SATURATED POOL AND FLOW BOILING STUDIES WITH FREON-113 AND WATER AT ATMOSPHERIC PRESSURE

R. H. Chapman

OAK RIDGE NATIONAL LABORATORY

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AND WATER AT ATMOSPHERIC PRESSURE**

R. H. Chapman

NOVEMBER 1974

Submitted as a dissertation to the Graduate Council
of The University of Tennessee in partial fulfillment of
the requirements for the degree Doctor of Philosophy.

OAK RIDGE NATIONAL LABORATORY
Oak Ridge, Tennessee 37830
operated by
UNION CARBIDE CORPORATION
for the
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ABSTRACT

Research was undertaken to obtain an internally consistent data set from which convective flow boiling could be separated into its constituent pool boiling and forced-convection components and which could be used to test the adequacy of two commonly used correlations for predicting boiling heat transfer. Pool and flow boiling under saturation conditions at atmospheric pressure were studied using Freon-113 and demineralized water as test fluids. The same copper test surfaces, prepared by lapping with a 400-grit compound, were used for all the tests. Horizontal and vertical orientations were studied in the pool boiling facility; whereas the flow test facility was operated in the upflow mode. For the flow boiling tests, the mass flux was varied from 106,000 to 1,325,000 lb/hr·ft² with Freon and from 106,000 to 852,000 lb/hr·ft² with water. Convective boiling with Freon under annular flow conditions was investigated for a thermodynamic quality range of 2 to 8%, corresponding to a range of 4.1 to 0.94 for the Martinelli parameter. Most of the data were characterized as nucleate boiling; relatively few data were obtained under suppressed nucleation conditions.

All the Freon flow boiling data were well correlated by the Rohsenow additive method. The annular flow boiling data were not predicted by the relation proposed by Chen. This was attributed to failure of the Zuber expression, on which Chen's relation is based, to describe the pool boiling results adequately.

Aging effects of the copper boiling surface in the flow boiling tests with water (and to some extent in the horizontal pool boiling tests) caused excessive scatter and poor reproducibility in the data, negating correlation of the results with the Rohsenow additive technique. The data exhibited, however, the same general trends as observed with Freon.

Measured boiling coefficients and photography provided some evidence that steam bubbles in the water stream flowing past the test surface interfere slightly with the boiling performance; this effect was not observed in the Freon tests. Degradation in performance appeared related to bubble size; the steam bubbles were observed qualitatively to be two to three factors greater in diameter than the Freon bubbles.

No detectable surface orientation effect was found in the pool boiling studies with Freon. The results were well correlated by the Rohsenow pool boiling relation; the expression proposed by Zuber was inadequate. Measured coefficients for water boiling on vertical surfaces were found to be about 10% greater than those determined for horizontal surfaces. The results for pool boiling of water were reasonably well correlated with the Rohsenow expression but again the Zuber relation was deficient.

Based on the results of this study, the Rohsenow additive method is recommended for use with Freon-113 over the investigated range of mass flux and thermodynamic quality. Its use for higher mass fluxes and qualities is probably justified. Although the formulation of Chen's correlation facilitates its use, it is not recommended for application with Freon-113 in the mass flux and quality range studied. The results of this study suggest that modifications to improve the nucleate boiling formulation are needed.

Metallographic examination of specimens prepared identically to the boiling test surfaces revealed a highly irregular surface (on a microscopic scale). High magnification micrographs obtained by ordinary optical techniques and by use of an electron scanning microscope are included.

Recommendations for further investigations are given.

LIST OF SYMBOLS

A	area
A_i	coefficient in fitted equations; $i = 1, 2, 3, \dots$
a	empirical constant, Eq. (2-50)
b	exponent
C_p	specific heat
C_{sf}	Rohsenow surface-fluid combination factor
c	exponent
D	channel diameter
D_{eq}	equivalent hydraulic diameter = $4 \times$ flow area/wetted perimeter
$D_{eq,l}$	equivalent diameter for liquid film, Eq. (2-51)
E	voltage potential
E_{err}	thermocouple emf error from calibration
E_{meas}	measured thermocouple emf
F_{DA}	correction factor in Dengler-Addoms correlation
F_T	turbine meter temperature correction factor
F_{TP}	Chen's forced-convection factor
f	frequency
f_d	void fraction defined by Eq. (2-48)
G	mass flux
g	local gravitational acceleration
G_c	gravitational constant
ΔH	enthalpy loss, Eq. (A-10)
h	enthalpy; also heat transfer coefficient

