THE UNIVERSITY OF MICHIGAN

COLLEGE OF ENGINEERING Department of Chemical and Metallurgical Engineering

Final Report - Phase I

LITERATURE SURVEY ON LIQUID METAL BOILING

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ORA Project 04526

under contract with:

FLIGHT ACCESSORIES LABORATORY AERONAUTICAL SYSTEMS DIVISION AIR FORCE SYSTEMS COMMAND UNITED STATES AIR FORCE WRIGHT-PATTERSON AIR FORCE BASE, OHIO CONTRACT NO. AF 33(616)-8277-ITEM IIa

administered through:

OFFICE OF RESEARCH ADMINISTRATION ANN ARBOR

December 1961

JAPOJAM

FOREWORD

This report was prepared in the College of Engineering, The University of Michigan, on Air Force Contract AF 33(616)-8277 under Task 314507 of Project 3145. The work was administered under the direction of the Flight Accessories Laboratory, Aeronautical Systems Division, Wright Patterson Air Force Base, Ohio. Lt. Lloyd Hedgepeth and Mr. Kenneth Hopkins were project engineers for ASD. The survey began in June 1961 as the initial phase of a program which is to include an experimental investigation of liquid-metal-boiling phenomena and associated two-phase-flow problems. Professor R. E. Balzhiser of the Department of Chemical Engineering is the Project Director at The University of Michigan. Professors J. A. Clark and Herman Merte, Jr., have specific interests in the agravic portion of the program, and Professor E. E. Hucke has particular interest in the relation of interfacial effects to boiling processes. Messrs. C. Phillip Colver, Lowell R. Smith, and A. S. Teller are graduate students in the Department of Chemical and Metallurgical Engineering at The University of Michigan. Messrs. S. Kim and W. A. Niethammer have worked long and diligently in organizing the extensive bibliography and physical property charts. This report is the culmination of a joint effort of the above individuals.

Appreciation is expressed to the following individuals and groups for permission to reproduce figures originating in their publications: The Oil and Gas Journal, published by the Petroleum Publishing Company; Dr. John Vohr of Columbia University; American Institute of Chemical Engineers; Professor C. F. Bonilla; Consultants Bureau Enterprises Inc.; and the Advanced Technology Laboratories, a division of American Standard. The authors also wish to extend appreciation to the many investigators who have contributed information for this survey.

This report concludes the work on Phase I of Contract No. AF 33(616)-8277. Work on Phase II involving the experimental investigations of liquid-metal-boiling systems is currently in progress.

ABSTRACT

Recent interest in high-temperature, high-flux, heat-transfer processes has focused considerable attention on liquid metals as heat-transfer media. This survey was originated for the purpose of collecting and evaluating information pertaining to the current status of liquid-metal-boiling technology. The sparsity of information specifically about liquid-metal-boiling programs prompted the inclusion of additional material pertaining to boiling and two-phase-flow phenomena in general. Existing correlations for predicting heat-transfer coefficients in the nucleate- and film-boiling regimes have been summarized and analyzed in the report. Likewise, correlations which predict the critical heat flux (or burnout flux) have been presented and compared with the experimental data available.

The use of liquid metals as fluids in space-oriented Rankin cycles necessitates a thorough understanding of quality and gravity effects on boiling phenomena. Each of these variables is treated in separate sections, with pertinent investigations and conclusions summarized. Interfacial considerations of possible importance are cited and discussed. Particular attention is called to the solid-liquid interfacial energy and its importance in limiting heat transfer across the interface.

The importance of two-phase-flow considerations in understanding the heat-transfer phenomena prompted the inclusion of additional sections regarding flow regimes and the pressure drops in flowing two-phase media. Both of these sections describe correlations presently used for water-steam or water-air two-phase mixtures. Little work has been reported to date regarding two-phase-flow phenomena in liquid metallic systems.

Appendix B is a summary of physical properties for various liquid metals and water. Examination of these physical properties suggests in many instances that existing correlations for aqueous systems might be used with reasonable confidence in predicting liquid-metal behavior. Appendix D is a comprehensive bibliography of all aspects of boiling heat transfer, fluid flow, and corrosion and circulation problems associated with liquid-metal fluids.

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INTRODUCTION

The literature on boiling heat transfer contains relatively little information on liquid-metal systems. Early interest in connection with the mercury-turbine and binary-power cycles produced some data for mercury systems. Subsequent interest prompted by the need for high-temperature coolants for nuclear reactors led to the development of several programs in the last decade. However, few results have yet reached the unclassified literature. Summaries and analyses of the available reports are included in this review. Special attention should certainly be called to the recent translation from Russian by Consultants Bureau Inc. of Liquid Metal Heat Transfer Media, 617* edited by S. S. Kutateladze, which is devoted entirely to problems associated with utilizing liquid metals (up to 1958) and the recent paper of Gambill and Hoffman which summarizes the field of boiling-metal heat transfer up to mid-1961.

The sparsity of information available for boiling-liquid-metal systems makes it extremely difficult to engineer such systems. The experimental difficulties associated with a precise evaluation of the effects of many important variables on the heat-transfer process in liquid-metal media reduce the probability of obtaining directly the needed information. Correlations and studies for nonmetallic fluids are certain to fill the many voids in the liquid-metal picture. Therefore, summaries of the present status of boiling heat transfer in general have been included. The authors have attempted to summarize in reasonable detail the results of these investigations.

The phenomenon of surface boiling exhibits the three separate regimes descriptively shown in Fig. 1. These modes are nucleate boiling, transitional boiling, and film boiling. Nucleate boiling (region AB) is characterized by the generation of vapor bubbles at selective locations on the surface. These bubbles either collapse back to the surface (as when the bulk liquid is sufficiently subcooled) or detach themselves and are carried by inertial and buoyant forces into the bulk liquid. During nucleate boiling, the heat-flux density is not directly proportional to the driving force, as in normal convective heat transfer, but to some power of the driving force. The heat-transfer mechanism in this regime is not well understood and several mechanisms have been proposed. As the heat flux density is further increased, the population of nucleating sites increases until the growing bubbles tend to coalesce to form an unstable vapor blanket. This point is shown by point B and is referred to as the critical heat flux density.**

* * *

Manuscript released by authors on December 1, 1961, for publication as an ASD Technical Report.

^{*}Numbers in superscript after names refer to reference numbers listed in Appendix D.

^{**}This condition is frequently referred to as the lower critical, the first crisis, or the burnout point.

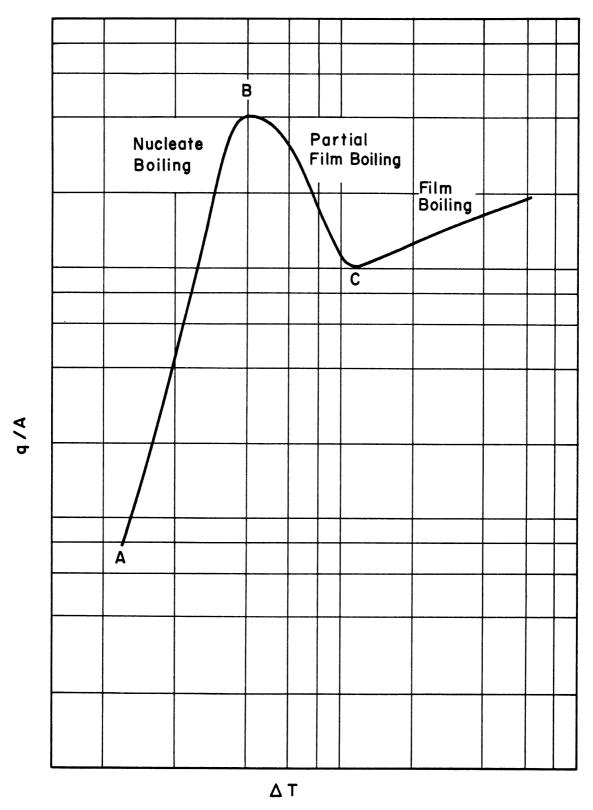


Fig. 1. Typical boiling curve.

The second regime (BC), partial-film boiling or transitional boiling, is characterized by the existence of an unstable vapor blanket that releases great patches of vapor at more or less regular frequencies. It is seen that the heat-transfer rate diminishes as a result of the insulating action of the vapor. As the temperature of the surface is increased, the heat flux is observed to pass through a minimum. At this temperature (C), a stable vapor film covers the entire surface and film boiling occurs. Heat transfer is accomplished principally by conduction and convection through the vapor film with radiative contributions becoming more significant as the surface temperature increases.

When ebullition is governed by the heat-flux density (electrical heating), as opposed to control by the temperature-driving force (condensing media), it is obvious that any increase of heat-flux density above the critical heat flux causes the surface temperature to rise rapidly in an effort to compensate for the decreasing coefficient. If this rise in temperature causes the surface to exceed the melting point of the surface, the phenomenon of "burnout" occurs. For this reason it is desirable to operate a boiling system as close to as possible, but without fear of exceeding, the critical heat-flux density.

MECHANISM OF NUCLEATE BOILING

A detailed knowledge of the heat-transfer mechanism for nucleate boiling is very important since it presumably would permit the calculation of heat-flux densities for various liquids at different pressures, forced convective velocities, superheats, surface conditions, agravic conditions, etc. A considerable amount of research has been performed in this area and several mechanisms have been suggested. In essence, however, each mechanism yet proposed has in some way been an alteration or extension of one or more of three intrinsic modes of transferring heat in nucleate boiling. These modes are: (1) microconvection heat transfer; (2) latent heat transport; and (3) vapor-liquid exchange.

- (1) <u>Microconvection heat transfer</u>.—The rapid growth of the vapor bubble at a nucleation site imposes a quantum of kinetic energy to the surrounding liquid, thus accelerating the liquid to a velocity in excess of its natural convection velocity. This pulsating action at each site creates currents in the normally stagnant or laminar sublayer near the boiling surface.
- (2) <u>Latent heat transport</u>.—This mechanism is essentially the transfer of latent heat from the boiling surface to the liquid to form and grow a vapor bubble. Heat may be transferred through the vapor bubble by mass transfer; i.e., some of the heat used to vaporize the liquid near the base of the vapor bubble is carried as vapor to the bubble cap where it is transferred to the liquid bulk by condensation of the vapor.

(3) <u>Vapor-liquid exchange</u>.—This mechanism allows the growing, collapsing, and departing vapor bubbles to act as heat pumps, first by pushing superheated liquid into the liquid bulk and then by allowing bulk liquid to replace the void left by the collapsed or departed vapor bubble. As the vapor bubble grows, it displaces the superheated liquid near the surface by pushing it into the liquid bulk. When the bubble either collapses back to the surface or departs from the surface, the liquid fills the void left by the vapor bubble thus allowing large amounts of heat to be transferred at each nucleation site. The cycle is repeated.

With the aid of photographic results, Jakob⁵²¹ and later Rohsenow and Clark⁹³⁵ and Gunther and Kreith⁴²⁰ concluded that only microconvection could account for the majority of the heat exchange during nucleate boiling. Edwards made calculations for subcooled liquids and found that mass transfer through a growing and collapsing vapor bubble could account for a major portion of the heat transfer to the liquid. Forster and Greif³⁴⁴ have made speculative calculations and have concluded that "the amount of heat transferred by the liquid-vapor exchange taking place every time a bubble grows and then collapses on, or detaches from, the heating surface is by itself sufficient to account for the heat flux in nucleate boiling."

Treshchov, 1084 Zuber, 1180 and Chang 182 have recently proposed mechanisms including more than one intrinsic mode. Treshchov 1084 and Chang 182 have proposed a nucleate boiling-heat-transfer mechanism including all three modes. They state that at the initiation of nucleate boiling the greater part of the heat is transmitted by microconvection, but with an increase of heat-flux density the share of heat transmitted by microconvection is decreased. In turn, the heat transferred by the bubbles in both latent-heat and vapor-liquid exchange is increased to the point at which, when nucleate boiling is fully developed, all the heat is essentially transferred by the bubble. Zuber's analysis is similar; however, he neglects vapor-liquid exchange.

A more recent mechanism advanced by Moore and Mesler 791 postulates microlayer vaporization. These experimenters believe that as the vapor bubble grows on the surface it traps a very thin layer of liquid beneath it which rapidly evaporates, transferring great quantities of heat. With the aid of a thermocouple which measures transient temperatures on the surface, they were able to account for 70 to 90% of the heat transferred.

NUCLEATE-BOILING CORRELATIONS

Many investigators have suggested semi-empirical expressions relating the heat-flux density to various properties of nucleate-boiling systems. For the most part analysis has been made using dimensionless parameters and fitting the various empirical constants with experimental data.

Rohsenow proposed the following expression for pool boiling:*

$$\frac{\operatorname{Cp}_{\ell} \Delta T_{W}}{\lambda} = \operatorname{C} \left[\frac{\operatorname{q/A}}{\mu_{\ell} \lambda} \sqrt{\frac{\operatorname{g_{c}} \sigma}{\operatorname{g}(\rho_{\ell} - \rho_{V})}} \right]^{0.33} \operatorname{Pr}_{\ell}^{1.7}$$
 (1)

where C is a constant for a particular heating surface-fluid combination. He assumed that energy transfer occurs primarily from surface to liquid and extended the Nusselt analogy. He used bubble diameter as the characteristic system dimension for the Nusselt and Reynolds numbers and succeeded in correlating the data of Addoms, 7 Cichelli-Bonilla, 199 and Cryder-Finalborgo. 229 The above equation expressed explicitly in q/A follows:

$$q/A = C' \mu_{\ell} \lambda \left[\frac{Cp_{\ell} \Delta T_{W}}{\lambda} \right]^{3} Pr_{\ell}^{-5.1} \left[\frac{g(\rho_{\ell} - \rho_{V})}{g_{C} \sigma} \right]^{1/2}$$
(2)

Application of this equation is somewhat restricted because of the need for experimental data to evaluate the constant.

Levy 658 derived a general equation to represent nucleate boiling of saturated liquids by postulating that as the generated bubbles attain their maximum diameter, they carry all heat transferred at the heat-transfer surface. His expression,

$$q/A = \frac{1}{B_{L}} \frac{k_{\ell} C p_{\ell} \rho_{\ell}^{2}}{\sigma T_{s} (\rho_{\ell} - \rho_{v})} \Delta T_{w}^{3}$$
(3)

is (except for secondary effects) independent of pressure and the heat surface-liquid combination. The constant B_L is empirically determined and found to be well represented by plotting it against $\lambda\rho_V$. This relationship correlates reasonably well the ethanol and normal heptane data of Cichelli-Bonilla and the water data of Addoms. 7

Forster and Grief³⁴⁴ in their analysis decided which dimensionless parameters were significant and then correlated them with experimental data. The results produced two expressions. The first utilizes a specific coefficient for each liquid; the second expression employs the same constant for all liquid-surface combinations. This permits extension to systems previously unexplored, but some sacrifice in accuracy is inherent. This latter expression has been shown to correlate well with the boiling data for mercury in the range 1-3 atmospheres. Por liquid-metal-boiling systems where few data are available, the second form seems to possess greater utility. It is

^{*}Symbols and their definitions are given in Appendix A.

$$q/A = 4.3 \times 10^{-5} \frac{\alpha Cp_{\ell} \rho_{\ell} T_{S}}{\sqrt{\sigma} (\lambda \rho_{V})^{3/2}} \left[Cp_{\ell} T_{S} \sqrt{\alpha} \right]^{1/4} \left[\frac{\rho_{\ell}}{\mu_{\ell}} \right]^{5/8} Pr_{\ell}^{1/3} \Delta p^{2}$$
(4)

This equation also correlates Lyon's sodium results. 696

Chang and Snyder¹⁸⁵ applied dimensional analysis to the fundamental equations for motion and energy to produce parameters which characterize the nucleate-boiling phenomena. The concept of a thermal-eddy diffusivity was incorporated in their analysis. The following equation, which is good for vigorous boiling, resulted from this study:

$$q/A = 4 \times 10^{-4} \frac{k_{\ell} \Delta p^{1.4}}{(\rho_{v} \lambda)^{0.8} \sigma} \left[Cp_{\ell} T_{s} (\rho_{\ell} - \rho_{v}) \right]^{0.4} \Delta T_{w}$$
 (5)

The authors state that this expression is directly applicable for liquid metals. Comparison with the data of Bonilla supports this supposition.

From experiments with nonmetallic liquids, Kutateladze 617 has derived the following:

$$q/A = 0.44 \operatorname{Pr}_{\ell}^{0.35} \left[\frac{q/A \ p \ 10^{-4}}{\lambda \rho (\rho_{\ell} - \rho_{V})} \right]^{1.7} \left[\frac{k_{\ell}^{2} (\rho_{\ell} - \rho_{V})}{\sigma} \right]^{0.5} \Delta T_{W}$$
 (6)

or explicitly in q/A

$$q/A = 0.76 \text{ Pr}_{\ell}^{1.17} \left[\frac{p \cdot 10^{-4}}{\lambda \rho_{v} (\rho_{\ell} - \rho_{v})} \right]^{2.33} \left[\frac{k_{\ell} (\rho_{\ell} - \rho_{v})}{\sigma} \right]^{1.67} \Delta T_{w}^{3.33}$$
 (7)

Borishanskii and Menchenko, ⁶¹⁷ after experimenting with different liquids, concluded that for ordinary liquids the power on the Prandtl number of Kutateladze's first equation should be changed to 0.7 and the value of the coefficient changed to 0.55. The predictions of this correlation are compared with data for magnesium, mercury amalgams, and sodium in a later section of this report.

