easy to handle at room temperature yet has similar characteristics to the fluorocarbon liquids used for microelectronic cooling. The R-113 was circulated in a closed loop that had provision for filtering, metering, preheating and cooling.

The sides and bottom of the test chamber shown in Fig. 2 were constructed of aluminum alloy. A visual port was provided for observation of the test section. In addition, the bottom section of the cover was transparent. A flexible polyethylene tube joined the two sections of the cover. The jet tube of inside diameter 1.07 or 1.81 mm was attached to the upper aluminum section and made stationary. The chamber could be adjusted by means of a milling machine table so that the position was altered in three dimensions with respect to the test section. The placement was accomplished to within 0.025 mm.

The fluid in the chamber was close to atmospheric pressure. Four immersion heaters and side guard heaters helped to maintain the desired pool temperature. When boiling under saturated conditions, the vapor condensed in the reflux condenser. Under subcooled conditions the jet throughflow was sufficient to maintain the pool at the desired temperature level. The heat fluxes in pool boiling were so low that the entire boiling curve could be generated without appreciably raising the temperature. Thermocouples were provided to measure the pool temperature and the temperature of the liquid in the jet tube. The entire chamber was insulated except when the test section was viewed.

The test section required considerable development effort. The final arrangement shown in Fig. 3 involved

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NOMENCLATURE

h	heat transfer coefficient	r_c	radius of largest active nucleation site
h_{fg}	enthalpy of vaporization	T_{avg}	average temperature, $(T_w + T_s)/2$
k_l	liquid thermal conductivity	T_b	bulk liquid temperature
q''	heat flux	T_s	saturation temperature
q''_c	single-phase convection heat flux, equation (1)	T_w	wall temperature
		ΔT_{sub}	subcooling, $T_s - T_b$
q''_{fd}	heat flux in fully developed boiling	u_0	velocity of jet
$q''_{fd,i}$	fully developed heat flux at superheat for incipient boiling	v_{fg}	specific volume change during vaporization
q''_i	heat flux at incipient boiling	σ	surface tension.

a strip of 10-μm-thick constantan foil with heated sections of 5×5 mm or 3×3 mm exposed to the coolant. The strip on either side of this active section was soldered to copper bus blocks, which were in turn connected to power leads. The active section of the foil was cemented to a bakelite block inserted between the copper blocks. The assembly was cemented in a plexiglas disc of 45 mm maximum diameter. The disc was inserted in the aluminum housing and retained by a plastic cylinder with a screwed flange. The outer surface of the cylinder was fitted with a guard heater. The temperature of the heater's inner surface at the midpoint was measured by a 40-gage iron–constantan thermocouple, which was electrically insulated from the heater yet in close thermal contact.

Direct-current power to the test section was supplied by a 50-A power supply. The voltage drop across the test section was measured at the copper bus blocks.

Procedure

The area of the heat transfer surface was carefully measured for each test section assembly. The surface was left in the original, highly polished condition and simply cleaned with acetone.

Before testing, the R-113 was thoroughly degassed by raising the temperature in the chamber to the saturation condition and allowing the noncondensables to vent through the condenser.

When the jet was oriented normal to the heater, the nozzle-to-heater spacing was fixed at twice the jet tube's inner diameter. The nozzle centerline coincided with the midpoint of the heater or was translated horizontally two diameters from the midpoint. The latter position provided the 'parallel flow' at the midpoint.

Most tests progressed from low, nonboiling power to high, pre-CHF power and back to low power. All measurements were recorded with precision digital voltmeters.