h_{conv}	convective heat transfer coefficient
$h_{\text{conv,mod}}$	modified convective heat transfer coefficient
h_{fg}	latent heat of vaporization
h_{macro}	Chen's modified convective heat transfer coefficient
h_{micro}	Chen's modified nucleate boiling heat transfer coefficient
$h_{\text{nat conv}}$	natural convection heat transfer coefficient
h_{PB}	pool boiling heat transfer coefficient
h_{TP}	two-phase boiling heat transfer coefficient
I	electrical current
j	Joule constant, 778 $\text{lb}_f \cdot \text{ft}/\text{Btu}$
K_J	turbine meter constant, Eq. (A-3)
K_T	turbine meter constant corrected for temperature
k	thermal conductivity
L	channel length
m	exponent
N	number of nucleation sites
$NBCF$	nucleate boiling correction factor, Eq. (2-9)
N_{Bo}	boiling modulus
N_{Fr}	Froude modulus
N_{Nu}	Nusselt modulus
N_{Pr}	Prandtl modulus
N_{Re}	Reynolds modulus
N_{St}	Stanton modulus
n	empirical constant, Eq. (2-50)
P	pressure
P_{HL}	heat meter power loss

ΔP_e	Chen's effective pressure difference, Eq. (2-33)
Q	volumetric flow rate
q	heat flow rate
q''	heat flux
r	exponent in Rohsenow pool boiling correlation
r_b	bubble radius
r^*	stable bubble radius, Eq. (2-9)
S	Chen's nucleate boiling suppression factor
T	temperature
ΔT	temperature difference
ΔT_e	Chen's effective superheat temperature
ΔT_i	temperature difference defined by Eq. (2-4)
ΔT_{sat}	superheat temperature, $\Delta T_{sat} = T_w - T_{sat}$
V	stream velocity
v	specific volume
X	distance measured from boiling surface in heat meter
x	thermodynamic quality, weight fraction vapor
Z	absolute temperature function defined by Eq. (C-9)

Greek Symbols

δ	laminar sublayer thickness; also finite differential
Δ	finite difference
ϵ	Chawla's two-phase flow parameter
μ	molecular viscosity
ν	kinematic viscosity
ρ	density
σ	surface tension

X_{tt} Martinelli parameter defined by Eq. (2-3)
 ϕ function

Subscripts

b bubble; bulk; boiling
conv convection
f film; fluid
g vapor
H horizontal
l liquid
meas measured
NB nucleate boiling
PB pool boiling
sat saturation
TP two-phase
w wall; surface

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CHAPTER I

INTRODUCTION

It has long been recognized that the boiling mode of heat transfer can sustain very high heat fluxes with modest temperature differences. For single-phase, forced-convection heat transfer, the heat flux varies approximately with the first power of the temperature difference; whereas for boiling, the dependence is to about the fourth power. Using heat removal per unit coolant volume as a figure of merit, forced-convection boiling has the greatest potential for application in systems characterized by high volumetric heat generation. Notable examples employing convective boiling as the primary mechanism for heat removal include nuclear reactor cores, rocket engines, and once-through evaporators.

For fully developed nucleate boiling, the temperature difference, which is the driving force for the transfer of heat, is nearly constant over a wide range of conditions. This attribute, coupled with the heat removal capability per unit coolant volume, has led recently to a new area of general application for boiling heat transfer; namely, the cooling of solid-state electronic devices. Proper functioning of these devices is highly dependent on the temperature level, and, in order to fully exploit miniaturization, convective boiling is considered a necessity.