Mumm proposed an interesting correlation that considered variations in vapor fraction. Four dimensionless parameters were selected to characterize the nucleate-boiling phenomena. The correlation was based on data obtained for the water-steam system and is supposedly applicable for qualities up to 40%.

$$q/A = \left[4.3 + 5 \times 10^{-4} \left(\frac{V_{v}}{V_{\ell}}\right)^{1.64} \chi\right] \left[\frac{q/A}{G\lambda}\right]^{0.46} \left[\frac{Gd_{e}}{\mu_{e}}\right]^{0.8} \left[\frac{k_{\ell}\Delta T_{w}}{d_{e}}\right]$$
(8)

In a recent publication ${\rm Chang}^{183}$ employed theory developed from the Maxwell-Boltzman distribution law to derive the following empirical expression:

$$q/A = c \frac{\rho_{\ell} Cp_{\ell} \Delta T_{w} N_{A} (KT_{s})^{3/2}}{H\sigma^{1/2}} \exp \left[-n \frac{1b \pi}{3} \left(\frac{\sigma}{\Delta P}\right)^{2} \frac{\sigma}{KT_{s}} \left(\frac{\rho_{\ell} Cp_{\ell} \Delta T_{w}}{\rho_{v} \lambda}\right)^{-m}\right] (9)$$

where m = 1 and 2 for organic and inorganic liquids respectively, and c and n represent dimensionless numbers whose values depend on the liquid and surface conditions. The equation is valid for liquids, including liquid metals, under the following conditions: saturated pool boiling from either rough or smooth surfaces; saturated or subcooled forced-convection boiling from rough surfaces; and early stages of forced-convection boiling (saturated or subcooled) from smooth surfaces. A comparison with Lyon's data showed good agreement.

CRITICAL HEAT-FLUX CORRELATIONS

At the upper limit of the nucleate-boiling regime, sufficient nucleation sites have become active to cover the surface. As the number of vapor columns emanating from the surface increases, the cross-sectional area remaining for liquid flow to the surface decreases. This necessitates an increased velocity if the liquid supply to the surface is to be replenished.

Early investigators of this type of phenomenon observed instabilities in the system when a certain relative velocity was achieved between the two phases in countercurrent flow. More recent theoretical attempts to relate this observed instability in two-phase flow to the critical heat-flux limitations have aroused much attention and have produced some encouraging results. Experimental verification of their predictions is difficult, particularly for liquid-metal media. However, some burnout data for water and organics are available and have been used to check (in part) some of the theoretical treatments. At the same time, it has led to empirical and semi-empirical correlations for the critical flux. Several of the more promising results are summarized along with a brief discussion of the effects of pressure, velocity, and subcooling on the location of the critical point. The effects of quality, interface conditions, and agravic considerations are treated in greater detail later in the report.

Numerous analytical expressions have been derived to predict the critical heat flux. Even though most expressions are limited to water, there are several with presumably general application. For saturated pool boiling, Rohsenow and Griffith 937 have proposed the following:

$$\left[q/A\right]_{c} = 143g^{1/4} \rho \lambda \left[\frac{\rho_{\ell} - \rho_{v}}{\rho_{v}}\right]^{0.6}$$
(10)

This correlation was compared with the data of Cichelli and Bonilla 199 and produced approximate deviations of about $\pm 11\%$.

Considering the critical heat flux as a phenomenon governed by hydrodynamic limitations, Kutateladze 624 employed dimensional analysis to derive the relationship:

$$[q/A]_{c} = K\lambda \rho_{v} \left[g^{2}\sigma \left(\frac{\rho_{\ell}-\rho_{v}}{\rho_{v}^{2}}\right)\right]^{1/4}$$
(11)

When compared with water and some organic liquids for saturated pool boiling, the best value of K was found to be in the range 0.14 to 0.18. For subcooled pool boiling, K is no longer constant but a function of the groups,

$$\frac{\lambda}{\text{CpAT}_{\text{sub}}}$$
 and $\frac{\rho_{\text{V}}}{\rho_{\ell}}$

These give rise to a new expression:

$$[q/A]_{c,sub} = [q/A]_{c} \left\{ 1 + (1-n) \left[\frac{\rho_{\ell}}{\rho_{V}} \right] \frac{Cp_{\ell}\Delta T_{sub}}{\lambda} \right\}$$
 (12)

When correlated with data for water, alcohol, and isooctane, Eq. (13) resulted.

$$[q/A]_{c,sub} = [q/A]_{c} \left\{ 1 + 0.065 \left[\frac{\rho_{\ell}}{\rho_{v}} \right]^{0.8} \frac{Cp_{\ell}\Delta T_{sub}}{\lambda} \right\}$$
 (13)

Zuber and Tribus 1175 considered the critical heat flux as a hydrodynamic limitation arising from Taylor-Helmholtz instabilities at the vapor liquid interface. Their expression,

$$\left[q/A\right]_{c} = \frac{\pi}{24} \lambda \rho_{v} \left[\frac{\sigma g(\rho_{\ell} - \rho_{v})g_{c}}{\rho_{v}^{2}}\right]^{1/4} \left[\frac{\rho_{\ell}}{\rho_{\ell} + \rho_{v}}\right]^{1/2}$$
(14)

is seen to differ slightly from Kutateladze's in the value of the constant and includes an additional term which is near unity. For subcooled liquids, Zuber and Tribus 1175 extend Eq. (14) to

$$[q/A]_{c,sub} = \frac{\lambda \pi}{24} \frac{\lambda_o}{\tau} + \frac{\pi}{24} \rho Cp_{\ell} (T_s - T_{\ell}) \frac{\lambda_o}{\tau} + \sqrt{2\pi} \frac{k}{\sqrt{\alpha \tau}} [T_s - T_{\ell}]$$
 (15)

where

$$= 2\pi \left[\frac{\sigma}{g(\rho_{\ell}-\rho_{V})}\right]^{1/2} \left[\frac{\rho_{V}^{2}}{\sigma g(\rho_{\ell}-\rho_{V})}\right]^{1/4} \left[\frac{\rho_{\ell}}{\rho_{\ell}+\rho_{V}}\right]^{1/2}$$
(16)

and

$$\frac{\lambda_{O}}{\tau} = \left[\frac{\sigma g(\rho_{\ell} - \rho_{V})}{\rho_{V}^{2}}\right]^{1/4} \left[\frac{\rho_{\ell}}{\rho_{\ell} + \rho_{V}}\right]^{1/2}$$
(17)

This equation was compared to the critical heat flux data for water, 420 ammonia, 1175 and carbon tetrachloride 295 with fair agreement.

Griffith 409 has proposed an empirical equation applicable for different levels of subcooling, force convective velocities, and pressure. It is

$$[q/A]_{c,sub} = f\left[\frac{p}{p_c}\right](h_v-h_b)\rho_v\left[\left(\frac{\rho_{\ell}-\rho_v}{\mu_{\ell}}\right)g\left(\frac{k_{\ell}}{\rho_{\ell}Cp_{\ell}}\right)^2\right]^{1/3}$$
 F (18)

where

$$F = 1 + Re_{I} \cdot 10^{-6} + 0.014J + 5 \times 10^{-4} \left[J Re \right]^{1/2}$$
 (19)

and

$$J = \frac{\rho_{\ell} C p_{\ell} (T_S - T_b)}{\rho_{\nu} \lambda}$$
 (20)

Over 300 data points from various liquids (including water, benzene, n-heptane, and ethane) have been correlated, with 94% of the points having less than a $\pm 33\%$ deviation.

At low pressures for saturated boiling, it has been experimentally shown that as the pressure is raised the critical heat flux markedly increases. 101,295,1081 Kazakova 622 has experimentally determined the critical heat flux for water boiling from flat disks. Her data indicate that the critical heat flux increases with pressures up to 30 to 40% of the critical pressure, then slowly decreases to zero as the pressure approaches the critical value. This behavior is in qualitative agreement with all the equations and with most investigations reviewed.

For saturated forced convection, Aladyev et al. 16 present data for water with flow rate as a separate parameter showing similar behavior to that described above. At higher flow rates the critical heat flux appeared less sensitive to pressure changes.

Subcooling has been shown to have a greater effect at low pressures, as demonstrated by the following relationship:

$$\left\{ \frac{\left[\frac{q}{A} \right]_{c}, \text{sub}}{\left[\frac{q}{A} \right]_{c}} \right\}_{p_{1}, T} > \left\{ \frac{\left[\frac{q}{A} \right]_{c}, \text{sub}}{\left[\frac{q}{A} \right]_{c}} \right\}_{p_{2}, T}$$
(21)

where $p_1 < p_2$.

Verification of this behavior is clearly shown by the data of Kutateladze and Shneiderman. 631

For water and organic liquids, velocity increases have been shown to increase markedly the critical heat-flux density. The growing vapor bubbles on the surface are swept away at a smaller diameter by the flowing stream, thus permitting more nucleating sites to become active at the surface before coalescence results. Consequently, a greater ΔT is required to activate these sites. Since the heat transfer coefficient would not be expected to decrease, the critical heat flux must increase.

Aladyev et al. 16 found that for water at constant subcooling a change from 1 meter/sec to 2 meters/sec affected the critical heat flux insignificantly at 20 atm, but at 180 atm increased it approximately 50%. On the other hand, Torikai 1080 found that for water at 1 atm and constant subcooling the critical heat-flux density was increased as much as 50% for an increase in velocity from 1 meter/sec to 2 meters/sec. With an increase from 1.1 ft/sec to 5 ft/sec, at constant subcooling and 16 psia, Ellion found that the critical heat-flux density was increased 100%.

Many investigators have shown that subcooling increases the critical heat-flux density above that in saturated boiling. In their experiments, Kutateladze and Shneiderman have clearly shown that for pool boiling with ethanol, isooctane, and water, at constant pressure, the critical heat-flux density increases for a decrease in the bulk temperature.

Ellion 295 found, for water flowing at 1 ft/sec, that changing from 50°F to 100°F subcooling increased the critical heat-flux density approximately 50%. This behavior was confirmed by Aladyev et al.

In their derivation of Eq. (15) for subcocled boiling, Zuber et al. 1177 assumed, for subcooled pool boiling, that the same hydrodynamic behavior is exhibited at the critical as for saturated boiling, but that an additional quantity of energy is transferred to the subcooled liquid.

FILM-BOILING CORRELATIONS

Unlike the forced-convection and nucleate-boiling regimes, the film-boiling region has been the subject of relatively few analytical studies and very few experimental investigations. However, today's technology includes areas where it is essential to transfer large quantities of heat across minimal surface areas, thus increasing the probability of encountering this phenomenon. The process is characterized physically by a layer of vapor that separates the liquid from the heat surface. The energy transfer through the vapor layer occurs by conduction, convection, and radiation processes. At reasonable temperature levels the attainable fluxes are substantially suppressed.

Several excellent literature surveys on film boiling have been compiled. Drew and Mueller²⁶⁸ as part of a general review on boiling surveyed film-boiling literature up to 1937. Westwater¹¹⁴⁵ reviewed the literature on film boiling up to 1955 and summarized the work in terms of description of photographic studies; theoretical treatments; and experimental results where the effects of type of liquid, type of solid and its surface texture, geometric arrangement, pressure, surface tension, agitation, and impurities were separately considered. McFadden and Grosh⁷⁵⁰,⁷⁵¹ extended the coverage to 1959 in their review. This summary includes findings discussed in these reviews and attempts to bring the subject up to date by including several of the more important recent contributions.

Although the first observation of the phenomenon of film boiling was made as early as 1746, an analytical development did not appear in the literature until 1950 when Bromley, prompted by the earlier work of Colburn, presented a theory of stable laminar film boiling. Bromley based his derivation on Nusselt's derivation for heat transfer during laminar-film condensation. It treated specifically free-convection film boiling on the outside of an isothermal horizontal tube and incorporated the following assumptions: a vapor blanket exists between the liquid and the tube wall, heat is transferred through the film by conduction and radiation, the vapor rises due to buoyant forces, the liquid vapor interface is smooth, viscous drag retards the rise of the vapor, the enthalpy of vaporization is the major energy supplied to the film, the kinetic energy of the film is negligible, the liquid is at rest and saturated, and properties may be evaluated at an average temperature. The resulting theoretical equation is modified with an experimentally determined constant to fit the physical situation. Bromley's results can be expressed as

$$h_{co} = 0.62 \left[\frac{k_{\ell}^2 \rho_{v} (\rho_{\ell} - \rho_{v}) g \lambda' C p_{\ell}}{d \Delta T_{w} Pr} \right]^{1/4}$$
(22)

where 0.62 is empirical, a compromise between 0.724 and 0.512. The former value corresponds to the situation in which the liquid is moving with the same velocity as the vapor (hence zero sheer stress at the vapor-liquid interface), and the second value arises where the liquid is considered to be at rest, thus producing a large sheer stress at the interface. Bromley corrected his heat-transfer coefficient for radiation by assuming infinite parallel-plane-plate radiation. The radiation coefficient was expressed as follows:

$$h_{r} = \left[\frac{\sigma'}{\frac{1}{\epsilon_{w}} + \frac{1}{\epsilon_{\ell}} - 1} \right] \left[\frac{T_{w}^{4} - T_{\ell}^{4}}{\Delta T_{w}} \right]$$
 (23)

The radiation coefficient was combined with the convection coefficient in the following manner to obtain a total heat-transfer coefficient.

$$h = h_{co} + h_{r} \left[\frac{3}{4} + \frac{1}{4} \frac{h_{r}}{h_{co}} \left(\frac{1}{2.62 + h_{r}/h_{co}} \right) \right]$$
 (24)

For low wall temperatures the total coefficient was expressed as follows:

$$h = h_{co} + \frac{3}{4} h_{r}$$
 (25)

For vertical tubes Bromley used the same expression for the convection coefficient, but substituted the height, L, of the tube for D as the characteristic dimension in his expression. For such small values of L that the vapor film was laminar, this correlated the data satisfactorily. These correlations were substantiated with data taken on the following liquids: water, nitrogen, carbon tetrachloride, absolute ethyl alcohol, benzene, diphenol oxide, and normal pentane. Bromley's experimental program showed that the value of the coefficient was independent of the physical characteristics of the liquid although his experiments were limited to a rather narrow viscosity range. The physical or chemical character of the tube or tube surface appeared to have little or no effect as long as it was fairly round and smooth. The effect of diameter was as predicted in the equation. Total coefficients ranging from 18 to 80 Btu/hr (sq ft)(°F) were measured, and the following conclusions were obtained:

- (1) The liquid vapor interface is substantially smooth along the bottom two thirds of the tube except at high heat fluxes. At the top it is always uneven due to bubble formation and departure.
- (2) Heat-transfer coefficients are independent of the tube material except for the radiation contribution.
- (3) The effect of variables such as pressure may be calculated from their effect on the physical properties of the liquid and its vapor.
- (4) A decrease in surface tension does not affect the calculated coefficients, but the minimum critical heat flux and the corresponding temperature-heat difference are both decreased.
- (5) Film boiling persists for subcooled liquids with higher coefficients resulting.
- (6) Mercury was shown to exhibit film boiling at very low ΔT's. Bromley observed that the potassium additions did not substantially affect this behavior.
 - (7) Heat-transfer coefficient is increased for forced convection.
- S. S. Kutateladze⁶²⁶ summarized Russian efforts in the area of film-boiling heat transfer up through 1952. He considers the laminar flow of a vapor layer along a vertical plate, assuming that all vapor moves along the heated surface. He, too, considers the two extremes discussed by Bromley (above), and observes that the value of the coefficient differs by a factor of 1.59 for the two extremes. For free convection he presents the following correlation for the average value of the heat-transfer coefficient:

$$h = \beta^{3/4} \phi^{1/4} \left[\frac{k_{\ell}^2 C p_{\ell} \lambda \rho (\rho_{\ell} - \rho_{V})}{Pr \Delta T L} \right]^{1/4}$$
(26)

where β = 0.436 - 0.690 and ϕ = 1 + (Cp/2\lambda) ΔT . For forced flow of a liquid such that

$$\frac{2\mu_{\boldsymbol{\ell}}u_{\boldsymbol{\ell}}}{(\rho_{\boldsymbol{\ell}}-\rho_{v})} \gg 1 \tag{27}$$

the differential coefficient of heat transfer during film boiling is directly proportional to the liquid velocity. For situations in which radiative transfer must also be considered, the following expression results for the average heat-transfer coefficient:

$$h = \frac{4}{3} \beta' \left[\frac{k_{\ell}^{2} Cp_{\ell} \phi \lambda \rho_{v} (\rho_{\ell} - \rho_{v})}{Pr_{\ell} \Delta T_{w} L(1 + \psi)} \right]^{1/4}$$
(28)

where $\psi=h_r/h$ and $\beta'=0.500$ - 0.705. For high liquid velocities with a constant ΔT_w , the following expression correlates the average heat-transfer coefficient.

$$h = \left[\frac{k_{\ell} \phi \lambda \rho_{v} u_{\ell}}{\Delta T_{w} L(1+\psi)}\right]^{1/2}$$
 (29)

As is evident, the heat-transfer coefficient for these conditions is proportional to the square root of the liquid velocity.

Kutateladze's summary mentions the early experimental work of Styrikovich and Semonovker, 1045 who studied the transfer of heat to boiling mercury. At about the same time, Kutateladze and Zysina617 studied heat transfer to mercury boiling under conditions of free convection. In 1947 Lukomskii 688 measured the heat-transfer coefficient of carbon dioxide during film boiling in vertical tubes. Experimental data for the film-boiling of water at atmospheric pressure on a 3-mm vertical heater with a flux of 500,000 Kcal/meter²/hr produced a vapor-film Reynold's number of 62, thus confirming the assumption of laminar flow within the film. Pressures up to 12 atmospheres were also studied with the water system. Equations (26) and (28) were shown to correlate the data very well. Equation (28) also does a satisfactory job of correlating Bromley's data. Subcooling is shown to increase the value of the average transfer coefficient. However, the effect of subcooling decreases with an increase in the absolute pressure of the system because of a decreasing density ratio.

In 1954 Ellion 295 analyzed an isothermal vertical plate with laminar stable film boiling. He made essentially the same assumptions as Bromley and arrived at a result approximating Bromley's. This correlation was substantiated with water data for a velocity range of 1.1-5 ft/sec, subcooling from 50 to 100°F and

pressures from 16 to 60 psia. He reports film boiling to be independent of water pressure, velocity, and subcooling over these ranges.

Browley 156 later analyzed a case of stable laminar film boiling for an isothermal horizontal cylinder for uniform, vertical, upward flow. For low water velocities his results were the same as for free convection. For high water velocities his results can be expressed as follows:

$$Nu = 2.7 \left[\frac{u\rho \lambda d}{k \Delta T} \right]^{1/2} \quad \text{or}$$
 (30)

$$h = 2.7 \left[\frac{\text{kup}\lambda}{\text{d} \Delta T} \right]^{1/2}$$
 (31)

This study included four different liquids with velocities up to 14 ft/sec.