Data reduction

The heat flux was calculated from the measured values of the voltage drop across the test section, current and the area of one side of the heated surface. Calculations established that for these forced convection conditions the heat flux at the center of the heater was not significantly reduced by conduction to the substrate, conduction to the copper bus blocks or by thermocouple lead losses. The temperature reading was corrected for the very small temperature drop across the foil to get the surface temperature T_w. The saturation temperature T_s was calculated from the local pressure. The results are generally presented in

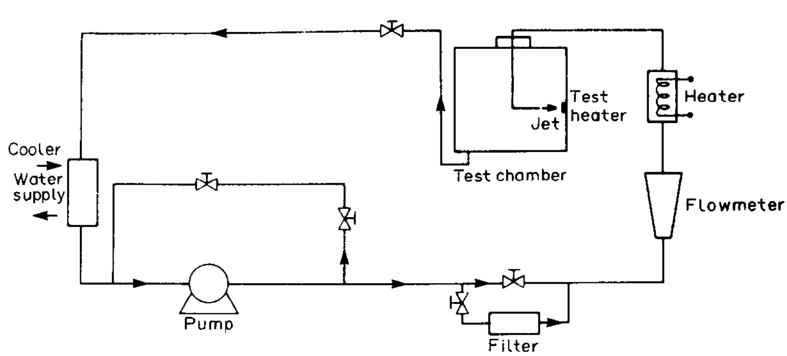

FIG. 1. Schematic layout of flow loop.

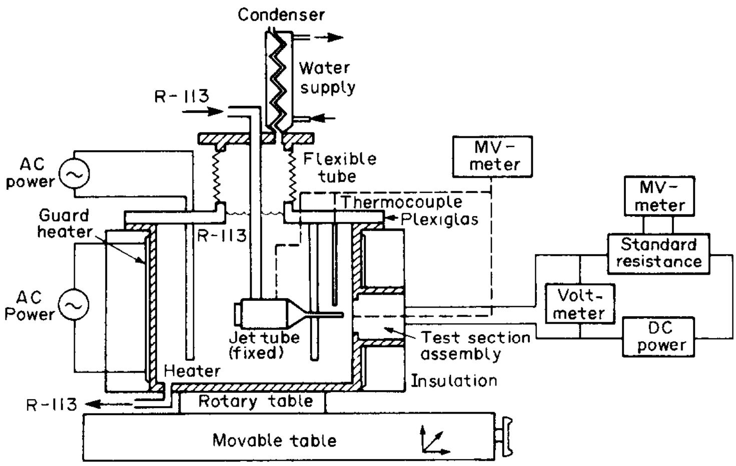

FIG. 2. Details of test chamber and instrumentation.

terms of the boiling curve, q'' vs $T_w - T_s$, at a given velocity and subcooling.

3. EXPERIMENTAL RESULTS AND DISCUSSION

Basic forced convection behavior and effect of velocity

Over 30 boiling curves were generated with 10 different test sections for saturated or subcooled conditions with a range of velocities. Typical boiling curves are shown in Fig. 4. The solid lines denoted as equation (5) will be explained later in the paper. In the q''–$(T_w - T_s)$ plane, as heat flux is increased, the slope increases gradually according to the single-phase relationship

$$q''_c = h(T_w - T_b) = h[(T_w - T_s) + (T_s - T_b)]. \quad (1)$$

For the lower jet impingement curve at a superheat of about 30 K, there is a slight dislocation in the curve that suggests a temperature overshoot prior to the inception of boiling. The slope then increases sharply, and ultimately a log–linear curve characteristic of fully developed boiling is obtained. The test was not continued to burnout. The decreasing heat flux curve follows essentially the same path, except that the small temperature overshoot at incipient boiling is not present.

Figure 4 also shows the effect of velocity. The superheat at low heat flux is reduced because of the high heat transfer coefficient, in accordance with equation (1). At high heat flux, the curves merge into a common, fully developed boiling asymptote. There is virtually no evidence of temperature overshoot for the higher velocity.