Heat transfer to boiling liquids has been studied extensively, both experimentally and theoretically; and the literature on the subject is overwhelming. Although there appears to be general acceptance of the mechanisms involved with simple pool boiling systems, there remains a diversity of opinion on the manner best suited for correlating the derived

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knowledge of an admittedly complex phenomenon. As one departs from the simple pool boiling system to systems of more interest in industrial applications, the complexities increase and the areas of agreement decrease. This is not unexpected since there are many poorly understood factors that influence the boiling process to varying degrees.

Logic supports the belief of early investigators of boiling heat transfer that an understanding of convective boiling could best be obtained by eliminating the effect of flow and studying the phenomena in stagnant liquids first. This led, literally, to thousands of pool boiling investigations, beginning apparently with Nukiyama's^{1†} classic study in 1904. Based on and/or guided by these studies, investigators then turned their attention to flow-boiling systems. Collier² published an excellent review of the world literature on the subject during the period 1949-1957. Subsequently, in a 1961 review on forced-convection boiling, Zuber and Fried³ concluded that reliable correlations were not then available.

It was recognized early that bubble agitation was a significant factor in producing the high heat transfer coefficients inherent to the boiling process. Recognition of this fact gave impetus to a large number of investigations in which the flow boiling process was simulated by two-phase, gas-liquid systems, both adiabatic and diabatic. A number of reviews and bibliographies have been published in this research area.⁴⁻¹⁰ Many of these studies simulated surface boiling by injection of gas through porous boundaries. Nixon et al.¹¹ used a novel technique for

[†] Superscript numbers refer to similarly numbered References at the end of the thesis.

studying boiling by generating hydrogen bubbles electrolytically on a heated surface.

Significant interactions exist in convective boiling between the liquid-vapor phase velocities and the temperature gradient at the liquid-solid interface; these factors must not be neglected if meaningful information is to be gained from convective boiling investigations. It is also apparent that, while studies based on overall average conditions give gross behavior and provide general guidance, only detailed investigations of local conditions can lead to definitive information. This is, of course, the reason for the many studies undertaken to delineate the mechanisms involved with the dynamics of bubble nucleation, growth, and detachment. In order to make progress these studies have, of necessity, relied on grossly simplified systems, most often employing single bubble generators and/or artificial nucleation sites. While such systems give important information — in fact, if it were not for them, our knowledge of boiling would be seriously limited — the conditions studied are not altogether representative of those existing in industrial applications. There is need for study in more realistic systems and the present research has been directed to that end.

Experimental observations for saturated flow boiling in tubes show that the local heat transfer coefficient increased smoothly along the length of the tube, reached a maximum value, and then decreased sharply. Consistent with the observations of numerous investigators, it is generally accepted that three distinct heat transfer regimes predominate

along the tube in the direction of flow. These are:

1. A "nucleate boiling" region in which the heat transfer mechanism is characteristic of nucleate boiling. Observed coefficients for this region reflect the influence of forced convection by increasing with heat flux and mass flow; however, they do not appear to be strongly influenced by the vapor mass fraction (i.e., bulk stream quality). This mechanism appears to be dominant in the 0-15% quality range.

2. A "nucleate boiling suppressed" region, in which the heat transfer mechanism is characteristic of turbulent forced convection. Observed coefficients in this region exhibit an 0.8 power dependence on mass velocity and an independence of heat flux on temperature difference (wall minus bulk), both distinctive features of forced convection. There appears to be a strong dependence on quality, with the coefficient increasing with increasing quality. This mechanism appears to be dominant in the 15-70% quality range. Annular flow (with a liquid film on the wall and a central vapor core) exists in this region. Evaporation from the liquid film to the vapor core is most likely the dominant heat transfer mechanism.

3. A "liquid deficient" region in which the wall is no longer completely covered by a liquid film. (Liquid may be present as a mist or fog dispersed in the continuous vapor phase.) Observed coefficients in this region are characteristic of forced convection to a vapor. Obviously this regime extends to the limiting case of single-phase (all vapor) heat transfer regime.

Distinct boundaries cannot be identified for each region, since they normally overlap each other as evidenced by the observed smooth variation of the heat transfer coefficient as a function of length (i.e., quality). The research reported herein is concerned only with the "nucleate boiling"

and "suppressed nucleate boiling" regions (i.e., regimes 1 and 2 as discussed above).