In 1958 Chang 186 presented his wave theory of heat transfer and film boiling from both horizontal and vertical isothermal surfaces. He considered heat transfer in both saturated and subcooled systems. He utilizes the concept of an equivalent thermal diffusivity to produce a generalized model. He generates a general formula for both convection and boiling. His results can be expressed as follows:

$$Nu = C \left[Pr_{\ell} Gr \right]^{1/3}$$
 (32)

$$= \left[\frac{g(\rho_{\ell} - \rho_{V})L^{3}}{8\pi^{2} \mu_{\ell}g_{c}\gamma_{c}}\right]^{1/3}$$
(33)

where γ_c is defined as follows:

$$\gamma_{\rm c} = \frac{\Delta T_{\rm sub}}{2(\lambda \rho_{\rm w} + \Delta T_{\rm w} C \rho_{\rm s} \rho_{\rm s})}$$
 (34)

For vertical plates the value of the exponent for the product (\Pr_{ℓ} Gr) is 1/4. Chang concludes that the heat-transfer coefficient for film boiling from the horizontal surface is in general higher than from a vertical plate. For boiling from tubes the reverse is observed to be true. He suggests that the effect of different variables be calculated from their effect on the physical properties of the liquid and vapor. Increases in pressure will increase the heat-transfer coefficient, but not as significantly as might be anticipated because the boiling point of the liquid will also increase with pressure. A higher wall temperature is then required to maintain film boiling which increases the radiative contribution.

McFadden and $6 \cos h^{750}$, 751 performed an analytical study of stable, free convection, laminar film boiling in which they consider transfer by conductive and convective processes only. The boundary-layer equations were solved using trans-

formation techniques for the following conditions: (1) compressible flow with variable specific heat; (2) variable specific heat and density variations considered only in the evaluation of the buoyant force; and (3) the case of constant properties. Numerical solutions were obtained for the following conditions: (1) water at 2800 and 3100 psia with wall-to-liquid temperature differences of 250, 500, and 1000°F; (2) for fluids with Prandtl numbers of 2/3, 1 and 2; and (3) for mercury and methanol film boiling at 1 atmosphere considering constant properties. An approximate analysis for nonisothermal wall condition, including radiation effects, was also performed. Radiation was shown to be the controlling factor in film boiling for high-emissivity walls at high temperatures.

The investigators concluded that for water at 2800 and 3100 psia, radiation is of more significance than the consideration of variable properties. They suggest, however, that as the critical pressure is approached, property variations will play a more important part in film-boiling heat transfer.

A comparison of Lyon's experimental data 695,696 for the film boiling of mercury with their theoretical results yielded satisfactory agreement. McFadden and Grosh pointed out that had Lyon chosen to measure his surface temperatures along the bottom 2/3 of the tube, better agreement would have been achieved. Their values for film boiling of methanol on the outside of a horizontal tube also yielded values slightly above the experimental data of Westwater and Santangelo. Again they postulate that the surface temperatures measured were not representative.

In 1960, Hsu and Westwater 1147 proposed an approximate theory for film boiling on vertical surfaces. The equation was developed for saturated liquids in the absence of forced flow and postulated the following conditions: (1) that vapor flow near the low end of the heating surface is viscous and Bromley's equation is applicable; (2) that turbulence develops with a local Reynolds number of about 100; and (3) that in the turbulent region of the heating surface thermal resistance is due entirely to the laminar sublayer. The results produced the following equation for the Nusselt number averaged over the upper and lower portions of the heating surface:

Nu =
$$\frac{2\lambda' \mu Re^* k_{\ell}}{3L \Delta T_W} + \frac{B + 1/3}{A} \left\{ \left[\frac{2}{3} \frac{A}{B + 1/3} (L - L_0) + \left(\frac{1}{y^*} \right)^2 \right]^{3/2} - \left(\frac{1}{y^*} \right)^3 \right\}$$
 (35)

where Re* represents the vapor-film Reynolds number; y^* , the critical vapor film thickness; and A and B are functions of the system properties and Re*. Experimental data were obtained for five liquids, methanol, benzene, carbon tetrachloride, nitrogen and argon. Tube lengths were varied from 2.0 to 6.3 inches. Hsu and Westwater's prediction appears to be much improved over Bromley's and Chang's, with the results being particularly good for nitrogen and argon, both of which have high ΔTs (560-780°F). For the organics with lower ΔTs (150-310°F) the correlation is not as reliable over the ΔT range investigated. The data that show h decreasing as ΔT increases are characteristic of the transitional region. Data at higher ΔTs , where film boiling is certain, might produce better agreement.

As the tube lengths increase from zero, the predicted heat-transfer coefficient passes through a minimum and then increases steadily. The minimum local heattransfer coefficient occurs at the point where turbulence first develops. This value corresponds to 1.9 in. of water and about 1/2 in. for nitrogen for a ΔT of 700°F. The minimum heat-transfer coefficient averaged over the length occurs at greater lengths, 4.8 in. for water and 2.4 in. for nitrogen for the same AT. Bromley's equation predicts a decreasing value of h as L is increased. Increases in AT cause increases in the average heat-transfer coefficient for water, but the reverse is predicted for nitrogen. The Hsu-Westwater correlation produced an average deviation of about 32% for the predicted Nusselt number as compared to experimental values. This is shown to be an improvement over the predictions of Bromley or Chang. Recently Berenson⁸⁹ developed an analytical expression for the heat-transfer coefficient near the minimum in film pool boiling from a horizontal surface. He utilizes Taylor-Helmholtz hydrodynamic instability to formulate a model from which he derives the following expression for the heat-transfer coefficient:

$$h = 0.425 \left[\frac{k_{\ell} \lambda \rho_{v} (\rho_{\ell} - \rho_{v})g}{\mu \Delta T_{w} \sqrt{\frac{g_{c} \sigma}{g(\rho_{\ell} - \rho_{v})}}} \right]^{1/4}$$
(36)

A comparison of his expression with Bromley's shows that the diameter has been replaced with

$$\sqrt{\frac{g_{c}\sigma}{g(\rho_{\ell}-\rho_{V})}}$$

as the geometrical scale factor for horizontal surfaces.

The applicability of Berenson's expression at fluxes substantially above the minimum flux is questionable. Radiation effects, which the author suggests become appreciable at temperature differences above $1000^{\circ}F$, and velocity effects would both tend to produce higher values of the coefficient. Berenson's article contains an expression for the minimum flux, which occurs at the onset of stable film boiling, and also the expression for the ΔT at which film boiling can occur. Experimental results obtained for normal pentane and carbon tetrachloride agree within 10% of his theoretical predictions.

Cess and Sparrow¹⁸⁰ investigated film boiling in forced-convection boundary-layer flow. Their results can be expressed in the following manner:

$$\frac{\text{Nu}}{\sqrt{\text{Re}}} \left(\frac{\mu_{\text{V}}}{\mu_{\ell}} \right) \left[1 + \sqrt{\pi \frac{\text{Nu}}{\sqrt{\text{Re}}} \frac{\mu_{\text{V}}}{\mu_{\ell}}} \right]^{1/2} = 0.5 \left[\frac{(\rho \mu)_{\ell} \text{Cp } \Delta T_{\text{W}}}{\rho_{\text{V}} \mu_{\text{V}} \lambda \text{ Pr}_{\ell}} \right]^{-1/2}$$
(37)

A simplified, but less accurate, expression can be obtained by ignoring the square root bracket on the left side of the equation. It can be seen from this

equation that the heat-transfer coefficient is inversely proportional to the square root of the temperature difference. Therefore, in the film-boiling regime at low fluxes, q is proportional to ΔT to the 1/2 power, which is a smaller ΔT dependence than exhibited by other convective-transfer phenomena. Cess and Sparrow extended their analysis to include subcooled liquids. Subcooling was shown to produce an appreciable increase in the heat-transfer coefficient. The effect is expected to be most pronounced for low Prandtl liquids, such as metals.

Lin et al. 664 performed experimental studies with pure mercury at 1 atmosphere. The system was observed to enter the film-boiling regime for very low temperature differences. As the flux increased, the coefficient was observed to decrease. An expression $h = 4850q^{-0.26}$ correlated their data. The experimental values correlated by this equation fell about 50% above the theoretical line corresponding to Bromley's prediction. These investigators observed that increases in pressure changed the boiling type from film to nucleate with corresponding increases in both heat-flux and heat-transfer coefficient. The experimental work of Lyon with mercury systems also confirmed the tendency of mercury under nonwetting conditions to exhibit film boiling at relatively low temperature differences. The addition of magnesium and titanium in very small quantities was observed to promote wetting. Coefficients and fluxes characteristic of the nucleate regime were then comparable for temperature differences. Preliminary calculations for sodium at temperatures where conduction would be expected to predominate yielded a value for the coefficient of 43 Btu/hr ft2. This value is of the same magnitude as that observed by Bromley with other fluids. It appears that unless significant radiative contributions occur at higher ATs, the flux in the film-boiling regime will remain below the critical flux for reasonable values of ΔT .

Investigations to date have shed some light on the effect of certain variables. Liquids studied thus far have not indicated a radical difference for heat-transfer coefficients in the stable film-boiling regime. The main difference between liquids seems to be due to differences in wettability on particular surfaces. Film boiling will occur at smaller temperature differences for nonwetting fluids. Lyon's results with mercury demonstrated this phenomenon. Similarly, the surface from which heat is transferred has relatively little effect on the transfer coefficient. However, it should be remembered that extreme roughness might change the character of flow, producing changes in the coefficient. Likewise, at high temperatures the emissivity of the surface becomes important in determining relative importance of radiative contributions. Differences for horizontal and vertical surfaces have definitely been established. Likewise, cylinders have been observed to yield results differing from those obtained on plane surfaces. Most investigators observe h to increase as pressure increases. Likewise, liquid velocity increases produce increased coefficients, according to most investigators.

QUALITY EFFECTS IN BOILING HEAT TRANSFER

The influence of net vapor generation on the heat-transfer coefficient in the nucleate-boiling regime parallels forced-convection effects. In the low-quality regions the vapor phase will likely remain dispersed in the liquid matrix, thus resulting in a reduction of the average fluid density. Under these conditions slip can be considered negligible and an increase in the velocity will occur. The film at the heating surface will remain essentially the same, except that the boundary-layer thickness will decrease as the velocity is increased. Eventually a velocity will be reached at which the bubbles are sheared from the wall shortly after nucleating. At this point, the film thickness has been reduced to where it no longer offers the resistance to heat transfer that it would at lower velocities. A given heat flux can be sustained at lower ΔTs , and hence the surface temperature drops. This in turn deactivates sites and decreases the vapor generation at the surface. The effect of the growth and collapse of bubbles on the boundary layer becomes less significant.

Sterman and Styushin 1030 observed that the critical flux was increased by quality increases. Their observations with isopropyl alcohol in stainless steel tubes showed that the critical flux was always approached first in the low-quality regions. They postulate that since bubbles are removed from the surface at smaller diameters for increased flow rates, a greater number of sites can be activated before the growing bubbles begin to merge and blanket the surface. This requires a greater ΔT at the critical point, and hence a greater heat flux. Mumm 803 also observed that the heat-transfer coefficient increased with quality for qualities ranging up to 50%. For higher values a rapid decrease in the coefficient was observed, with burnout resulting for qualities of about 70%. His correlation for the Nusselt number includes quality as a parameter [see Eq. (8)]. McAdams et al. 734 and Rohsenow and Clark 938 observed an increase in h with quality increases.

Most investigators agree that for qualities below 50%, improved coefficients will be observed as χ increases. However, at higher qualities considerable disagreement exists as to the exact behavior to be expected. An examination of the flow pattern sheds some light on the heat-transfer phenomena. At low qualities the flowing stream is essentially liquid, with vapor dispersed as a discontinuous phase. At higher qualities the vapor coalesces, but liquid remains as the continuous phase. For sufficiently high vapor velocities such annular flow eventually develops that vapor with dispersed liquid droplets moves along the tube axis, while a liquid film flows along the tube wall. For liquids which wet the surface, high heat-transfer coefficients persist in this flow regime.

Forced-convection effects have probably suppressed any surface boiling, but the high velocity of the gas phase through the core removes all but a thin liquid film at the tube wall, thus reducing the resistance to heat transfer. Eventually the liquid film is reduced to a point where it is difficult to detect. This stage is referred to as fog or mist flow. However, the surface is still

supplied with sufficient liquid to remove the necessary heat load by vaporization. As the quality continues to rise, a point is finally reached where insufficient liquid reaches the surface to dissipate the high energy fluxes. This "dry wall condition" results in rapid temperature increases at the surface, and burnout occurs. Investigators refer to this type of critical condition as two-phase burnout.

Several investigators have measured high quality heat-transfer coefficients. McAdams et al. 734 observed for water-steam a drop in the heat-transfer coefficient for qualities above 40% at 24 psi and 71 psi. Dengler 255 observed three mechanisms operative over the quality range he studied. At low qualities nucleate boiling seemed to control; at higher qualities forced convection effects appeared to dominate. For qualities from 47% (G = .171 x 10^6 lb/hr ft²) to 84% (G = .044 x 10^6 lb/hr ft²) sharp decreases in the heat-transfer coefficient were observed. This phenomenon was attributed to "dry wall conditions."

Parker and Grosh 833 studied the heat transfer characteristics in the mist-flow regime for steam and water droplets moving vertically upward in a tube. Heat flux was varied from 3,020 to 20,700 Btu/hr ft² with inlet qualities from 89-100%. Their results showed that equilibrium was not necessarily attained between the droplets and vapor, and that considerable superheating of the vapor was possible in the presence of droplets. They also observed the heat-transfer coefficient to be a strong function of surface temperature. Above a certain critical temperature, spheroidal behavior was observed with coefficients approximately the same as for dry steam. Surface temperatures below this critical produced coefficients 3 to 6 times greater than dry steam values. Flux and quality effects on this temperature appeared interrelated. Higher qualities and/or fluxes tend to promote the spheroidal state. Any method of directing the dispersed liquid phase toward the walls is likely to increase the heat-transfer coefficient in the very-high-quality regions.

Guerrieri and $Talty^{418}$ have attempted to separate the mechanisms of boiling and convection in high-quality heat transfer. They present the following expression for the two-phase heat-transfer coefficient:

$$h_c = 3.4 h_{\ell} \left[\frac{1}{x_{tt}} \right]^{0.45}$$
 (38)

where \textbf{h}_{ℓ} is the single-phase liquid coefficient given by the Dittus-Boelter equation, and χ_{tt} is the Martinelli parameter. They relate boiling-film coefficients when superimposed on convective effects by the following formula:

$$h = 0.187 h_c \left[\frac{r^*}{\delta} \right]^{-5/9}$$
 (39)

where r^* is the radius of a minimum-sized thermodynamically stable bubble and δ is the laminar film thickness.

These investigators, and others, concur in the conclusion that a convective mechanism becomes controlling for high-quality systems.

AGRAVIC EFFECTS IN BOILING HEAT TRANSFER

Space applications of small nuclear reactors cooled by boiling liquid media have necessitated a better understanding of gravity effects on the heat transfer process. Zero gravity conditions create rather unusual conditions for processes which function due to density differences. It becomes necessary to replace the normal gravitational forces with others, perhaps centrifugal, which will permit the mechanisms usually operative to function at or above their normal efficiency. Investigations using vortex tubes have already demonstrated tremendous increases in the maximum heat flux that can be transferred from surfaces to fluids without incurring burnout. A summary of the agravic work to date follows. Little has been done experimentally with liquid metal systems although several programs are currently underway.

Merte and Clark 766 made a study of the influence of system acceleration on pool boiling heat transfer in saturated distilled water, at approximately atmospheric pressure. The heating surface was a flat disc 3 in. in diameter, with the acceleration vector (1-21 g's) away and normal to it. At low constant values of the heat flux, $\Delta T_{\rm sat}$ decreased as acceleration of the system increased. This is attributed to the increasing contribution of natural convection with acceleration. At high values of heat flux, $\Delta T_{\rm sat}$ increased with increasing acceleration. Some data are presented showing the influence of subcooling with the system under acceleration. Nonboiling data in the same range of a/g is presented.

Costello and Tuthill 223 used a flat, electrically heated ribbon mounted near the periphery of a cylinder filled with distilled water at essentially atmospheric pressure. The system was spun about its axis producing effective accelerations normal to surface of a/g = 20 to a/g = 40. The heat flux varied from q/A = 100,000 Btu/hr ft² to 200,000 Btu/hr ft². It was found for the given heat flux that $\Delta T_{\rm sat}$ increased with increasing acceleration, resulting in a decrease in the "heat-transfer coefficient." This increase in $\Delta T_{\rm sat}$ amounted to approximately 5-7°F for an increase in a/g from 1 to 40.

Costello and Adams 222 have measured the maximum heat flux for water from a carbon cylinder at approximately one atmosphere for a/g from 1 to 44. The acceleration was normal to the axis of the cylinder which was electrically heated. In other respects their test apparatus was similar to that previously reported by Costello and Tuthill. 223 The relationship between (q/A)c and a/g follows the $^{1/4}$ -power law for a/g in the range from 10 to 44. Below a/g of 10 a power-law representation between these quantities was also found, but with an exponent somewhat less than $^{1/4}$.

Gambill and Greene 370 , 372 attained a critical heat flux of 55 x 10 Btu/hr ft² with water flowing in a vortex in an electrically heated tube. This was attributed to the effect of the centrifugal acceleration estimated to be 18,000 times normal gravity on the bubbles forming at the heating surface. However, the contribution of forced, as well as free, convection could not be isolated.

Siegel and Usiskin⁹⁹⁷ performed a photographic study of boiling water at one atm from several heater configurations in the absence of a gravitational field. No attempts were made to measure heat flux or temperatures. The bubbles appeared to grow and remain in the vicinity of the heating surface.