It is of interest to compare the jet boiling curves with those for pool boiling, which represents the limiting case of zero velocity. The lowest curve on Fig. 4, for subcooled pool boiling, has a distinct, fully developed portion that can be extrapolated into the range of the subcooled jet data. For this set of data, at least, the extrapolated pool boiling data do not coincide with the common asymptote of the fully

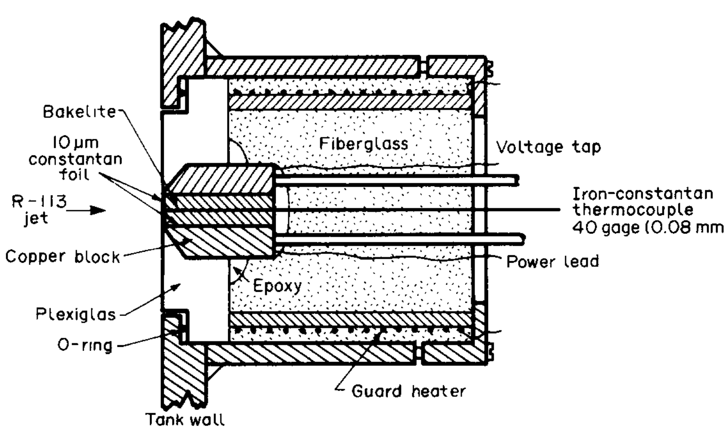

FIG. 3. Details of electrically heated test section.

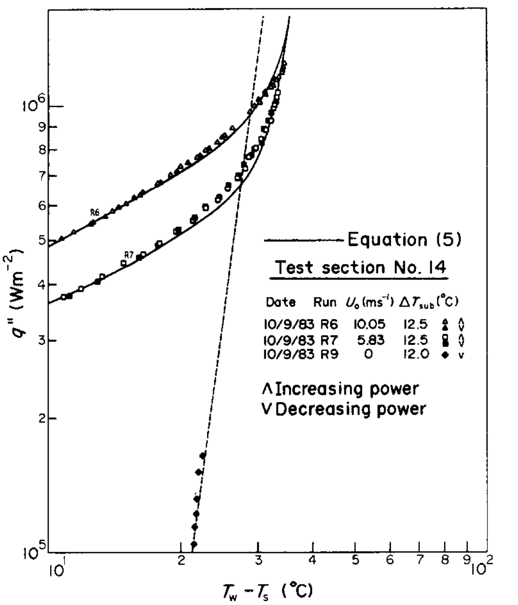

FIG. 4. Boiling curves for pool boiling and jet impingement boiling at various velocities.

developed jet boiling data. Of course, it should be recognized that the extrapolation is long and that a slight change in the slope of the pool boiling curve would bring about better agreement.

Effect of subcooling

The effect of subcooling is depicted in Fig. 5. As a fixed velocity, the superheat at low heat flux is reduced with increasing subcooling in accordance with equation (1). There is a pronounced temperature overshoot for the saturated case, but the overshoot is much less for the subcooled case. The high heat flux portions of the curves shift slightly to the left with increasing subcooling, in accordance with results reported recently by Del Valle M. and Kenning [6] for channel flow of water. The pool boiling curves also shift with subcooling, and the trend agrees with that reported by Duke and Schrock [7] (flat plane facing up) rather than with that demonstrated by Bergles and Rohsenow [8] (small diameter horizontal cylinder). The pool boiling shift seems to coincide with the shift in the asymptotes of the jet impingement boiling curves. In this case the extrapolated pool boiling data coincide approximately with the fully developed jet boiling data.

Normal flow vs parallel flow

Several tests were run with the nozzle displaced so that the measuring station was located in the wall jet region rather than at the stagnation point. The general trend was for the fully developed boiling curves for normal and parallel flow to merge at low velocity. At high velocity, the normal flow curve was displaced to the right of the parallel flow curve. Examples of this behavior are shown in Fig. 6.

Effect of surface condition

In spite of the care that was taken to assure uniform surface conditions—by cleaning and degassing—significant day-to-day shifts in the fully developed boiling curves were observed. For the comparisons presented in Figs. 4–6, only runs taken with the same test section on the same day are presented.