Most attempts at correlating turbulent forced-convection boiling heat transfer¹²⁻²⁶ in the regions of interest (saturation conditions) to this research have been based on the addition of two terms, one related to the mechanism of nucleate pool boiling and the other related to the mechanism of turbulent forced-convection in closed conduits. There appears to be general agreement among previous investigators¹³⁻²⁶ that the term related to the forced-convection mechanism can be well correlated as a function of the Martinelli parameter,²⁷ which is strongly dependent on vapor quality. On the other hand, there is less agreement on treatment of the nucleate boiling region. Some investigators^{13-17,24-26} have used modifications of pool boiling correlations, while others¹⁸⁻²³ have used correlations based on the boiling number concept introduced by Davidson et al.²⁸ The boiling number, N_{Bo} , is defined as the ratio of the perpendicular mass flux away from the wall due to boiling to the total axial mass flux. This ratio may be interpreted as a measure of the suppression of nucleate boiling; nucleate boiling is more likely at high values of N_{Bo} .

It thus seems apparent that a more basic investigation is needed to delineate the mechanisms involved in forced-convection boiling of saturated fluids. One gets the impression that most investigations have pursued the empirical correlation of extensive test data. While this in itself is not objectionable (from a practical viewpoint, it does make available information useful to the designer), such an approach is not likely to lead to better correlations until supplemented by an increased

understanding of the interactions of the various mechanisms involved in the convective-boiling process.

The present research was initiated to study bubble enhancement and to elucidate and clarify the mechanisms involved in turbulent, forced-convection boiling of saturated fluids at atmospheric pressure. The approach chosen was to separate the mechanisms involved by a series of experiments (in a given geometry) that examine successively the various modes of heat transfer from forced-convection nonboiling through suppressed nucleate boiling. Although significant progress has been achieved, no claim is made that all the questions have been answered. Indeed, many facets of this complicated problem remain to be studied; because to paraphrase an obscure quotation, "In boiling heat transfer, unlike human codes, de minimis curat lex - the law is concerned with trivia."

CHAPTER II

LITERATURE REVIEW

Introduction

Introduction of high power density systems during the early 1950's created a need to predict with reasonable accuracy the rate of heat transfer in flow boiling systems and provided the impetus for extensive research in boiling heat transfer and two-phase flow. While the effort expended in the subsequent years was great, the results were not altogether satisfactory. Even the basic question of the dominant mechanism of boiling heat transfer, i.e., whether the high heat transfer rates associated with boiling are caused by the agitation and stirring action of bubbles in the laminar sublayer adjacent to the heated surface or whether it is due to latent energy transport, has not been completely settled. The former mechanism is believed to be dominant at low heat flux densities, whereas the latter is most likely important at high flux densities. This situation is less than desirable, since no reliable method exists by which the designer can predict with confidence and reasonable accuracy the boiling heat flux for a given surface temperature. Correlations are available, but their accuracies range from about 25% upwards to, in some situations, hundreds of percent.

In single-phase convective systems, knowledge of the pressure and temperature distributions as well as of the flow regime (laminar or turbulent) is sufficient to analyze system performance. Additionally needed information on the volumetric vapor fraction and the two-phase regime (bubbly, slug, annular, etc.) is required for detailed analysis of

two-phase convective systems. Where nucleate boiling is present in the two-phase system, the nucleating qualities of the heated surface can become of major importance. Obviously, an extraordinary amount of detailed information is needed to make a complete analysis of a typical two-phase system. As a rule this information is unavailable; and as noted the analyst must resort to available correlations, many of which are of questionable validity.

The studies undertaken to gain the necessary understanding of the mechanisms involved and/or to improve the predictive techniques are multitudinous. In addition to the many journals currently reporting boiling heat transfer research, a number of texts²⁹⁻³¹ and bibliographies⁶⁻⁹ devoted to two-phase flow and heat transfer have been published. The reader interested in a more complete discussion and/or extensive bibliographies is referred to these sources; only those references having a direct relationship to the present research will be reviewed.