Measurements of the critical heat flux from a platinum wire 0.0453 in. in diameter were made 1102 in saturated distilled water in various force fields of $0 \le a/g \le 1$. The burnout heat flux decreased with reduced force fields but still had a finite value at a/g = 0. Measurements were also made of bubble sizes at departure and of bubble rise velocities with reduced gravities.

Merte and Clark⁷⁶⁵ have studied the boiling of saturated liquid nitrogen at atmospheric pressure from a l-in.-diameter sphere for standard gravity and at near-zero gravity for 1.4 sec duration. The sphere is used as a dynamic calorimeter for continuous measurements from film through nucleate boiling. In the nucleate-boiling region, the characteristics are the same as at standard gravity, indicating perhaps that buoyant forces play a minor role in promoting the turbulence associated with boiling.

In Ref. 661 various liquid configurations, based on the principle of minimum energy, are presented for containers partially filled with a liquid and subjected to zero gravity. Consideration of tank outlet vents under this condition are examined. For liquids which wet the container wall, it is probable that the final zero gravity configuration is a wetted wall with an internally centered gas bubble. For nonwetting liquid, roughly the opposite effect is anticipated.

A feasibility study was made 911 for boiling and condensing mercury with zero gravity using parabolic flight of an aircraft. No quantitative heat transfer measurements were made. The authors discuss problems regarding slug motion of mercury in flow passages and undesired movement of condensed mercury back into the boiler which they encountered in their study.

Reference 1093 discusses general problem areas of heat transfer, and those anticipated in future space vehicles. Tests of the behavior of gases released in fluids and in mercury condensing tests are described. Presentation is qualitative.

INTERFACE CONSIDERATIONS IN BOILING HEAT TRANSFER

There is a substantial agreement, in the published works on boiling, that homogeneous nucleation, i.e., the nucleation of a bubble from within the bulk liquid is, in general, seldom obtained, 438 because the formation of a bubble must create surface at the expense of volume-free energy. Adequate quantitative treatments of this subject are available in the literature. 1094 They show that the superheat required to obtain a bubble by homogeneous nucleation is larger than that obtained experimentally. The critical size of the bubble nucleus is shown to be proportional to the liquid-vapor surface tension, and the free energy of activation to form the bubble is proportional to the surface tension cubed. The surface tensions of liquid metals are from 4 to 200 times greater than those of aqueous solutions and, therefore, the improbability of homogeneous nucleation of liquid metals is even greater than that of the systems that have received more attention.

To explain the relatively low superheats generally found in boiling systems, heterogeneous nucleation is indicated. In liquid-metal systems, the savings of energy through heterogeneous nucleation are even greater than those in aqueous or organic systems. The essential condition for the operation of an effective heterogeneous nucleation catalyst is that its surface be more susceptible to wetting by the newborn phase than by the mother phase. 1094 In short, a nonwetted surface would tend to promote nucleation in boiling. It is not necessary that lack of wetting be general over the whole surface, but rather that suitable specific locations, as discussed above, be provided. In the limiting case where the surface is completely nonwettable by the liquid, the vapor film would always exist and nucleation is unnecessary.

Unfortunately, the conditions for nucleation of the vapor bubbles and for the prevention of film boiling are diametrically opposed. For easier bubble detachment from the surface, the highest possible affinity of the liquid for the solid and the lowest possible affinity of the vapor for the solid are desired. These conditions would be met when the resultant force of the surface stress tensor would have its component at a given location under the liquid, pointing out of the surface, as opposed to a location under a vapor spot where it should point into the surface (see Appendix C).

Increasing the relative preference of liquid for solid has been shown to have the following effects on heat transfer. First, under conditions where convection is the predominant mode of transfer, a wetted condition at the wall gives higher heat transfer coefficients for a given ΔT . Larson Larson Larson has postulated that as the temperature differential is raised, a well-wetted surface, as opposed to a surface not so well wetted, has a slower rate of increase of heat flux. This would be due to the more difficult nucleation of bubbles. However, it has been shown 150 that alteration of the surface energies will prolong the nucleate regime and give a higher critical heat flux. Russian workers have shown that additions of magnesium to mercury in controlled amounts continue

to raise both the critical heat flux and the corresponding critical temperature difference. Extension of the nucleate regime is to be expected from the more favorable conditions for bubble detachment, as opposed to the spreading of the vapor over the solid surface, which would result in the onset of film boiling.

To obtain maximum heat transfer from a surface, the following conditions should be met by the solid-liquid combination. First, the surface should be completely wetted by the liquid to an extent limited by loss of strength due to stress corrosion or penetration of grain boundaries (see Appendix C). Secondly, the surface should have a controlled amount and distribution of a very fine second phase chosen so that the liquid does not wet it. This phase will then serve as a nucleation catalyst. And finally, external stimuli such as the application of elastic stress might be used to increase further the degree of heterogeneity of the surface, allowing the more complete wetting of at least some of the grains.

SUMMARIES OF EXPERIMENTAL LIQUID-METAL-BOILING PROGRAMS

Considerable activity in liquid-metal-boiling heat transfer has taken place during the past decade. Earlier efforts associated with the mercury boiler had produced some results both in the United States and in Russia. However, the first comprehensive boiling study in which other metals were considered was performed by R. E. Lyon at The University of Michigan in 1953. Since that time C. F. Bonilla at Columbia has performed boiling studies on mercury and sodium-potassium systems. He has also contributed several other studies, including several liquid-metal-condensing investigations. Several other programs during this period have produced results which have appeared in the literature. Summaries of these are included in the following text.

The renewed emphasis on high-flux, high-temperature heat transfer has resulted in the establishment of experimental programs in laboratories throughout the world. Table I summarizes most of these programs. Some are designed to yield corrosion data and others to measure heat-transfer coefficients. Effects of pressure, velocity, subcooling, quality, surface characteristics, and fluid properties on heat-transfer characteristics are all receiving attention. Much of this work is in its early stages, and results are still unavailable.

The summaries which follow include a description of the equipment and the experimental procedure as well as an analysis of the results. Conflicting data and conclusions are reported. Subsequent results will undoubtedly clarify many of today's uncertainties.

KUTATELADZE, S. S. 617

This book is a supplement to the Soviet Journal of Atomic Energy (1958) and is devoted entirely to the problems of utilizing liquid metals as heat-transfer

TABLE I
LIQUID-METAL EXPERIMENTAL PROGRAMS*

Organization	Test Fluid	Test Objective	Status	
Aerojet-General Nucleonics	Rb, Cs	Operating Loops		
AiResearch Manufacturing Co.	K	Materials	Fabrication	
AiResearch Manufacturing Co.	K	O-g Boiling Exp.	Fabrication	
Argonne National Laboratory	K	Boiling Heat Transfer	Design	
Argonne National Laboratory	Cd	Boiling Heat Transfer	Operation	
Argonne National Laboratory	Hg-N2	Two-Phase Studies		
Atomics International	Na-K(78)	Heat Transfer and Hydraulics		
Atomics International	Hg	Boiling Heat Transfer	" wire soon com	
Atomics International	Na	Burnout Studies		
Atomics International	Na	Condensing Studies		
Brookhaven National Lab.	K	Boiling Heat Transfer		
Columbia University	Na	Condensing Studies		
Electro-Optical Systems, Inc.	Hg	Condensing O-g Studies		
General Electric	K, Na	Materials	Fabrication	
General Electric	K, Na	Heat Transfer	Fabrication	
General Electric	K	Turbine	Design	
Marquandt Corporation	Li	√ 40 = = =		
The Martin Company	Li	Operating Loops		
MSA	Na	Materials	Operating	
NASA-Lewis	Na	Turbine	Design	
NASA-Lewis	Na	Turbine	Fabrication	
NASA-Lewis	Na	Pumps	Fabrication	
NASA-Lewis	Na	Boiling Heat Transfer	Fabrication	
NASA-Lewis	Na	Condensing Studies	Design	
NASA -NDA	K, Na	Materials	Design	
Nuclear Develop. Corp.				
Oak Ridge National Laboratory	K	Boiling Heat Transfer	Operating	
Pratt and Whitney Aircraft	Li, Na-K	Materials	Operating	
Rocketdyne	K	Materials	Operating	
Sundstrand Aviation	Rb	Condensing	Fabrication	
Sundstrand Aviation	Rb	Boiling Fabri		
Sundstrand Aviation	Rb	Heat Storage	Design	

^{*}From Aeronautical Systems Division, Wright-Patterson Air Force Base.

media in nuclear power. It presumably contains all important liquid-metal heat-transfer data collected during the previous ten years in Russia as well as some from the United States and other countries. There are chapters covering selective liquid-metal properties, general areas of liquid-metal applicability, corrosion studies, and instrumentation in liquid-metal systems. The present summary covers only that portion dealing with heat transfer of liquid metals during boiling.

A discussion concerning wettability and the hydrodynamic characteristics in vapor-liquid mixtures is given. The degree of wettability as characterized by the premature origination of film boiling is qualitatively discussed. If the liquid does not wet the wall of the tube, it is pointed out that the flow pattern thus formed is one in which the vapor bubbles that form remain adjacent to the wall, retarding the heat transfer to the liquid. From an investigation by Lozh-kin and $Krol^{684}$ for mercury boiling in a glass tube, the fraction of the surface covered by vapor varied from 34% to 78.5-87.5% for $10,000 \, \text{Kcal/(m}^2)(\text{hr})$ and $25,000 \, \text{Kcal/(m}^2)(\text{hr})$, respectively.

Several plots are given showing the fundamental hydrodynamic characteristic of two-phase flow as obtained by Gremilov. 403 The vapor fraction is shown as a function of the ratio of the reduced vapor velocity to the convective velocity with the Froude number (u^2/d) as a parameter. A separate plot is given, correcting for tube inclinations. For foreed-convective boiling Gremilov's results clearly show that the effect on the integral hydraulic characteristics of two-phase flow is small and that mercury-vapor systems behave much like steam-water systems. Pressure drop may be expressed to an accuracy of $\pm 15\%$ by the following:

$$\Delta p = \frac{f u_m \rho_{\ell} L}{2gd} \left[1 + \left(1 + \frac{\rho_{v}}{\rho_{\ell}} \right) \frac{u_{v}}{u_{m}} \right]$$
 (40)

As further proof that nonwetting metals approximate the film-boiling regime of wetting metals, a number of investigations 403 , 545 , 591 , 626 , 683 , 684 , 1044 , 1046 are cited which agree with this hypothesis. It is shown from an investigation by Lozhkin that turbulent promoters markedly improve the boiling heat-transfer coefficient. Two tables are given comparing the heat-transfer coefficients at three stages in the heat-transfer loop for baffled and nonbaffled flow. For example, in boiling at 25,400 Kcal/(m²)(hr), the heat-transfer coefficients are 470 and 5640 Kcal/(m²)(hr)(°C) for non-vortex and vortex flow, respectively. The most significant increases were observed in the upper portion of the tubes where boiling became more fully developed and net vapor generation resulted.

The effect of the tube diameter does not have any appreciable effect on the nucleate-boiling regime. This has been verified for diameters up to 40 mm.

The dependence of the heat-transfer coefficient on the heat-flux density is discussed. It is stated that when a liquid metal wets the heat-transfer surface and the heat load is below the critical, the following equation is applicable:

$$h = \phi (q/A)^n \tag{41}$$

Korneev's data⁵⁹¹on the boiling of a magnesium amalgam on a vertical steel tube placed in a large volume of liquid could be correlated with this equation with n equaling 0.59. His data also demonstrated that the above correlation was independent of magnesium concentration over the range of .01-.03%. However, an increased magnesium concentration was observed to shift the critical flux upwards (see Fig. 2).

Three semi-empirical equations derived from investigations with nonmetallic liquids are compared with data on liquid metal systems: (1) the equation of Averin and Kruzhilin, 45

$$\frac{h}{k_{\ell}} \left[\frac{\sigma}{\rho_{\ell} - \rho_{V}} \right]^{1/2} = 0.082 \text{ Pr}_{\ell}^{0.45} \left[\frac{\rho_{V} \lambda \text{ q/A}}{\text{AT}_{S} k (\rho_{\ell} - \rho_{V})} \right] \left[\frac{c \sigma^{0.5} \text{AT}_{S} \rho_{\ell} (\rho_{\ell} - \rho_{V})^{0.5}}{(\lambda \rho_{V})^{2}} \right]$$
(42)

(2) the equation of Kutateladze, 625

$$\frac{h}{k_{\ell}} \left[\frac{\sigma}{\rho_{\ell} - \rho_{V}} \right]^{0.5} = 0.44 \operatorname{Pr}_{\ell}^{0.35} \left[\frac{q/A \ p \ 10^{-4}}{\lambda \rho_{V} (\rho_{\ell} - \rho_{V}) \nu} \right]^{0.7}$$
(43)

(3) Borishanskii's and Minchenko's 131 alteration of Kutateladze's equation

$$\frac{h}{k_{\ell}} \left[\frac{\sigma}{\rho_{\ell} - \rho_{V}} \right]^{0.5} = 0.55 \operatorname{Pr}_{\ell}^{0.7} \left[\frac{q/A \ p \ 10^{-4}}{\lambda \rho_{\ell} (\rho_{\ell} - \rho_{V}) \nu} \right]^{0.7}$$
(44)

The equations are compared with experimental data on magnesium-mercury, and sodium. 623,695,696,697,698 The results are reproduced in Table II. It is seen that Eq. (43) gives the best value for magnesium-mercury amalgams while Eqs. (43) and (44) show nearly equal deviations for sodium.

The data of Styrikovich, Semenovker, and Sovin 1045 on heat transfer to mercury during forced convection inside vertical steel tubes show an increase in h as velocity increases and as tube diameter decreases. Their fluxes ranged from 25,000-70,000 Kcal/(m²)(hr) diameters from 21-40 mm and velocities up to 0.9 m/sec. Additional data for nonstratified flow in inclined tubes with fluxes extended to 98,000 produced coefficients up to 1100 Kcal/(m²)(hr)(°C) for nonwetting mercury.

For nonstratified flow vertical and inclined tubes yield indistinguishable values. For stratified flow a reduction is observed in the coefficient, particularly at the top of the tube. For nonwetting fluids the decrease was observed to occur before boiling actually occurred, at about the point where the wall temperature reached the saturation temperature.

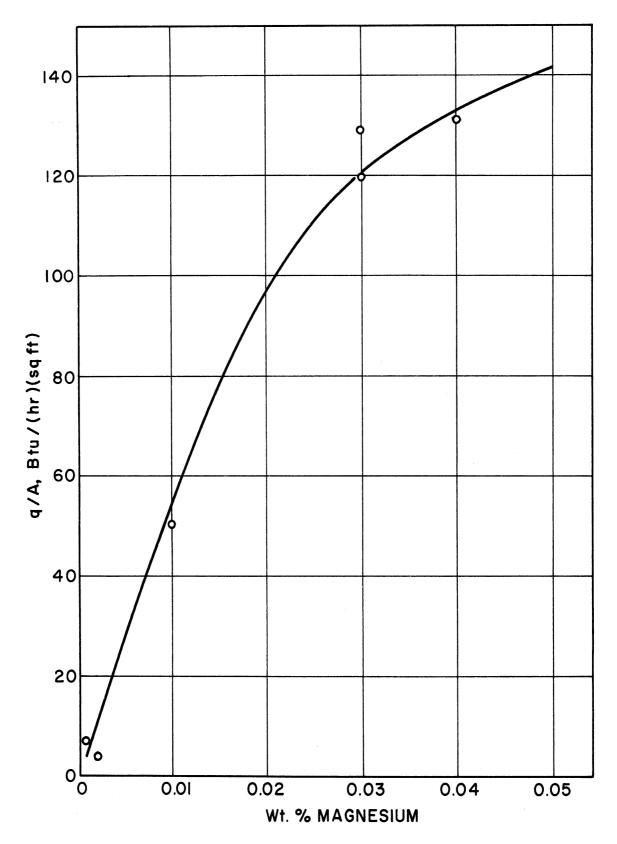


Fig. 2. Effect of magnesium concentration in mercury on the critical heat supply for boiling in a large volume (Kutateladze 617).

TABLE II

COMPARISON OF THE EXPERIMENTAL VALUES OF THE COEFFICIENT IN EQ. (41);

CALCULATED VALUES

Metal	Pressure (atm)	Experimenta	1	Eq. (42)	$\emptyset = \frac{1}{q/A}$ Eq. (43)	Eq. (44)
Magnesium-mercury amalgam	1.03		3.30	24.6	5.19	1.1
Magnesium-mercury amalgam	10.3	$[q/A < 2x10^{+5}]$	4.75	27.5	6.42	1.5
Sodium	1.03	$[q/A < lxl0^5]$	8	56.6	22.6	3.72

Korneev's data^{591,592} show the heat transfer coefficient of mercury as a function of velocity with parameters of the heat-flux density at the top, middle, and bottom of a horizontal boiling tube (see Fig. 3). As is evident from the figure, the velocity has a pronounced effect on the coefficient at the top of the tube. The velocity above which heat transfer in the upper portion of the tube remains at an almost constant level is given by the following:

$$u = 22 \times 10^{-5} \text{ q/A}^{0.42} \text{ d}^{0.76} \text{ m/sec}$$
 (45)

where q/A is in $Kcal/(m^2)(hr)$ and d is in mm. The heat-transfer coefficient at this velocity level is

$$h = 12 q/A^{0.67} u^{0.3} d^{-0.45}$$
 (46)

for 5000 < q/A < 70,000 Kcal/(m²)(hr); 13 \leq d \leq 40 mm, 1 \leq 12 atm, 1 < u \leq 19 m/sec.

LYON, R. E. 695,696

Lyon was the first investigator in the United States to make an extensive study of the heat-transfer characteristics of liquid metals in the boiling regime. The metals investigated were mercury, mercury containing 0.10% sodium, mercury containing 0.02% magnesium and 0.0001% titanium, sodium, sodium-potassium alloy (56-59 wt %K), and cadmium. Water was also boiled as a basis for comparison.