To illustrate the extent of the boiling curve shifts, the data are collected in Fig. 7. The general trend is toward an improvement in boiling performance with time. This is consistent with the work of Joudi and James [9] who studied pool boiling of R-113. They

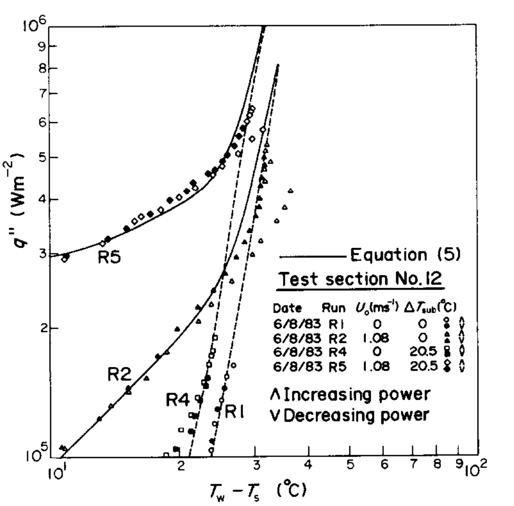

FIG. 5. Boiling curves for pool boiling and jet impingement boiling at various subcoolings.

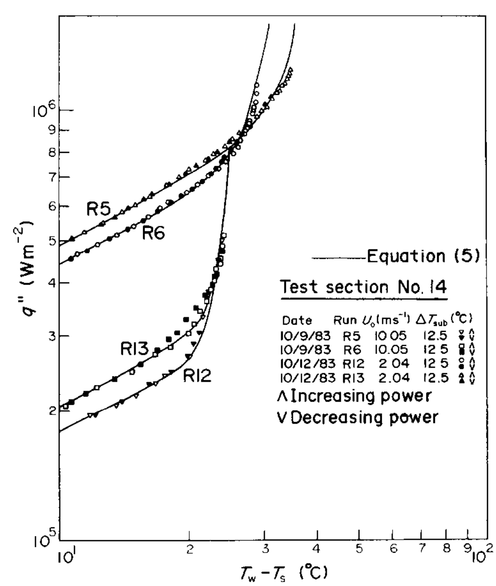

FIG. 6. Effect of flow direction on jet impingement boiling.

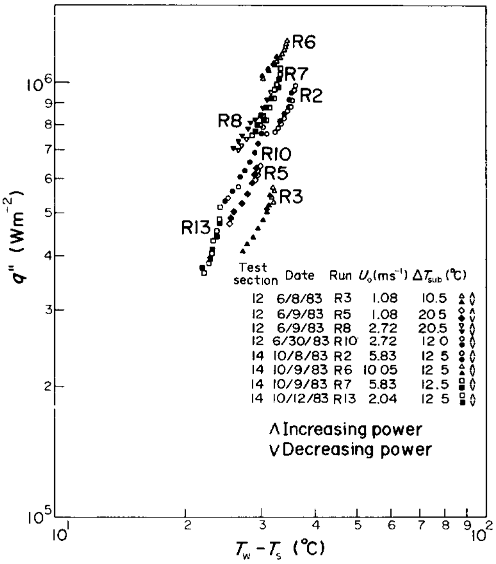

FIG. 7. Composite of fully developed boiling data showing effect of surface condition.

observed 'rejuvenation' of surfaces due to dissolution of contaminants that appreciably reduced cavity size of filled cavities.

4. INCIPIENT BOILING

It is of interest to examine incipient boiling from the highly polished constantan heaters since they approximate the smooth finish of silicon microelectronic chips. The expectation is that a full range of cavity sizes is not available for nucleation.

As proposed by Bergles and Rohsenow [8] incipient boiling is defined as the first significant increase of the heat transfer coefficient from that predicted for single-phase forced convection. A typical example is shown in Fig. 8. An incipient boiling point is observed at a superheat of about 20°C for increasing heat flux. When the heat flux is decreased, boiling persists until a superheat of about 17°C. For purposes of characterizing the surface, the latter 'incipient boiling point' is of interest since it represents stable and established boiling behavior.