Forced-Convection Boiling

Collier,² in a critical review of the published literature for the period 1937-1957, concluded that little progress had been made toward an understanding of the processes taking place in progressive vaporization along the length of a heated tube. Numerous researchers had observed that the heat transfer coefficient increased smoothly along the length of heated tubes, reached a maximum value, and decreased rather abruptly. Several researchers had proposed empirical correlations based on their work, but, in general, these were inadequate to predict the results reported by other researchers. A few years later (1960), Zuber and Fried⁷

again reviewed the forced-convection boiling literature and concluded that the situation had not changed significantly in that reliable correlations still were not available.

Dengler and Addoms

Dengler and Addoms^{14,15} were among the first to postulate the existence of different heat transfer mechanisms to explain the observed behavior. In analyzing their forced-convection boiling data, they concluded that vaporization in tubes is governed by the combined effects of forced convection and nucleate boiling and that nucleate boiling is dominant only under conditions of low liquid flow velocities. As the velocity of the liquid-vapor mixture increases (induced by increased vapor production along the length of the tube), nucleation is suppressed; and the forced-convection mechanism becomes the primary mode of heat transfer.

Dengler¹⁴ employed a 20-ft-long, 1-in.-ID copper tube with vertical upflow of water under forced circulation to acquire the data analyzed by Dengler and Addoms.¹⁵ Heating was accomplished with five 3-ft-long steam jackets equally spaced along the length of the copper tube. Local (averaged over the 3-ft section) heat fluxes were obtained by measurement of the condensate collected from each jacket. Provision for pressure drop and void fraction measurements, using a radioactive tracer technique, were made between each steam jacket. Wall temperatures were measured with thermocouples embedded in longitudinal slots along the length of the tube. The pressure range tested was 8 to 40 psia. Saturated fluid entered the test section, with exit qualities ranging up to 100%. Mass flow rates were varied from 240 to 5500 lb/hr, corresponding to inlet velocities of 0.2 to 4.0 ft/sec.

In the range of suppressed nucleation, they were able to correlate 85% of their data within $\pm 20\%$ with the equation:

$$\frac{h_{TP}}{h_{conv}} = 3.5 \left(\frac{1}{x_{tt}} \right)^{0.5} \quad (2-1)$$

They found, for the conditions of their investigation, that nucleation was suppressed for values of $1/x_{tt}$ between 0.25 and 70. In Eq. (2-1), the term h_{conv} was defined as the single-phase (all liquid) forced-convection coefficient as given by the Dittus-Boelter equation:

$$h_{conv} = 0.023 \frac{k_L (DG)^{0.8} \left(\frac{uG}{k} \right)^{0.4}}{D \left(\frac{u_L}{u_L} \right)} \quad (2-2)$$

The physical properties were evaluated at the local saturation temperature. The term x_{tt} is the Martinelli¹² two-phase turbulent flow parameter as given by:

$$x_{tt} = \left(\frac{1-x}{x} \right)^{0.5} \left(\frac{v_L}{v_g} \right)^{0.5} \left(\frac{u_L}{u_g} \right)^{0.1} \quad (2-3)$$

The reciprocal of x_{tt} is a strong function of the vapor weight fraction (steam quality) and also, by means of the viscosity and specific volume terms, provides (according to Dengler¹⁴) a near-quantitative correction for pressure on the two-phase coefficient.

Dengler and Addoms¹⁵ believed nucleate boiling was dominant in their investigation at values of $1/x_{tt}$ less than 0.25. Although they were of the opinion that the effect of nucleate boiling was additive to that of convection, they suggested an empirical correction factor in the form of a multiplier for Eq. (2-1) to account for the presence of nucleate boiling. The choice of a multiplier instead of an additive correction

was strictly for convenience. They obtained the correction factor by postulating that the temperature difference, ΔT_i , necessary to initiate nucleation was a function of the average stream velocity. From the measured void fraction data, they deduced for ΔT_i :

$$\Delta T_i = 10 (V_{ave})^{0.3} \quad (2-4)$$

A qualitative pressure dependent factor was also included in the final form of the correction; thus:

$$P_{DA} = 0.67 \left[(\Delta T - \Delta T_i) \left(\frac{dP}{dT} \right)_{sat} \cdot \frac{D}{c L_w} \right]^{0.3} \quad (2-5)$$

where $\frac{dP}{dT}|_{sat}$ and c are evaluated at the wall temperature. Thus, the final form of the general correlation is given by:

$$\frac{h_{TP}}{h_{conv}} = 3.5 \left(\frac{1}{x_{tt}} \right)^{0.5} \cdot P_{DA} \quad (2-6)$$

with P_{DA} being used only when it exceeds unity.