The experimental apparatus was constructed principally of 304 stainless steel with a 3/4-in. (OD), 16-gauge, type-316, stainless-steel boiler tube

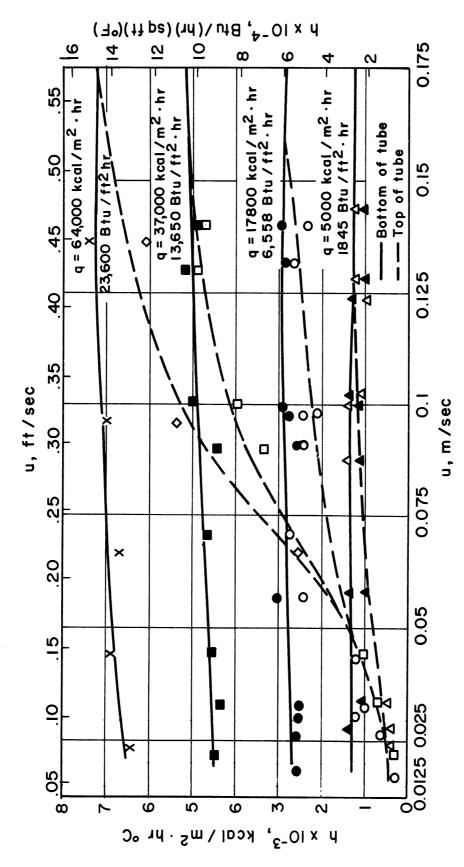


Fig. 3. Dependence of heat transfer from the wall to magnesium amalgam on the reduced velocity of the liquid u_0' and the heat supply q/A for a horizontally heated tube (D = 17.6 mm) with a vapor velocity u = 2 mm/sec (Kutateladze, et al. (617)

five inches long. Chromel-alumel thermocouples imbedded in the wall of the heating tube measured temperatures that could be used to determine the boiling surface temperature. An electrical resistance element inserted inside the boiler tube supplied a heat source up to 130,000 Btu/(hr)(sq ft). All tests were performed at atmospheric pressure under conditions of natural convection with the entire system blanketed with nitrogen. Condensing of the bulk boiling metals was accomplished in an air-cooled condenser.

Before each liquid was boiled, the system was evacuated and re-pressurized with nitrogen. A known quantity of the test metal was emitted to the boiler and the heater element turned on and adjusted to the desired level. After steady-state conditions had been established, the required readings were recorded.

In the analysis of the experimental error, Lyon found the probable error in measuring the boiling surface temperature to be 3.2°F at 50,000 Btu/(hr)(sq ft), the error from potentiometer calibrations and readings to be ± 0.4 °F, yielding a probable error in the temperature difference of ± 0.8 °F; and the overall error in computing the heat flux density to be $\pm 8\%$ at 10,000 Btu/(hr)(sq ft) and $\pm 6\%$ at 100,000 Btu/(hr)(sq ft).

Figures 4, 5, and 6 show plots of the heat flux density (q/A) as a function of the temperature difference between the bulk liquid and the heat-transfer surface (ΔT) for all the test liquid metals. Figure 7 shows the heat-transfer coefficient (h) as a function of ΔT . It is seen that the sodium, sodium-potassium alloy, and mercury containing magnesium and titanium give extremely good heat-transfer characteristics. Nucleate-boiling heat-transfer coefficients of nearly 15,000 Btu/(hr)(sq ft)(°F) were found for both sodium boiling at 1620°F and sodium-potassium alloy boiling at 1500°F, for a ΔT of less than 10°F. For mercury with 0.02% magnesium and 0.0001% titanium, a heat flux of 100,000 Btu/(hr) (sq ft) at a ΔT equal to 12°F was attained with no apparent indication of the critical heat flux being reached.

Certain pertinent conclusions were drawn from the investigation. Cadmium and pure mercury experienced only film boiling upon reaching the saturation temperature. This effect was attributed to their nonwetting features. The effects of additives in mercury are to increase the heat-transfer coefficient (in the case of 0.10% sodium the heat-transfer coefficient reached ten times as high as with pure mercury) because they promote wetting. Temperature fluctuations at low heat fluxes were observed during the nucleate boiling of sodium-potassium alloy and mercury containing magnesium and titanium, and were explained on the basis of the high heat-transfer rates in the liquid metals.

With the exception of pure mercury and cadmium, there was no indication that the critical had been approached. The condensing capacity prohibited operation with higher fluxes.

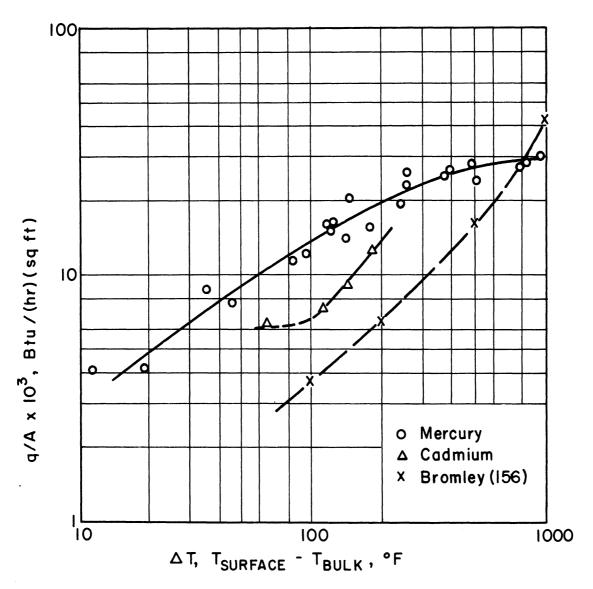


Fig. 4. Heat flux vs. temperature difference for film boiling of mercury and cadmium (Lyon 695,696).

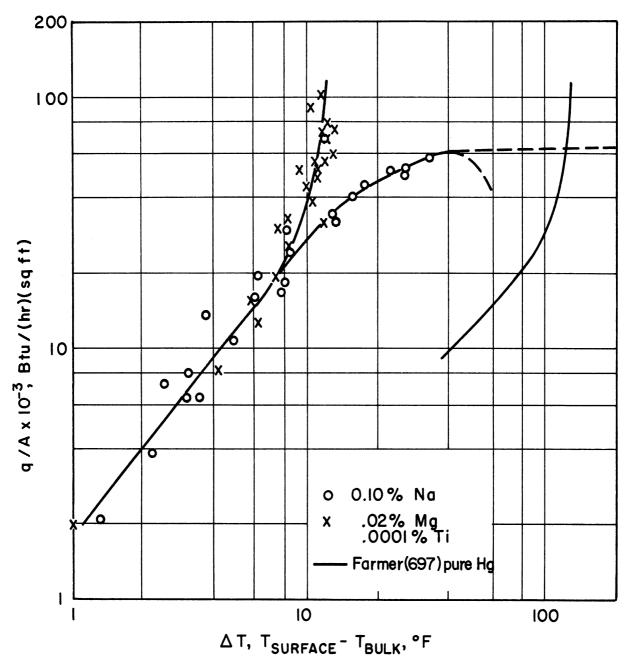


Fig. 5. Heat flux vs. temperature difference for boiling mercury with wetting agents (Lyon 695,696).

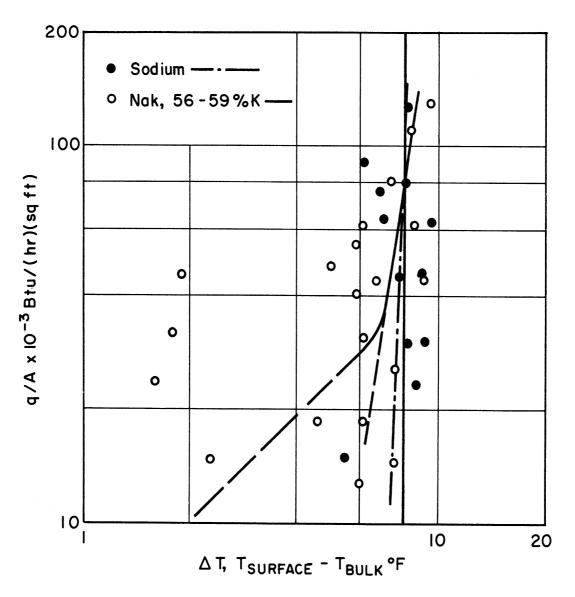


Fig. 6. Heat flux vs. temperature difference for boiling sodium and boiling NaK (Lyon, et al. 695,696).

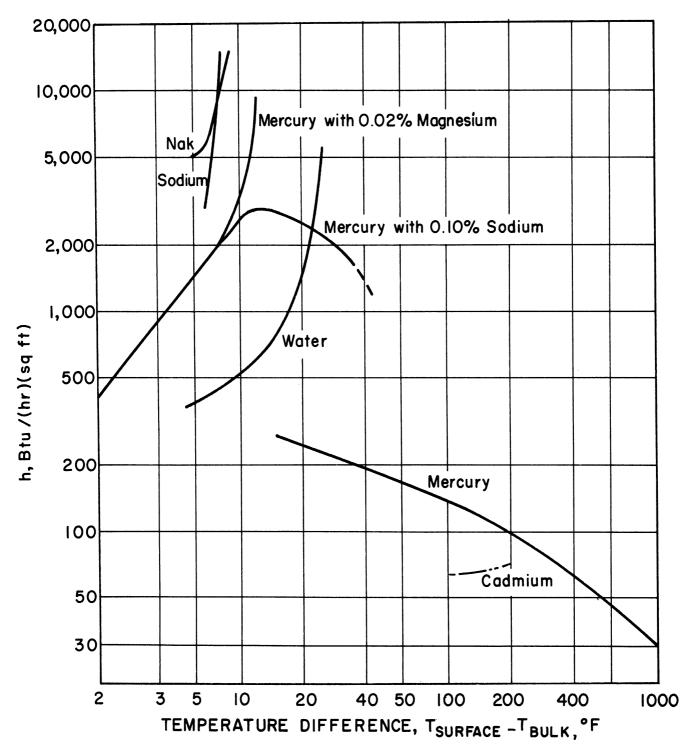


Fig. 7. Comparison of experimental boiling heat transfer coefficients for water and liquid metals (Lyon, <u>et al.</u> 695,696).

BONILLA, C. F. 126

Boiling of mercury was accomplished both with and without wetting agents. The apparatus consisted of a horizontal heating surface of low-carbon steel fitted with a 3-in.-OD stainless-steel tube. The upper portion of the tube served as the condenser. The main heater consisted of a wound Nichrome strip over mica on flat copper fins extending from the bottom of the heating surface plate. The system was blanketed with nitrogen and was operated at pressures from 4 mm mercury to 45 psia with heat-flux densities ranging from 4,000 to 200,000 Btu/(hr) (sq ft). The boiling pool depth varied from 2 to 10 cm. At various times 0.002% magnesium and 0.0001% titanium were added to the mercury to increase its wettability. A guard heater was used to minimize heat loss. The boiling-surface temperature was attained by extrapolating temperatures measured by iron-constantan thermocouples inserted in the boiling block at varied distances from the heating surface. Bulk boiling temperatures were measured using three iron-constantan thermocouples placed in the liquid.

The experimental procedure was quite simple. The apparatus was properly assembled, pressurized to check for leaks, filled with mercury (and additives), evacuated, refilled with nitrogen, and then the heater was turned on to the desired level. After steady-state conditions had been reached (15 to 30 min), the required temperature and power readings were recorded.

No mention was made of the experimental accuracy achieved in the apparatus.

Figures 8, 9, and 10 show the boiling curves for mercury boiled in 2- and 10-cm-deep pools; system pressure is the parameter. Data for each pressure run seem to correlate reasonably well. It can be seen that the effect of pressure diminished as the pool depth increased. It was stated that over a period of a few weeks of constant use, film boiling was not obtained with pure mercury systems. The authors attributed this to mechanical removal of oxygen or oxide from the surface. This may partially account for the fact that Lyon's pure mercury data deviate somewhat from the present data. Lyon experienced film boiling when mercury was boiled, thus yielding a boiling curve with a negative slope and displaced slightly to the right of the present data. 695,696

Figure 11 shows the boiling curve for mercury with the addition of 0.02% magnesium and 0.0001% titanium. The heat flux at constant ΔT is increased some 25% over that obtained by boiling pure mercury. The agreement of Bonilla's data with those of Lyon⁶⁹⁵,696 and Farmer⁸⁸¹ should be noted.

Conclusions reached are as follows:

- (1) Prolonged boiling on stainless steel promotes wetting and increases the heat-flux density for the same temperature-driving force;
- (2) Increasing the pressure of the system reduces the temperature-driving force for the same heat-flux density;

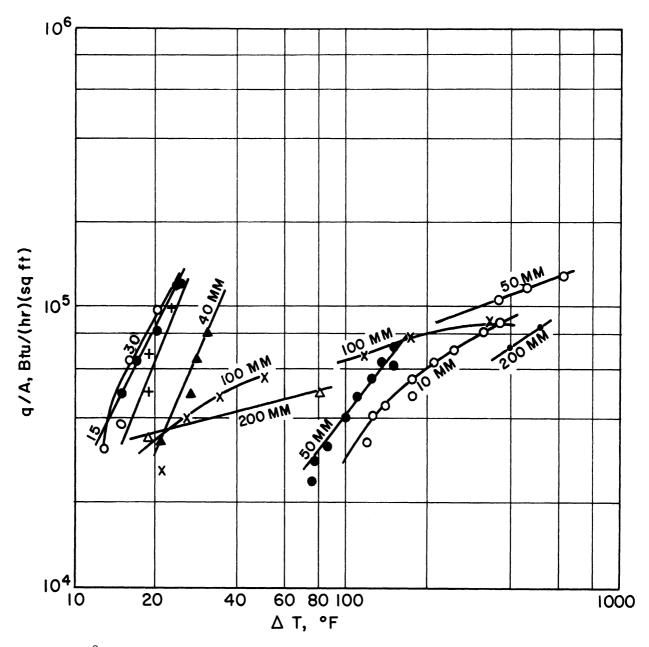


Fig. 8. Boiling of pure mercury 2 cm deep on a horizontal low-carbon-steel plate; parameter: pressure over the liquid in mm Hg absolute or in lb/sq in. gauge (Bonilla126).

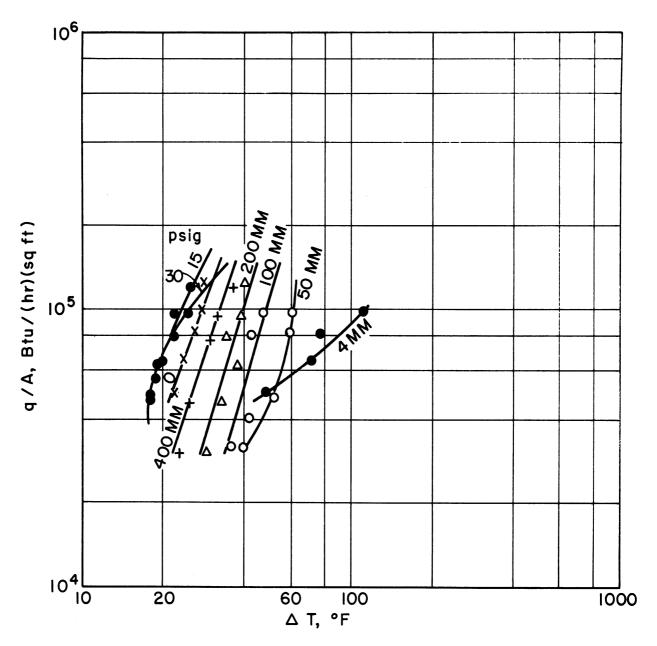


Fig. 9. Boiling of pure mercury 10 cm deep on a horizontal low-carbon-steel plate; parameter: pressure over the liquid in mm Hg absolute or in lb/sq in. gauge (Bonilla 126).

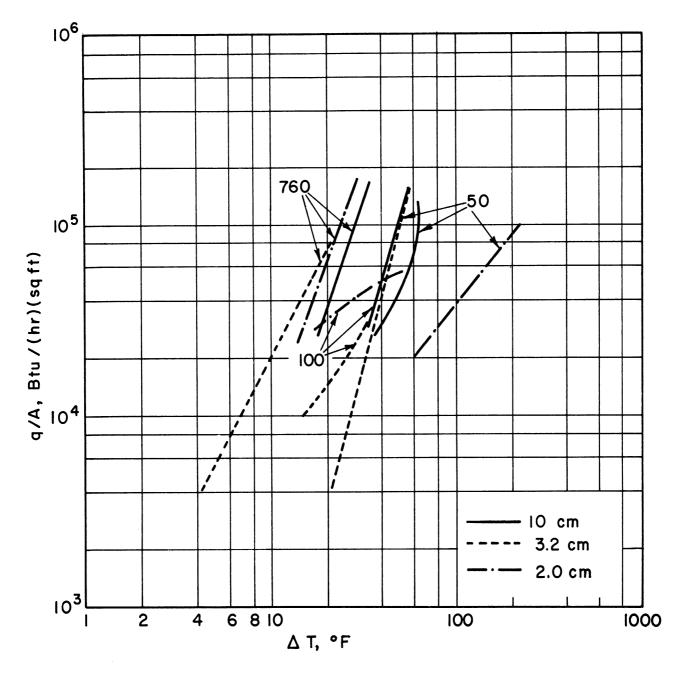


Fig. 10. Effect of depth on the nucleate boiling or pure mercury on a horizontal low-carbon-steel plate; parameter: pressure over the liquid in mm Hg absolute (Bonillal 26).

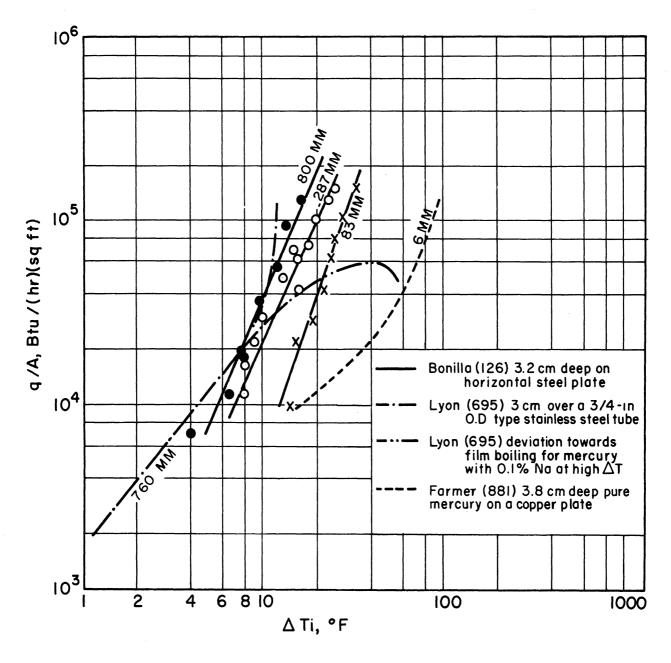


Fig. 11. Boiling of mercury containing 0.02% Mg and 0.0001% Ti; parameter: pressure over the liquid in mm (Bonilla, $\underline{\text{et}}$ $\underline{\text{al}}$. $\underline{\text{126}}$).