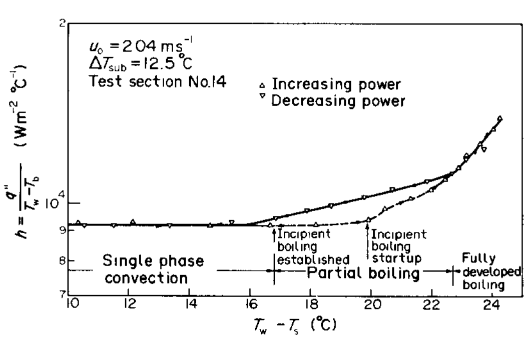

FIG. 8. Experimental determination of incipient boiling.

The theoretical basis for the inception of nucleate boiling was presented by Bergles and Rohsenow [8] and Davis and Anderson [10]. The analyses lead to curves defining incipient boiling in the heat flux–wall superheat plane. The predictions are quite sensitive to assumptions regarding the pressure–temperature dependence along the saturation line. Kim [11] recently showed that the following relationships are accurate for R-113 at atmospheric pressure.

For the case where active nucleation sites of all sizes are present:

$$q_i'' = \frac{h_{fg}k_l}{8T_{avg}v_{fg}\sigma}(T_w - T_s)^2. \qquad (2)$$

For the case where active nucleation sites of equivalent radii less than r_c are present:

$$q_i'' = \frac{k_l}{r_c}(T_w - T_s) - \frac{T_{avg}v_{fg}2\sigma k_l}{h_{fg}r_c^2} \qquad (3)$$

where $T_{avg} = (T_w + T_s)/2$.

The experimentally determined incipient boiling points are compared with equations (2) and (3) in Fig. 9. The data clearly suggest that there is a maximum active cavity size that lies within the range r_c = 0.4–0.6 μm.

In order to corroborate these data, and thereby the theory, the surface was examined with a scanning electron microscope. The surface was coated with a 300-Å layer of gold to improve imaging quality and the photomicrographs were taken at 3000 ×. It was observed that the pits and depressions on the surface

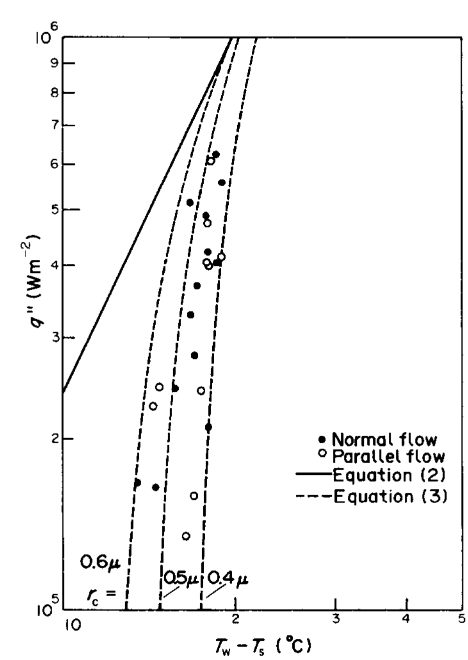

FIG. 9. Comparison of incipient boiling data with theoretical predictions.

were in the range 0.2–1.2 μm, the characteristic depression being the diameter at the surface of a circular pit or the width of a scratch. These observations are consistent with the measurements of incipient boiling.

Partial nucleate boiling

With the incipient boiling point located it is possible to construct the boiling curve in the partial boiling region of the 'knee' of the boiling curve. This requires the appropriate fully developed boiling curve and its extrapolation. For convenience, the fully developed boiling curve is taken as log–linear:

$$q''_{\mathrm{fd}} \sim (T_{\mathrm{w}} - T_{\mathrm{s}})^n. \quad (4)$$

The interpolation method proposed by Bergles and Rohsenow [7] was used to correlate the partial boiling data:

$$q'' = [q''^{2}_{\mathrm{c}} + (q''_{\mathrm{fd}} - q''_{\mathrm{fd,i}})^2]^{1/2} \quad (5)$$

where q''_{c} is given by equation (1). The value of $q''_{\mathrm{fd,i}}$ is the fully developed heat flux at the superheat corresponding to the inception of boiling. Basically, this equation reduces to single-phase, forced convection at low heat flux and blends into the fully developed boiling curve at high heat flux.