While Dengler and Addoms, using Eq. (2-6), correlated all their data for the nucleate boiling region (18 data points $\pm 25\%$ and two points $\pm 50\%$), the physical significance of their correction term is not readily apparent. No attempt was made to correlate their data obtained for dry-wall or liquid deficient conditions. It was noted that dry-wall conditions, characterized by a sudden and sharp reduction in heat transfer coefficient with increasing vaporization, were encountered at values of $1/x_{tt}$ as low as 25 ($x \approx 0.4$) with general occurrence at values of $1/x_{tt}$ around 30 ($x \approx 0.33$).

Guerrieri and Talty

Guerrieri and Talty¹⁶ investigated boiling with a number of organic liquids under natural-convection conditions in single, vertical tube boilers. The test fluids were cyclohexane, methanol, benzene, pentane, and heptane. Brass tubes, 0.75-in. ID x 6.0-ft long and 1.00-in. ID x 6.5-ft long, were heated externally by oil flowing through a concentric jacket. Thermocouples were used to measure the tube wall temperature at 6-in. intervals. Subcooled liquid entered the test section and exited with qualities in the range of 2.8 to 16%. The range of mass fluxes tested was 28.8×10^4 to 35.8×10^4 lb/hr·ft².

In the region of suppressed nucleation, Guerrieri and Talty represented their data by the relation:

$$\frac{h_{TP}}{h_{conv,mod}} = 3.4 \left(\frac{1}{x_{tt}} \right)^{0.45} \quad (2-7)$$

This agrees fairly closely with the Dengler-Addoms correlation, Eq.

(2-1). The term $h_{conv,mod}$ is a modified Dittus-Boelter forced-convection correlation:

$$h_{conv,mod} = 0.023 \frac{k_l}{D} \left(\frac{DG(1-x)}{\mu_l} \right)^{0.8} \left(\frac{\mu C_p}{k_l} \right)^{0.4} \quad (2-8)$$

Comparing Eqs. (2-8) and (2-2), the modification consists only of basing the coefficient on the liquid fraction of the stream. The lower limit of $1/x_{tt}$ in Eq. (2-7) was stated by the authors to be about 0.3.

To account for bubble agitation in the region of nucleation, Guerrieri and Talty deduced from their data a nucleate boiling correction factor (designated NBCF) involving the ratio of the radius of the minimum size of a thermodynamically stable bubble to the thickness of

the laminar sublayer. This correction factor, used as a multiplier for Eq. (2-8), was expressed as:

$$NBCF = 0.187 (r^*/\delta)^{-5/9} , \quad (2-9)$$

and is to be applied only if it exceeds unity. Thus, Guerrieri and Talty's general correlation becomes:

$$\frac{h_{TF}}{h_{conv,mod}} = 3.4 \left(\frac{1}{x_{tt}} \right)^{0.45} \cdot NBCF , \quad (2-10)$$

again not too different from that proposed by Dengler and Addoms.

Equation (2-10) successfully correlated 88% of Guerrieri and Talty's data to within $\pm 20\%$.

The restriction on Eq. (2-10) that $NBCF \geq 1.0$ implies that $r^*/\delta \geq 0.049$. This is interpreted physically to mean that nucleation is suppressed when the radius of the minimum thermodynamically stable bubble exceeds about 5% of the laminar sublayer.

Schrock, Grossman, et al.

Schrock and Grossman,¹⁸ in the first of a series of two-phase boiling research investigations at the University of California at Berkeley, studied bulk boiling of upward flowing water under forced circulation in electrically heated round tubes having diameters of 0.1162 to 0.4317 in. and heated lengths of 15 to 40 in. Inlet pressures ranged from 42 to 505 psia, and exit qualities varied from 5 to 57%. Mass fluxes were varied over the range of 1.76×10^5 to 3.28×10^6 lb/hr·ft².