- (3) Additives in mercury promote wetting;
- (4) The liquid-metal pool temperature does not change with increased depth;
- (5) Different noise levels are observed for different heat-flux densities.

LIN, C., <u>ET</u> <u>AL</u>. 664

Boiling of mercury containing magnesium was accomplished for heat loads from 5,000 to 47,000 Kcal/(hr)(sq m) and pressures of 1 and 11 atmospheres. No mention is made in the article of the experimental apparatus, procedure, or experimental error.

For pure mercury boiling under atmospheric conditions, the authors found that the heat-transfer coefficient could be expressed by the following:

$$h = 4850 \text{ q/A}^{-0.26} \text{ Kcal/(hr)(sq in.)(°C)}$$
 (47)

For boiling pure mercury under superatmospheric pressures, the authors give the following expression:

$$h = Ap^b q/A0.46 (48)$$

For the pressure interval 4-11 atm the equation gives:

$$h = 7p^{-0.29} q/A^{0.46}$$
 (49)

This indicates that increasing the pressure lowers the heat-transfer coefficient for pure mercury and gives a behavior different from ordinary liquids. Using $q/A = h\Delta T$ and eliminating h from the above equation gives:

$$q/A = 37 p^{-0.537} \Delta T^{1.85}$$
 (50)

Comparing this equation for pure mercury with the one in identical form determined by Madsen and Bonilla 706 for Na-K, one finds that the powers of ΔT and the coefficient seem to be in accord, but the effect of pressure is inverted. The only major difference in behavior of the two metal systems is wettability, which, it is felt, could hardly account for this unusual behavior.

Mercury was then boiled with varied amounts of magnesium added (0.02 to 0.05%). The data indicated that the heat-transfer coefficient could be represented by the formula

$$h = A q/A^n (51)$$

where the constants A and n are given in Table III. It was stated that for the same heat-flux density, 0.05% magnesium results in a heat-transfer coefficient

15-50% higher than that for 0.02% magnesium. Variations in wettability accounted for this effect. Pressure had little or no effect on the heat-transfer characteristics.

TABLE III

CONSTANTS IN EQ. (51)

Magnesium Content (%)	А	В
0.02	13.7	0.43
0.05	2.43	0.63

ROMIE, F. E., BRORARNEY, S. W., AND GIEDT, W. H. 946

Mercury with a small amount of magnesium and a trace of titanium was boiled in a thermal-syphon-type heat-transfer loop fabricated from 7/16-in.-OD, 304 stainless steel. A 4-in. heating section made of 7/16-in.-OD, 1018 steel served as the boiling surface. Electrical power was applied to the heating section giving heat-flux densities as high as 600,000 Btu/(hr)(sq ft) with a 5 mole percent vapor quality. Temperature of the heating surface was determined by measuring the outside wall temperature with a thermocouple and equating to the inner wall. It was found that by cleaning the surface and depositing a thin copper layer on the inside wall that the test fluid readily wet the surface.

Experimentation was begun by completely filling the loop with mercury and then draining out a specified amount of mercury. The pressure in the system could be changed by simply controlling the water rate to the condenser.

The test results are reproduced in Table IV. Probable error in determining the heat flux was estimated at $\pm 20\%$. The exit quality of the mercury was calculated by means of an energy balance. During certain runs, hydrodynamic oscillations in the mercury flow were observed. In all cases these oscillations could be removed either by increasing the heat-flux density and/or increasing the pressure of the system.

Even though a heat-flux density of 600,000 Btu/(hr)(sq ft) was the maximum reached in these tests, it was emphasized by the authors that the thermal and hydrodynamic performance of the loop gave every indication that even higher heat-flux densities could be achieved before reaching the critical heat-flux density.

TABLE IV

RESULTS OF BOILING MERCURY WITH ADDITIONS IN A THERMO-SYPHON HEAT TRANSFER

Exit Quality, (mole/mole)	1	;	1 8	1	t I	!	8	!	0.015	!	;	1	0.001	0.032	0.061	0.10	0.049
Heat Out Heat In	1	2.16	1.65	2.0	1.78	2.50	1.66	1.0	1	;	1	1.03	1	!	1	8	1
Inside Wall Temp. Less Saturation Temp.	1	1	1	1	1	1	1	1	100	!	1	1	69	88	48	75	49
Inside Wall Temp.	8	249	894	504	562	587	659	701	741	453	1	4/9	710	759	922	922	825
Temp. Increase Through Test Section (°F)	!	137	169	217	227	221	221	240	545	214	!	135	153	122	168	165	112
Inlet Temp.	i i	99	282	256	293	320	355	386	392	247	i	7480	488	905	524	929	649
Flow Ve- locity at Inlet to Test Section (ft/sec)	1	2.9	29.0	1.0	1,1	1.9	1,5	1,3	1.1	1	!	1,3	7,4	1.1	0.89	47.0	1.5
Test Section Pressure (psia)	!	1	1	3 8	E 6	i i	l I	8	10	i B	1	1	10	14	17	19	33
Heat Flux, q/A (Btu/hr-ft ² x l0-3)	1	54	92	150	190	230	260	004	094	29	:	210	280	340	470	550	009
Run	\vdash	a	~	†	2	9	_	∞	0	10	11	12	13	14	15	16	17

AVERY, G. W. 47

The effect of the surface geometry on boiling mercury and mercury with 0.1% sodium was studied. The experimental apparatus was similar to that used by Bonilla and co-workers 126 and consisted principally of a horizontal low-carbon steel boiling plate fitted with a 3-in.-diameter, 304 stainless-steel pipe 24 in. long. The heat supply was furnished by nichrome strips wound over 13 mica insulated copper fins brazed to the underside of the boiling plate. A guard heater and insulation surrounded the heater arrangement. Condensing of the metallic vapors was accomplished in 304 stainless-steel tubing extending from the top of the vapor chamber. For the most part, the system was operated at subatmospheric pressures under a cover of nitrogen gas.

The experimental procedure consisted of assembling the apparatus, calibrating the vessel for heat loss, filling with 125 cc of mercury, blanketing with nitrogen, and then setting the heat input to the desired level. After reaching equilibrium the required instrument readings were recorded. Upon completion of one set of runs the vessel was disassembled and the boiling surface grooved. This procedure was repeated.

During the course of the investigation two boiling plates were used. Data for boiling mercury from a smooth surface were obtained before burnout occurred. A similar plate was used to boil mercury and mercury with sodium additions first from a smooth surface and then from a surface milled with parallel, 0.003-in.-wide by 0.004-in.-deep, grooves both 3/8 in. and 1/8 in. apart. A groove spacing of 1/16 in. was also milled, but an equipment failure prevented obtaining data.

Figure 12 shows the data taken. It is seen from this plot that for any given surface, the surface geometry has a significant effect on boiling heat transfer to mercury both with and without additives. Unfortunately, the two different plates, despite efforts to reproduce initial surface conditions, gave considerably different heat-transfer coefficients. The first surface gave heat-transfer characteristics for pure mercury comparable to those obtained from the 1/8-in. grooved surface when boiling 0.1% sodium in mercury from the second plot. This fact leaves many questions unanswered. The author suggests that this may be due to the differences in the microscopic geometry of the surfaces.

The author concluded the following from this investigation:

- (1) Heat-transfer coefficients can be improved by grooving the surface;
- (2) Nitrogen cover gas does not appreciably affect the heat-transfer characteristics;
- (3) "Bumping" is observed primarily during atmospheric nucleate boiling at 40,000 Btu/(hr)(sq ft) or greater;

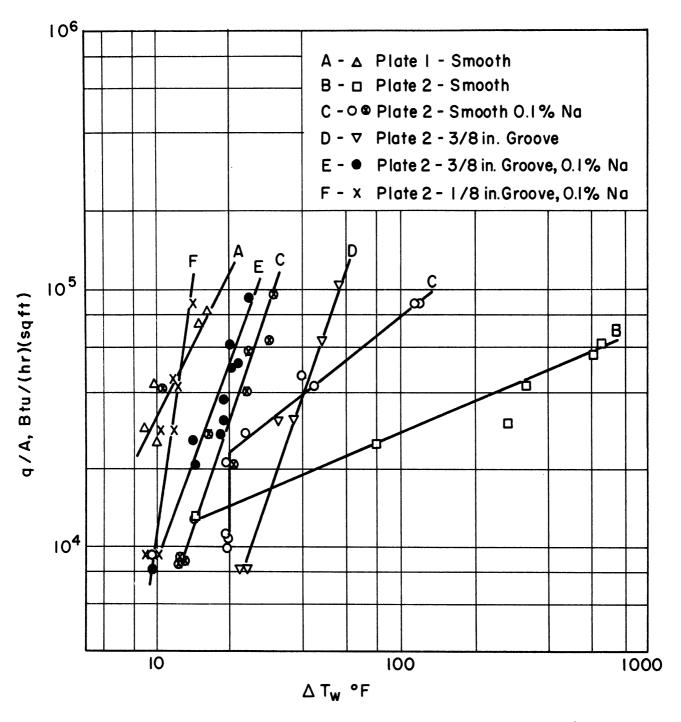


Fig. 12. Mercury boiling on smooth and grooved plates (Avery 14 7).

(4) The effect of surface grooves increases percentagewise with increasing heat-flux density.

MADSEN, N., AND BONILLA, C. F. 706

A sodium-potassium alloy (44 wt % potassium) was pool-boiled from a horizontal low-carbon nickel plate at temperatures in the neighborhood of 1600°F and pressures from 2 mm to 794 mm of mercury. The boiling chamber was fabricated from a 3.068-in.-ID stainless-steel pipe with a water-cooled stainless-steel plug at the top serving as the condensing surface and used to condense the metal vapor.

The vessel was constructed in such a way as to allow for a minimum number of welded joints, and hence reduce the possibility of sodium penetrating cracks or seams in the vessel. Heat to the boiling liquid metal was furnished by molybdenum resistance wire covered with alumina sleeves and wound around molybdenum fins brazed to the bottom of the heater plate. The entire system including the heater enclosure was blanketed with helium gas. Temperatures in the boiling plate were determined by six thermocouples inserted in holes radially drilled and at various depths from the boiling surface; a thermocouple inserted from the top of the boiling vessel measured liquid bulk temperatures.

After cleaning with concentrated hydrochloric acid and testing for leaks, the vessel was calibrated for heat loss and charged with Na-K under a helium blanket. Throughout the tests metal was maintained at a minimum of 900°F, thus reducing heater damage. After the desired heat flux density had been attained and steady state achieved, the necessary readings were taken.

Figure 13 shows all the experimental data taken. Even though the data are more or less random, the authors used the method of least squares twice to obtain the following empirical equation:

$$q/A = 134 p^{0.25} \Delta T^{1.24}$$
 (52)

where ΔT is the temperature difference between the heat-transfer surface and the liquid free surface equilibrium temperature. The probable error is estimated at +38 or -28% of the calculated value.

For a constant heat-flux density the heat-transfer coefficient can be estimated by the following:

$$h = C p^{0.20}$$
 (53)

where C is a constant.

It was found that a temperature gradient existed in the bulk liquid throughout all runs. This presumably would account for the large temperature differences as compared to Lyon. 695,696 The authors suggest that the geometry of

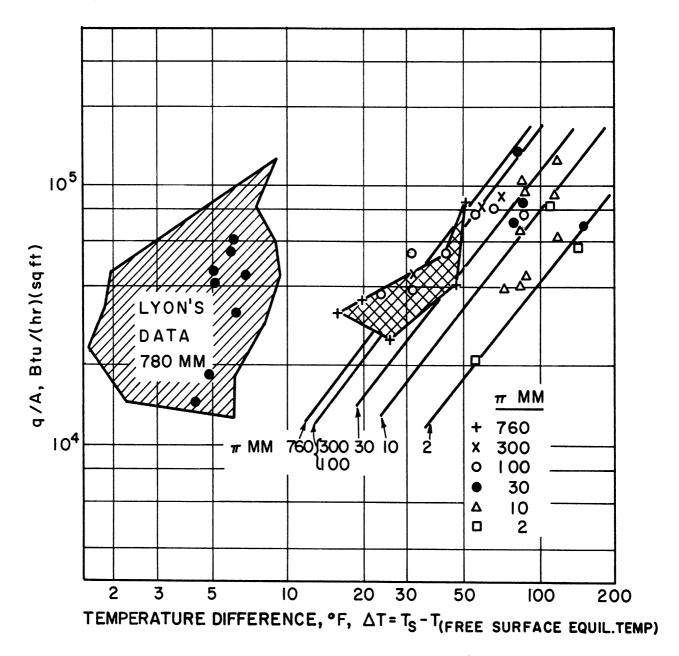


Fig. 13. Comparison of q/A vs. ΔT with Lyon's 695,696 data for boiling NaK (Madsen and Bonilla 706).

Lyon's heater in a larger vessel may promote strong natural-convection currents, thus reducing temperature gradients in the pool.

It was noticed during the investigation that the temperature on the boiling plate near the surface fluctuated randomly, the amplitude of fluctuation changing slightly with different heat-flux densities. Moreover, it was observed that liquid-bulk temperatures and pressures fluctuated, but they were not as significant as the surface-temperature fluctuation. Finally, at low heat-flux densities distinct "bumps" were heard followed by a pronounced temperature drop.

TWO-PHASE FLOW REGIMES

Two-phase (gas-liquid) flow patterns have been studied by a number of investigators, most of whom have used visual means of observation. A recent literature survey on this subject is given by John H. Vohr, lll6 who presents a total reference list of 35 items.

Gas-liquid flows appear in a complex variety of forms, and visual observances have produced a wide variety of terminology. Vohr points out, however, that observers seem to agree as to the basic types of flow patterns that occur, although they differ in classifying subdivisions of the basic patterns. Flow regimes are usually studied in horizontal or vertical flow. The principal difference between these two situations arises when gravity forces cannot be neglected with respect to dynamic forces.

The basic horizontal flow patterns are: 1116

- (1) Bubble flow, in which gas bubbles flow along with the liquid;
- (2) Plug flow, in which the gas bubbles coalesce to form long gas plugs;
- (3) Stratified flow, in which the gas flows in a continuous stream above a smooth gas-liquid interface;
- (4) Wavy flow, which is stratified flow with a wavy interface;
- (5) Slug flow, in which periodic slugs of liquid rapidly travel the length of the duct, leading to pulsating gas-liquid flow;
- (6) Annular flow, in which liquid flows in an annulus adjacent to the walls of the duct and the gas flows as a central core;
- (7) Spray flow, in which the liquid flows as a spray carried by the gas stream.

The following table summarizes and compares the parameters some investigators used in correlating horizontal flow patterns. An obvious consistency in the tabulation is that all authors reported no information concerning the dependence of flow regime upon fluid physical properties.

TABLE V

SUMMARY AND COMPARISON OF PARAMETERS USED IN CORRELATING HORIZONTAL FLOW PATTERNS

Investigators	Parameters Plotted							
Alves ²⁵	Superficial gas velocity vs. superficial li-quid velocity							
Bergelin and Gazley ⁹³	Water rate vs. air rate, both in lb/hr							
White and Huntington 1154	Liquid mass velocity vs. gas mass velocity, both in $lb/hr\ ft^2$							
Johnson and Abou-Sabe ⁵³⁶	Water rate vs. air rate, both in lb/hr							
Krasiakova ⁵⁹⁹	Water velocity vs. air velocity							
Richardson ⁹²¹	Water wt. flow vs. air wt. flow, both in lb/hr							

Kosterin⁵⁹⁶ studied air-water flow patterns in tubes of various diameters, and he presented his findings in a separate plot for each tube. His plots give some indication of the effect of pipe diameter on two-phase flow pattern. Kosterin stated that the transition from divided (stratified or wavy) flow to plug flow should depend on the Froude number, u^2/gD , and that the strong dispersion of gas should depend on the Weber number, $L\rho u^2/\sigma g_C$, where L is a characteristic length associated with bubble size.

Baker 54 proposed a correlation in which the parameters attempted to account for the effect of fluid physical properties on flow regime. His coordinates were G/λ^B and $\text{L}\lambda^B\psi^B/\text{G}$ where G and L are gas and liquid mass velocities and λ^B and ψ^B are given by

$$\lambda^{B} = \left[\left(\frac{\rho_{g}}{0.075} \right) \left(\frac{\rho_{l}}{62.3} \right) \right]^{1/2}$$
 (54)

$$\psi^{B} = \frac{73}{\sigma} \left[u \left(\frac{62.3}{\rho_{\ell}} \right)^{2} \right]^{1/3}$$
 (55)

Baker's plot is shown in Fig. 14. The plot was developed from data on air-water systems, and the extension of the parameters λ^B and ψ^B for correlating two-phase flow regimes in other systems needs verification.