The boiling curves predicted according to equation (5) are superimposed on the data presented in Figs. 4–6. The agreement is seen to be quite satisfactory.

5. CONCLUSIONS

The characteristics of nucleate boiling with jet impingement have been clarified. The R-113 jet impingement system developed for this study was quite suitable for determining the local heat transfer characteristics of small heaters. Boiling curves for various velocities merge into a common, fully developed boiling asymptote, whereas the curves for different subcoolings are displaced slightly. Extrapolated pool boiling curves match the jet boiling asymptotes for various subcoolings but not for various velocities. The boiling curves for normal and parallel flow merge at low velocity but not at high velocity. All boiling curves are influenced by surface condition.

The forced convection incipient boiling model of Bergles and Rohsenow [8] is accurate for decreasing heat fluxes provided allowance is made for the rather small cavities on the highly polished heaters. The partial boiling correlation of Bergles and Rohsenow accurately describes the low flux boiling data. In general, much of the experience and procedures developed for forced convection boiling inside tubes can be applied to jet impingement boiling.

Acknowledgements—This study was supported by IBM Corp., Data Systems Division, Poughkeepsie, New York, under the technical cognizance of Mr R. C. Chu. The authors are indebted to Mr R. D. Steed for assistance with the experimental facility and to Mr D. T. Xhang for obtaining the photomicrographs. The assistance of the Office of Editorial Services of the Engineering Research Institute is also appreciated.

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EBULLITION NUCLEEE AVEC IMPACT DE JET

Résumé—On étudie les caractéristiques d'ebullition nucléée avec impact de jet. On tient compte des effets de vitesse, de sous-refroidissement, de direction de l'écoulement, de la condition pariétale sur l'ébullition pleinement développée et sur la correspondance de l'extrapolation de l'ébullition en réservoir avec l'ébullition en jet développé. Les ébullitions commencante et partielle sont en accord avec les formules de Bergles et de Rohsenow. En général, la plupart des expériences et des procédures pour l'ébullition avec convection forcée dans les tubes peut être appliquée à l'ébullition avec impact de jet.

BLASENSIEDEN BEI PRALLSTRAHLEN

Zusammenfassung—Das Blasensiedeverhalten bei Prallstrahlen wurde untersucht. Eingeschlossen sind die Einflüsse von Geschwindigkeit, Unterkühlung, Strömungsrichtung und Oberflächeneigenschaften auf das voll ausgebildete Sieden und auf die Übereinstimmung der Extrapolation des Behältersiedens auf das Sieden bei Prallstrahlen. Siedebeginn und partielles Sieden stimmen mit der Korrelation von Bergles und Rohsenow überein. Allgemein kann ein großer Teil der Erfahrungen und Vorgehensweisen vom Strömungssieden im Rohr auf das Sieden von Prallstrahlen angewendet werden.

ПУЗЫРЬКОВОЕ КИПЕНИЕ ПРИ НАТЕКАНИИ СТРУИ НА ПОВЕРХНОСТЬ

Аннотация—Исследованы характеристики пузырькового кипения при падении струи на поверхность. В рассмотрение включено влияние скорости, недогрева, направления течения и состояния поверхности на полностью развитое кипение и на соответствие экстраполяции кипения в большом объеме на развитое струйное кипение. Начинающееся и частичное кипения согласуются с зависимостями Розенау и Берглеса. В общем, многие данные и методики для вынужденноконвективного кипения внутри труб применимы к кинению при натекании струи на поверхность.