These authors postulated for the range of variables tested in their investigation that the local forced-convection boiling heat transfer

coefficient (defined in terms of $\Delta T = T_w - T_{sat}$) could be expressed as:

$$N_{Nu} = \phi (N_{Re}, N_{Pr}, N_{Bo}, x_{tt}) , \quad (2-11)$$

where the boiling modulus, as used by ~~Mumma~~¹⁹ is given as:

$$N_{Bo} = q''/G h_{fg} , \quad (2-12)$$

and the other dimensionless moduli have their usual definitions. The boiling modulus can be interpreted physically as the ratio of the mass flux directed away from (perpendicular to) the wall due to boiling to the total mass flux directed along the axis of the tube. They were able to correlate their full range of data to within $\pm 35\%$ by the expression:

$$\frac{N_{Nu}}{N_{Re}^{0.8} N_{Pr}^{1/3}} = 170 (N_{Bo} + 1.5 \times 10^{-4} x_{tt}^{-2/3}) , \quad (2-13)$$

where the dimensionless moduli were evaluated using saturated liquid properties at the local pressure. Since the correlating equation is expressed in terms of dimensionless moduli, the authors were of the opinion that it would apply to liquids other than water. The correlation was recommended for the quality range of 0 to 50%; that is, for both the nucleate boiling region and the suppressed nucleation region.

Equation (2-13) can be rearranged to give:

$$\frac{h_{TP}}{h_{conv}} = 0.739 [N_{Bo} \times 10^4 + 1.5 (1/x_{tt})^{2/3}] , \quad (2-14)$$

with the minor modification that in Eq. (2-2) h_{conv} is written in terms of N_{Pr} to the one-third power rather than the usual 0.4 power. Written in this form a structural comparison of Schrock and Grossman's equation is conveniently made with those proposed by Dengler and Addoms [Eq. (2.6)] and by Guerrieri and Talty [Eq. (2-10)].

Examination of Eq. (2-14) reveals that for large heat flux to mass ratios (i.e., large values of N_{Bo}) nucleate boiling predominates; and the heat transfer becomes virtually independent of the Martinelli parameter, x_{tt} , the only parameter in the equation containing the steam quality. On the other hand, for low values of N_{Bo} , the convective mechanisms as reflected by the Martinelli parameter predominates.

Sani,²⁰ using Schrock and Grossman's experimental facility, investigated downward flow of water under forced-circulation conditions. An electrically heated stainless steel tube, 0.7194-in. ID x 68-in. long, was employed. Mass fluxes were varied over the range 1.84×10^5 to 5.94×10^5 lb/hr·ft², while the pressure ranged between 15.8 and 30.9 psia. Exit qualities up to 14% were obtained.

Sani correlated 95% of his data to within $\pm 15\%$ by an equation of the Schrock and Grossman form:

$$\frac{N_{Nu}}{N_{Re}^{0.8} N_{Pr}^{1/3}} = 340 (N_{Bo} + 1.5 \times 10^{-4} x_{tt}^{-2/3}) \quad (2-15)$$

This is identical with Eq. (2-13) except that the numerical coefficient in Sani's equation is twice that found to give the best fit to Schrock and Grossman's data. Sani speculated that the difference in coefficients could not be wholly attributed to the flow patterns (i.e., upflow versus downflow) and that other factors (possibly pressure) not specifically included in the correlation had an effect.

In a third investigation performed with the same test facility at the Berkeley campus, Wright²¹ reexamined forced-convection boiling of water in electrically heated, stainless steel tubes in the downflow mode. The tubes were 0.719- and 0.472-in. ID, with heated lengths of 5.67 and

4.69 %, respectively. The mass fluxes were varied over the range 3.96×10^5 to 2.52×10^6 lb/hr·ft², with exit qualities of 0 to 19%. Pressure was varied from 15.8 to 68.2 psia, while the boiling modulus ranged from 0.24×10^{-4} to 1.9×10^{-4} .