It should be questioned whether all regime transitions depend in the same manner on the same fluid properties. If different transitions depend on differ-

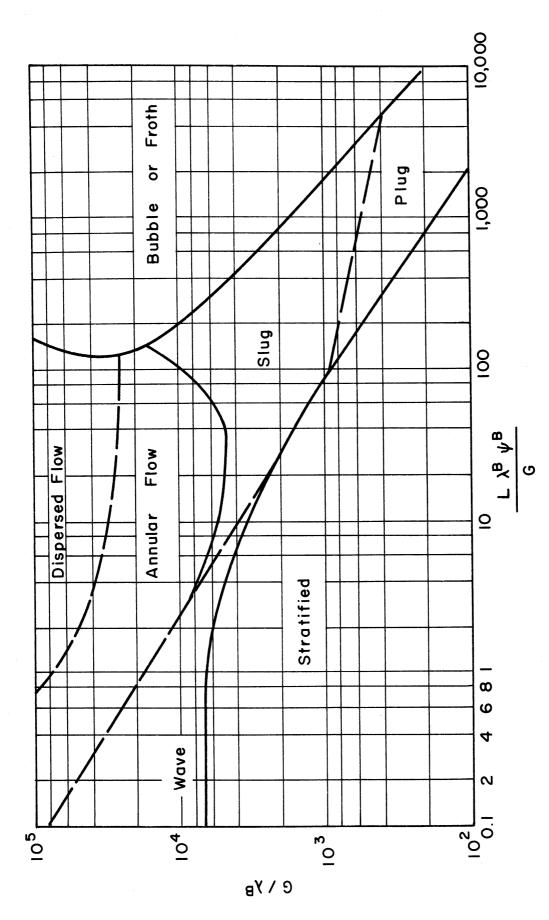


Fig. 14. Flow pattern correlation proposed by Baker.54

ent sets of physical properties, each transition might have to be correlated separately. Clearly, information on the relationship between fluid physical properties and stability of particular flow patterns should be important in gaining understanding of mechanics of two-phase flow regimes.

The flow-pattern plots given by various authors are dissimilar in appearance, and thus are difficult to compare quantitatively. Vohr compared the correlations of several observers by constructing a table, Fig. 15, in which the flow regimes were taken for a constant liquid velocity of 0.5 ft/sec with gas velocities ranging from 1 to 100 ft/sec.

Among those who studied vertical two-phase flow regimes were Govier, Radford, and Dunn, 395 Kosterin, 596 Dengler, 255 and Kozlov, 598

Kosterin and Kozlov plotted vertical flow regimes using delivered volumetric gas content (C_{vd}) and mean mixture velocity (V_{m}). Kozlov also presented mathematical expressions for regime transition boundaries based on C_{vd} and the Froude number (N_{F} or Fr).

The basic vertical flow regimes are: 1116

- (1) Bubble flow, defined as for horizontal flow;
- (2) Piston flow, in which gas flows up in periodic bullet-shaped slugs;
- (3) A region between piston flow and fully developed annular flow in which flow is agitated and complex. Some of the terms for this range are <u>dispersed-plug flow</u>, <u>emulsion flow</u>, <u>turbulent flow</u>, <u>semi-annular flow</u>;
- (4) Annular flow, defined as for horizontal flow;
- (5) Spray flow, defined as for horizontal flow.

Some studies have been made of flow patterns in natural-circulation boiling, and the results are quite similar to those for nonboiling, vertical, two-phase flow. Apparently no studies have been made on forced-circulation boiling flow regimes, but these regimes are expected to differ widely from those in non-boiling two-phase flow due to induced agitation and rapid generation of vapor at fluid boundaries. Vohr is commencing a visual and photographic study of flow regimes in forced-circulation boiling. Wallis and Griffith studied gas and liquid distributions in a two-phase boiling analogy. Their results indicate that flow patterns may be most strongly affected by bubble-formation rate, and that nonboiling and natural-circulation boiling patterns do not apply.

No flow regime studies have been reported for two-phase flow in metallic systems.

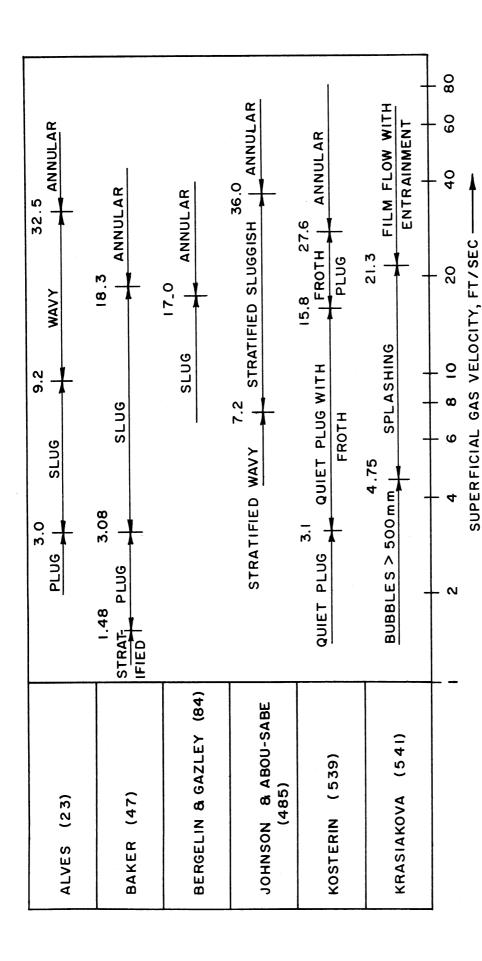


Fig. 15. Horizontal air-water flow pattern regimes for superficial water velocity = $0.5 \text{ ft/sec (Vohr}^{1116}$).

TWO-PHASE PRESSURE DROP

The pressure drop occurring during flow of a boiling mixture includes, in addition to the frictional loss, a loss resulting from the rate of increase of momentum of the mixture as it flows through the tube and vaporizes. Such momentum pressure drops are often quite significant, and in order to predict them one needs to know true gas velocity which in turn demands knowledge of vapor volume fraction. Homogeneous flow should not be assumed.

The first significant two-phase pressure drop study in the United States was made by Boelter and Kepner 116 around 1939. In 1944 Martinelli and coworkers 721 proposed a method for predicting horizontal, isothermal, two-phase pressure drop. The Martinelli method assumes that the frictional pressure loss is the same for each phase and is equivalent to the static pressure drop, i.e., momentum and head losses are neglected. The method proposes a two-phase flow modulus χ , a function of fluid properties, which is used to correlate parameters \emptyset .

$$\phi_{\rm g}^2 = \frac{(\Delta P/\Delta L)_{\rm TPF}}{(\Delta P/\Delta L)_{\rm g}}$$
 (56)

$$\phi_{\ell}^{2} = \frac{(\Delta P/\Delta L)_{TPF}}{(\Delta P/\Delta L)_{\ell}}$$
 (57)

where $(\Delta P/\Delta L)_{TPF}$ = two-phase frictional pressure drop and $(\Delta P/\Delta L)_{g(or \ \ell)}$ = pressure drop if gas (or liquid) phase were flowing alone in the tube.

Lockhart and Martinelli 670 improved the correlation in 1949 when they found that $(\Delta P/\Delta L)_{\ell}/(\Delta P/\Delta L)_{g}=X$ gave considerable improvement in the \emptyset correlations. The Martinelli procedures utilize a notion of "flow type" based on whether laminar (Re < 1000) or turbulent (Re > 2000) flow would exist if the phase considered were flowing alone. Several attempts have been made to improve analytically the Martinelli method. 92,657 Some investigators feel the method could be improved by considering flow-pattern effects.

Friction factor models have been proposed for both horizontal and vertical two-phase flow. This concept was most recently used for horizontal flow by Bertuzzi, Tek, and Poettmann. 105 The authors claim that the variables which set the flow pattern also determine pressure drop, making possible a generalized solution independent of flow pattern. The development is based on a steady-state, total-energy balance, and the two-phase "f" factor is correlated against a two-phase Reynolds number function.

A recent approach to the problem of vertical two-phase pipe flow was given by Ros, 948 who utilized a dimensional analysis. He considered twelve independent

variables which account for geometry, liquid and gas physical properties, flow properties, and interactions between phases. Ros used pressure gradient and liquid holdup as dependent variables, and he arrived at the following dimensionless groups:*

Diameter influence number	$N_d = d \sqrt{\rho_{\ell} g/\sigma}$
Relative roughness	€/D
Pipe inclination	Ø
Gas-liquid density ratio	$N_{\rho} = \rho_{g}/\rho_{\ell}$
Liquid viscosity-influence number	$N_{\ell} = u_{\ell} \sqrt[4]{g/\rho_{\ell}\sigma^3}$
Gas viscosity-influence number	$N_g = u_g \sqrt[4]{g/\rho_{\ell} \sigma^3}$
Liquid velocity-influence number	$N = V_{s} \ell^{4} \sqrt{\rho_{\ell}/g\sigma}$
Gas-liquid velocity ratio	$R = V_{sg}/V_{s\ell}$
Wall contact angle	θ
Dimensionless-pressure gradient (dependent)	$G = (1/\rho_{\ell}g)(dP/dL)$
${ m V_{s}}_{m \ell}$ and ${ m V_{sg}}$ are superficial velocities.	

By assumptions, Ros eliminated certain groups, and his experimental work was comprised of 4000 data runs which yielded 20,000 experimental points. His correlations for frictional-pressure gradient and liquid holdup involve a rather large number of constants which are related to the dimensionless groups. The prediction of pressure drop and liquid holdup by this method gives strong consideration to three flow regimes: liquid phase continuous, gas phase continuous, and alternating phases. The method gives impressive accuracy, the standard deviation between measured and predicted values in the three regimes being 3, 10, and 8%, respectively.

Ros's treatment is most significant in that he has used dimensionless parameters involving fluid physical properties, and that predictions of pressure drop depends on nature of the flow.

In 1948 Martinelli and Nelson 720 proposed a procedure for calculating pressure drop during forced-circulation of boiling water. The correlation is based on few data, but it represents one of the few attempts to estimate two-

^{*}Dimensionless groups given here are not to be confused with nomenclature in Appendix A.

phase pressure drop in situations where quality varies with flow length. The ϕ and χ values from the previous correlations, 670 , 721 derived from air-water data, were assumed valid for boiling water. The ϕ 's were corrected in order to have proper empirical dependence on pressure, and working charts are given which can be used (with caution) in determining frictional and momentum pressure drops for flow of boiling water.

Soviet investigators have been interested in two-phase flow in boiling systems. Armand 35 correlated the ratio of two-phase pressure gradient to the liquid-pressure gradient as a function of volumetric steam content. He considered the ratio of volumetric steam content to fraction of pipe cross-section occupied by steam as a parameter. Bankoff demonstrated the relationship between this parameter, the volume fraction, and the slip ratio. This relationship when combined with the void fraction and density ratio yields the quality which then allows prediction of pressure drop.

Two-phase pressure drop data for metallic systems are not available in the literature. In an AEC report 146 the authors derive a pressure drop equation in which they account for hydrostatic, friction, and acceleration losses. For friction losses they use the Lockhart-Martinelli-multiplier, modified for the mercury system at saturation temperatures. No data are given.

Kutateladze et al.617 report the results of Lozhkin, Krol, and Gremilov, who studied two-phase mercury flow. They report that wetting has negligible effect on two-phase mercury flow systems, and they propose the following equation for pressure drop.

$$\Delta P = \frac{f u \rho_{\ell} L}{2g_{c} d} \left[1 + \left(1 - \frac{\rho_{V}}{\rho_{\ell}} \right) \frac{V}{V_{\ell}} \right]$$
 (58)

No supporting data are given.

REMARKS ON TWO-PHASE METALLIC FLOW

Because the literature gives no information on two-phase flow behavior of metallic media, investigators and designers are compelled to extrapolate existing correlations (derived almost exclusively from air-water and steam-water data) for problems in metal flow. The reliability of such extrapolations has yet to be established.

Parameters involving physical properties will probably characterize flow regimes and also pressure-drop behavior. Ros's work in vertical two-phase flow is a clear illustration of the importance of physical properties. Experimentally, it would be desirable to approximate two-phase metallic flow by use of a more easily handled aqueous system. The physical properties of the steam-water sys-

tem have been compared with those for the sodium and potassium systems on a basis of reduced temperature. For the density and viscosity of sodium vapor and water vapor, the properties are of the same order of magnitude—indeed, nearly equal—over a T_r range of 0.5 to 0.7. Liquid phase densities and viscosities also show an encouraging agreement over the same reduced temperature range. The meager amount of data for potassium also shows a favorable comparison with water-steam properties, although the applicable T_r range is not yet adequately known. Surface tensions for these three substances are of the same order of magnitude.

The above-mentioned correspondence in physical properties between water and two alkali metals, although preliminary, indicates that extrapolation of water-steam pressure drop methods to sodium and potassium systems may be valid. The Martinelli-Nelson method for forced-circulation boiling pressure drop has been used for sodium calculations on a reduced-pressure basis. The results cannot be substantiated because of lack of data, but using the method on a reduced-property basis is believed to give the best predictions currently possible.

There is disagreement in the literature as to whether a significant relationship exists between two-phase pressure drop and flow regimes. Recent investigations indicate that pressure drop depends on flow pattern, but this area needs further work. Data definitely are needed for metallic systems.

Two-phase flow data are sparse for forced-circulation boiling, and none is available presently for metallic systems. Work is being conducted in this area at the Argonne National Laboratory. Lunde 691 cites an instance where pressuredrop data provided the best basis for a quantitative estimation of heat transfer to liquids in an atomized state. Thus, the ability to accurately predict two-phase flow behavior should be a decided help in designing boiling heat-exchange systems.







NOMENCLATURE

а	Acceleration (Lt ⁻²)*
A	Area (L^2) ; parameter defined in Eq. (35)
Аα-β	Area of interface between phases α - β (L ²)
В	Parameter defined in Eq. (35)
\mathtt{B}_{L}	Parameter defined by Eq. (3)
c, C	Constant
Ср	Heat capacity (L ² t ⁻² 9 ⁻¹)
C _{vd}	Volumetric gas content (dimensionless fraction)
d, D	Diameter (L)
f	Coefficient of resistance (dimensionless)
f ^α , f ^β	Helmholtz free energy per volume for phase α and β , respectively (mt-2L-1)
$\mathtt{f}^{\mathtt{S}}$	Helmholtz specific free energy (L2t-2)
Fr (or \mathbb{N}_{F})	Froude number (dimensionless)
$_{ m F}^{ m T}$	Total Helmholtz free energy (mL2t-2)
g	Acceleration of gravity (Lt ⁻²)
g _c	Gravitational conversion constant (32.17 ft/sec ²)
$g_{\mu \nu}$	Surface-stress tensor (mt ⁻² L ⁻¹)
G	Mass flowrate (mL ⁻² t ⁻¹); dimensionless pressure gradient
Gr	Grashof number, $L^3g\beta\Delta T/v^2$ (dimensionless)
h	Heat-transfer coefficient ($mt^{-2}\theta^{-1}$)

^{*}Dimensions are given in the following system:

m = mass, L = length, t = time, 0 = temperature.

h _{co}	Local convection heat-transfer coefficient based on conduction (mt $^{-2}\theta^{-1}$)
h _r	Local convection heat-transfer coefficient based on radiation (mt $^{-2}\Theta^{-1}$)
h	Enthalpy (mL ² t ⁻²)
Н	Planck's constant (6.624 x 10 ⁻²⁷ erg·sec)
J	Defined by Eq. (20)
k	Thermal conductivity (mLt ⁻³ 0 ⁻¹)
K	Boltzmann's constant (1.38 x 10^{-16} erg deg ⁻¹); constant defined by Eq. (11)
L	Length (L); liquid mass flow rate (mL ⁻² t ⁻¹)
L_{O}	Critical height of viscous-flow section of heat source (L)
m	Constant defined in Eq. (9)
n	Constant defined in Eq. (9)
N_A	Avogardro's number $(6.023 \times 10^{23} \text{ molecules/mole})$
Nu	Nusselt number, hL/k (dimensionless
$ exttt{N}_{ exttt{i}}^{ exttt{T}}$	Total moles of component i
$_{\mathrm{N_{i}}}^{\alpha}$, $_{\mathrm{N_{i}}}^{\beta}$	Moles of component i in α and β phases, respectively
p, P	Pressure (mL ⁻¹ t ⁻²)
Δρ	Pressure drop (mL ⁻¹ t ⁻²)
ΔP/ΔL, dP/dL	Pressure gradient and local pressure gradient, respectively $(mL^{-2}t^{-2})$
Pr	Prandtl number, Cpµ/k (dimensionless)
q, q/A	Heat-flux density (mt ⁻³)
r	Radius (L)
Re	Reynolds number, $dv\rho/\mu$ (dimensionless)

```
Vapor-film Reynolds number (dimensionless)
Re*
                            Temperature (⊖)
\mathbf{T}
                            Temperature difference (9)
\Delta T
                            Temperature difference between saturated vapor and bulk
ΔT<sub>sub</sub>
                             liquid temperature (\Theta)
                            Velocity (Lt<sup>-1</sup>)
v^{\alpha}, v^{\beta}
                             Volume of phase \alpha and \beta, respectively (L<sup>3</sup>)
                             Mean mixture velocity (Lt<sup>-1</sup>)
V_{\mathsf{m}}
                             Superficial gas velocity (Lt-1)
V_{sg}
                             Superficial liquid velocity (Lt-1)
V<sub>S</sub>1
                             Vapor quality (fractional, dimensionless); (\Delta P/\Delta L)_{\ell}/
Χ
                             (\Delta P/\Delta L)_g
                             Critical vapor film thickness (L)
у*
                             Thermal diffusivity, k/Cp \rho (L^2t^{-1})
α
                             Constant; volumetric coefficient of expansion (\Theta^{-1})
β
                             Defined in Eq. (C-3)
 Γ
                             Boundary-layer thickness (L)
 δ
                             Denotes unit matrix
 δμν
                             Emissivity, strain (dimensionless)
 \in
                             Contact angle (dimensionless)
                             Latent heat (Lt<sup>-2</sup>)
 λ
                             Latent heat using arithmetic mean vapor conditions (Lt<sup>-2</sup>)
 \lambda^{1}
                             Defined by Eq. (17)
 \lambda_{O}
 λβ
                             Defined by Eq. (54)
                             Viscosity (mL<sup>-1</sup>t<sup>-1</sup>)
 μ
```

```
Chemical potential of component i (L^2t^{-2})
\mu_{	exttt{i}}
                           Kinematic viscosity (L<sup>2</sup>t<sup>-1</sup>)
                           3.1416 (dimensionless)
π
                           Density (mL<sup>-3</sup>)
                           Surface tension (mt<sup>-2</sup>)
                           Stefan-Boltzmann constant (5.672 \times 10^{-5} \text{ erg cm}^{-2} \text{deg}^{-4} \text{sec}^{-1})
σ'
                           Defined by Eq. (16)
Τ
Ø
                           Constant defined by Eq. (26); pipe inclination angle (di-
                           mensionless); Martinelli two-phase flow correlation parame-
                           ter [see Eqs. (56) and (57)]
\chi or \chi_{\rm tt}
                           Martinelli's two-phase flow modulus
ψB
                           Defined by Eq. (55)
                           Defined by Eq. (34)
\gamma_{\rm c}
Subscripts
1, 2
                           Denotes condition
                           Bulk
Ъ
                           Critical; horizontal cylinder
С
                           Equivalent
е
fg
                           Change from; liquid to gas
                           Gas
g
                           Liquid
l
```

m

s

V

μ

TPF

sub

Mean

Vapor

Wall 1,2,3

1,2

subcooled

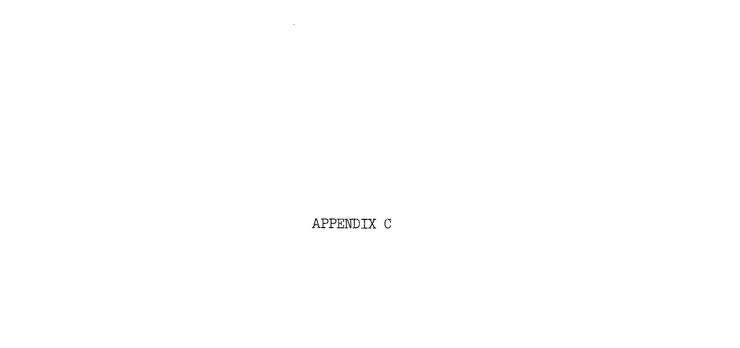
Saturated; solid

Two-phase frictional



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SUPPLEMENTARY DISCUSSION OF INTERFACE CONSIDERATIONS

Interfaces, whether liquid-vapor, solid-liquid, or solid-vapor are inherently very difficult to reproduce. Therefore, a major problem is encountered in the interpretation of experimental data where surface considerations are important. These difficulties often cause seemingly contradictory statements to be made concerning the effects of surface conditions on experimental results. The cause of the difficulties can be appreciated if the details of an interface are examined.

The simplest type of surface is that between a liquid and its vapor. Such a surface is very nearly smooth except when examined on the scale of atomic dimensions. Its energy and state-of-stress can be characterized by a single parameter dependent only on temperature, pressure, and composition of the liquid phase. This parameter, called the "surface tension" to be defined more specifically, can be directly measured. Interfaces involving a solid phase in contact with either a liquid or a vapor are by no means as simple. The geometrical surface is, even after very careful preparation, quite rough. The finest surface finishes on solids still give peak to valley roughness of from 2 to 5 microinches. In addition the solid is in general not homogeneous; that is, it will consist of grains each having different properties and property variations in different directions. In metals the very high affinity between the solid and ever present contaminants causes some degree of surface contamination. This contamination ranges from very lightly held, physically absorbed molecules to thin oxide layers. For most metals of engineering importance, the oxygen pressure necessary to avoid some form of oxygen contamination is far lower than the best obtainable vacuum. Thus, even with carefully cleaned surfaces the interface is generally covered with an oxygen-rich layer, on top of which is found a more weakly adsorbed stratum of other polar molecules.

The energies associated with metallic interfaces are in general much larger than those found for other types of materials such as organics and aqueous base solutions. The surface tensions of liquid metals range from several hundred to several thousand dynes/cm as compared to water with about 70 dynes/cm. The higher values of interfacial energy give rise to several problems since these energies can most easily be lowered by absorbing small amounts of a variety of elements present in the environment. This lowering of energy can take place rapidly or over a long period. It is often possible to replace one contaminant layer with another. The replacement may be accomplished by dissolution (atomby-atom removal) or, in some cases by a tunneling of a liquid phase under a superficial oxide layer.

The above behavior has been summarized by Bikerman and is an excellent review of the technical literature.

Before discussing the specific effects of surface parameters on the boiling process, a short review of surface thermodynamics is in order. Much of the literature on boiling makes use of thermodynamic concepts used in situations where they need not apply. This is particularly the case for the so-called "contact angle." Most standard treatments of surface thermodynamics are evolved in terms of the "surface tension." Such treatments are quite adequate for liquid-vapor or liquid-liquid interfaces, but entirely inappropriate for interfaces involving solids. A rather complete discussion of this point is given by Herring. 458 In the case of interfaces involving a solid there are three distinct quantities that should be differentiated. The first is the Helmholtz specific free energy. It is defined in Eq. (C-1).

$$f^{s} = \frac{F^{T} - V^{\alpha}f^{\alpha} - V^{\beta}f^{\beta}}{A_{\alpha-\beta}}$$
 (C-1)

where F^T is the total Helmholtz free energy of the system comprised of phases α and β , and V^{α} , V^{β} are the volumes of the respective phases; f^{α} , f^{β} are the Helmholtz free energies per unit volume; and $A_{\alpha-\beta}$ is the area of the interface between the two phases.

The second quantity which is called, somewhat reluctantly, "surface tension" is defined in Eq. (C-2).

$$\sigma = f_s - \sum_{i} \Gamma_{i} \mu_{i}$$
 (C-2)

where μ_i is the chemical potential of component i and Γ is the "surface excess" defined by Eq. (C-3).

$$\Gamma_{i} = \frac{N_{i}^{T} - N_{i}^{\alpha} - N_{i}^{\beta}}{A_{\alpha-\beta}}$$
 (C-3)

where $N_{\bf i}^T$ is the total number of moles of component i, and $N_{\bf i}^{\alpha}$, $N_{\bf i}^{\beta}$ are respectively the moles of i in the alpha and beta phases.

The third quantity is the surface-stress tensor $g_{\mu\nu}$, where the individual components are forces per unit length acting at the surface and arising due to the presence of the surface. The surface stress for solids is not equal to the surface tension, as has been shown by Herring⁴⁵⁸ and Shuttleworth.995 The two quantities are related by Eq. (C-4).

$$g_{\mu\nu} = \sigma \delta_{\mu\nu} + \frac{\partial \sigma}{\partial \epsilon_{\mu\nu}}$$
 (C-4)

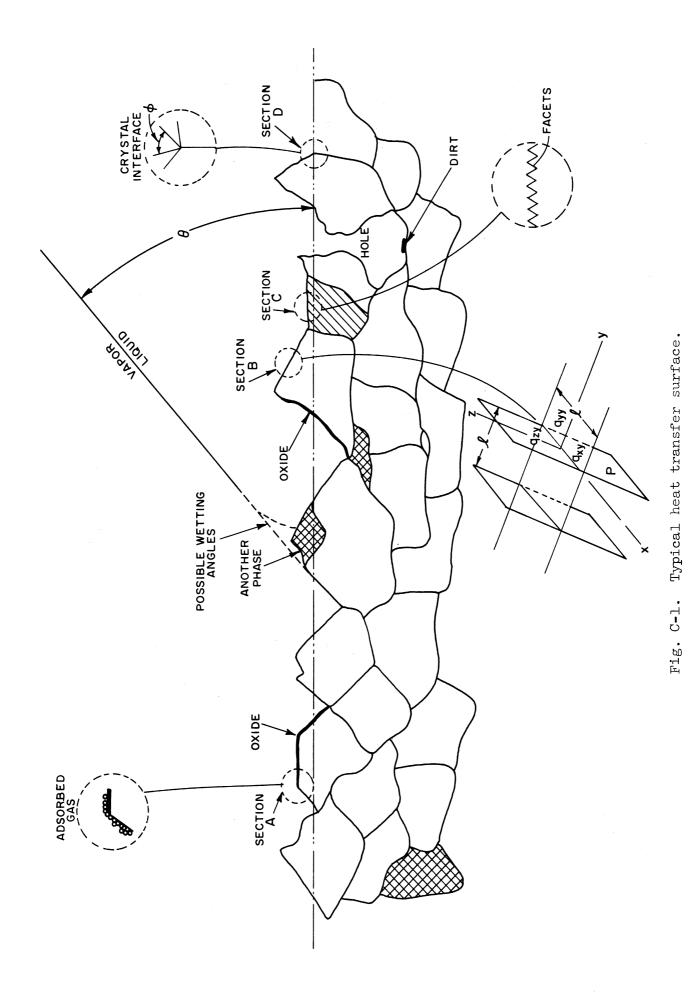
where μ = 1, 2, 3; ν = 1, 2, and $\delta_{\mu\nu}$ is a unit matrix. In general, an interface involving a solid will have a component of surface stress (tension or

compression) acting in the plane of the surface, a shear component acting in the plane of the surface, as well as a component acting normal to the surface. This set of surface forces is illustrated in Fig. C-1. These forces vary in magnitude with direction in an individual grain, and from grain to grain across a metallic surface. For the special case of a liquid-vapor or a liquid-liquid interface, the surface stress tensor can be represented with a single tension component. The second term of Eq. (C-4) is zero, since, upon stretching, the interface extends itself not by altering the relative density of atoms in the surface, but by causing new atoms to come into the surface from the bulk liquid. In such a case the surface stress is indeed numerically equal to the surface tension as defined in Eq. (C-2), and it is quite appropriate to interchange the concepts of force per unit length and free energy per unit area. It is interesting to note that, even in this case, σ is not generally equal to $f^{\rm S}$, the specific surface free energy. The two differ by the right-hand term of Eq. (C-2), which is zero only for one choice in the physical location of the dividing surface.

Recent work 483 has shown that the surface tension and the surface stress are functions of the elastic strain in a solid metal adjoining either a vapor or a liquid interface. When one analyzes the condition of adherence or spreading of a liquid on a polycrystalline solid the conditions on any individual grain are determined by the orientation of the crystallographic axis with respect to the surface area, the orientation of the area with respect to an external coordinate system, and the direction of all of the individual applied or induced strain components existing in the solid. Thus, the degree of macroscopic wetting of the surface is not truly indicative of the local conditions of wetting, which are much more appropriate in any discussion of nucleation.

Macroscopic contact angles are customarily defined in terms of the surface tensions. Such considerations lead to the often quoted equations for contact angle. These relations presume complete thermodynamic equilibrium. In particular, surfaces involved must be capable of migrating freely under the surface forces. In most cases σ is assumed to be independent of crystallographic orientation. Drops or bubbles in contact with solids virtually never come to complete equilibrium, as can be shown from the lack of balance of the vertical components of the "vectors" shown in Fig. C-1.

On the other hand, the equilibrium of surface forces acting at the junction of a mobile phase boundary (e.g., vapor-liquid on solid), is mechanical in nature and does not depend on the establishment of complete thermodynamic equilibrium. Thus, conditions for the movement of the phase boundaries shown in Fig. C-1 and the local contact angle are dependent on the existing state of balance of the sum of the components of the surface-stress tensor on the solid on either side of the liquid-vapor surface. At present there is no known experimental method for measuring directly the components of the surface stress tensor. However, it is possible to measure the change of the value in these components as the crystal is strained. Thus, it is found that a solid surface has a set of elastic moduli closely analogous to the elasticity coefficients for the bulk phase but differing markedly in value.



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Figure C-l has been drawn to indicate schematically the various types of behavior to be found when a vapor and a liquid is in contact with a solid crystalline substance such as a metal. The first feature is physical roughness. The second feature shows the possible presence of film or absorbed surface contaminance. It should be noted that almost all metals contain second phases introduced either from impurities or in many cases as a desirable feature of the metallic structure. Such phases are often present as extremely fine particles. They are often nonmetallic in character such as oxides or sulfides, and exhibit widely different surface characteristics than the parent metallic phase. individual grains comprising the geometric surface will, in general, present a variety of crystallographic orientations to the environment. The grain boundaries between these grains will also have varying energies due to the mismatch differences exhibited from place to place across the surface. Surface damage in the form of deep recesses such as shown in Fig. C-l are also common occurrences. They often contain minute particles of phases of nonmetallic character, i.e., dirt, oxide particles, etc.

For many considerations the features illustrated in Fig. C-1 are of no practical significance. When considering processes such as the nucleation of a new phase, however, the very nature of the process demands that attention be given to the conditions that exist on a very local scale. For example, the critical size required to nucleate bubbles is of the order of 1 micron, which is in general small compared to the grain size of most metals. Thus, the degree of wetting at individual locations on the surface is far more important than the degree of wetting of the gross surface.

The picture of a metallic surface that has been involved above then shows the surface conditions at any given time to be locally determined by the surface properties of the individual grains and minor phases as well as the bulk properties of the individual phases and the liquid or vapor. All these properties are, of course, functions of temperature, state-of-stress, and the usual composition variables. The effect of temperature may be particularly marked because heating of the metallic surface will in general cause changes in all the strain components in the individual grains due to thermal expansion. must then think of the metallic surface as an aggregate of tiny areas displaying varying affinity for the liquid or the vapor, depending on the local magnitude of the components surface-stress tensor. That the various crystalline phases have different affinities for the liquid and the vapor has been demonstrated experimentally. 438 Experiments at The University of Michigan 651 have shown that elastic straining in the solid can also influence the macroscopic constant. The above ideas are quite consistent with the bulk of experimental findings on boiling heat transfer which indicate nucleation occurring at highly selective points with more and more additional points being activated as the temperature is raised. However, only in poorly wetting liquids will vapor trapped in surface cavities account for the nucleation phenomenon. In systems completely wetted by a liquid, such vapor would be excluded from the cavity walls by the intrusion of liquid.

There is another limitation in attempting to characterize the behavior of liquid-solid systems in boiling by use of a macroscopic contact angle; namely, that for many instances no such angle exists. The relationship between the three surface tensions and the macroscopic contact is often given as in Eq. (C-5):

$$\cos \Theta = \frac{\sigma_{s-v} - \sigma_{\ell-s}}{\sigma_{\ell-v}}$$
 (C-5)

However, a contact angle, Θ , exists only for a special range of values of the surface tensions. In the case where,

$$\sigma_{s-v} - \sigma_{\ell-s} \ge \sigma_{\ell-v} \tag{C-6}$$

 Θ is zero and complete spreading occurs over the macroscopic surface. No further information can be obtained about the relative magnitudes of the three interfacial tensions. However, it is quite possible to have two systems, the first having the left side of Eq. (C-6) only slightly larger than $\sigma_{\ell-V}$; and the second having the left-hand member much larger than $\sigma_{\ell-V}$. In the second case, the relative preference of the liquid for the solid as opposed to the liquid for the vapor is much larger and could hardly be expected to behave in a similar fashion with respect to nucleation and bubble growth. In fact, the lowering of $\sigma_{\ell-S}$, that is, increasing the preference of liquid for solid might be expected to influence not only nucleation characteristics at the solid surface, but the transfer of heat across the solid surface to the liquid.

The case of complete spreading of a liquid metal on a solid metal is much more common than with organic or aqueous phases on solid metals. This spreading can in general be achieved by additives or other methods that influence one or all of the surface tensions. Quite commonly an additive to the liquid phase is made which exhibits quite strong bonding tendencies for the solid. Such an additive can decrease $\sigma_{\ell-s}$ without substantially affecting the other two values. liquid metals such a procedure has distinct limitations. There is, indeed, another condition of spreading, that is, the spreading of the liquid metal along the grain boundaries of the solid metal which can result in complete deterioration or catastrophic fracture of the solid. The equilibrium condition for this spreading is that σ_{l-s} be less than twice the grain-boundary energy σ_{bb} . Thus, the liquid-solid surface tension cannot be lowered without limit, without facing the consequences of complete grain-boundary penetration. Of course, such penetration is again a local affair, that is, the grain boundaries with the highest energies are those which fulfill the necessary conditions for a given liquidsolid energy. In this respect, certain heat-treatment steps can be taken in order to insure that the grain boundaries present in the solid are at relatively low energies. Metals that have undergone annealing tend to eliminate most of the high-energy grain boundaries. There are numerous examples of grain-boundary penetration by liquid metals. Among them are lithium on aluminum alloys, mercury on brasses, and bismuth on pure copper.

Another distressing factor associated with fully wetted metallic surfaces is the ability of the liquid to promote catastrophic fracture of the solid at low stress levels. Such embrittlement by liquid metals has been widely studied in recent years. As an example, copper at 650°F is in air a ductile material having a fracture strength exceeding 48,000 psi. When copper at the same temperature is immersed in liquid lead which only partially wets it, the fracture strength drops to 45,000 psi. As bismuth is added to the lead, the fracture strength and ductility drop rapidly. In pure bismuth the fracture strength is approximately 7,000 psi and the ductility is substantially zero. Similar losses of strength are encountered in many other solid-liquid metal combinations. In general the greater is the wetting tendency of the liquid for the solid, the greater influence will be exerted on the fracture strength.





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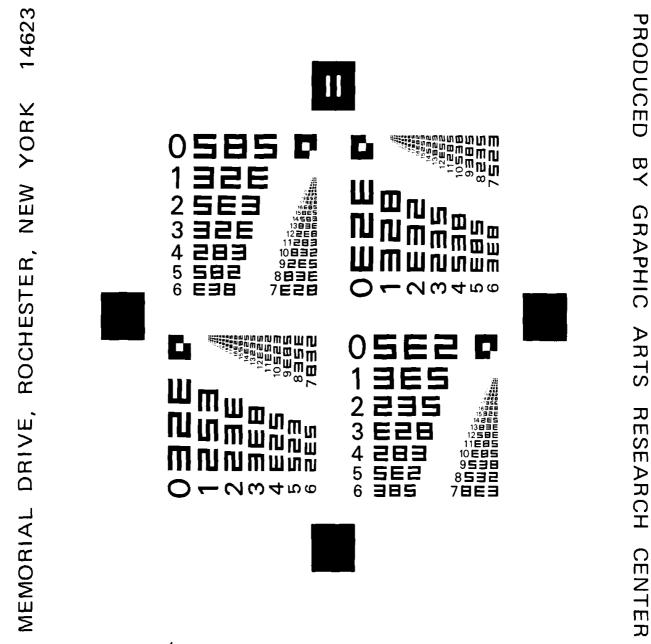
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