Wright correlated his data in a number of ways, with average errors ranging from 10.8 to 16.1%. One correlating equation was:

$$\frac{N_{Nu}}{N_{Re}^{0.8} N_{Pr}^{1/3}} = 320 [N_{Bo} + 1.5 \times 10^{-4} x_{tt}^{-2/3}] \quad (2-16)$$

This equation is essentially the same as that obtained by Sani [Eq. (2-15)] and, hence, yields (for a given N_{Re} and N_{Pr}) a N_{Nu} almost twice that obtained by Schrock and Grossman. Wright attributed most of the difference to pressure effects. He recast the boiling modulus in terms of volumetric ratios, which indeed have a large pressure dependence. Wright's modified boiling modulus is simply the modulus as defined by Eq. (2-12) multiplied by ρ_L/ρ_g or, in mathematical form,

$$N_{Bo, mod} = N_{Bo} (\rho_L/\rho_g) \quad (2-17)$$

Wright used this parameter to correlate his data with the least error ($\pm 10.8\%$); thus:

$$N_{St} = 0.003377 [N_{Re}^{0.106} N_{Pr}^{0.4} N_{Bo, mod}^{0.296} x_{tt}^{-0.467}] \quad (2-18)$$

Using the usual boiling modulus in a similar type correlation, he obtained:

$$N_{St} = 1.7310 [N_{Re}^{-0.258} N_{Pr}^{0.4} N_{Bo}^{0.188} x^{0.382}] \quad (2-19)$$

Although this equation correlated his data almost as well ($\pm 13.4\%$) as Eq. (2-18), the dependence on Reynolds modulus was opposite to that in

Eq. (2-15). [Note that the reciprocal of x_{tt} is a strong function of x .]

The fourth investigation performed with the same facility was that of Somerville²² for downward forced-convection boiling of n-butanol. Electrically heated, stainless steel tubes (0.467-in. ID) with heated lengths of 5.69 ft and 4.10 ft were used. Mass fluxes varied from 4.90×10^5 to 1.58×10^6 lb/hr·ft², with exit qualities up to 31%. The range of pressure investigated was 16.9 to 50.0 psia.

Somerville correlated his data in several ways. Thus, his correlation in the form suggested by Dengler and Addoms was:

$$\frac{h_{TP}}{h_{conv}} = 7.75 \left(\frac{1}{x_{tt}} \right)^{0.322} \quad (2-20)$$

and in the Schrock and Grossman form:

$$\frac{N_{Nu}}{N_{Re}^{0.902} N_{Pr}^{1/3}} = 563 [N_{Bo} + 1.5 \times 10^{-4} x_{tt}^{-2/3}] \quad (2-21)$$

The leading coefficient of Somerville's equation is almost a factor of 4 greater than that for Schrock and Grossman. This tends to contradict Schrock and Grossman's belief that their correlation, being based on dimensionless moduli, would also apply to organic fluids.

Since Somerville had used the same equipment as utilized by Wright, he selected an equal number of Wright's water data points and deduced a correlation for the two data sets:

$$N_{St} = 0.2005 [N_{Re}^{-0.296} N_{Pr}^{-0.233} N_{Bo}^{0.131} x_{tt}^{-0.292}] \quad (2-22)$$

This correlation was then shown to predict the data of a number of other investigators within reasonable limits (no error band was given).

Bennett, Collier, et al.

Bennett et al.¹³ performed an extensive series of two-phase boiling heat transfer experiments with upward flow of steam-water mixtures at essentially atmospheric pressure. An electrically heated stainless steel tube located concentrically within a glass tube formed an annular test geometry that was 29 in. in length. Two different annuli were used: one employed a heater of 0.375-in.-diam inside a 0.51-in.-ID glass tube; and the other, a heater of 0.623-in.-diam inside a 0.866-in.-ID glass tube. The range of mass fluxes tested was 1.12×10^5 to 2.17×10^5 lb/hr·ft². Exit steam qualities ranged from 0 to 100%, depending on flow rate and power level.

These investigators found that below 15% quality the primary mechanism of heat transfer was highly characteristic of nucleate boiling and could be correlated by the Rohsenow³² pool boiling equation:

$$\frac{q_{p,i}^{T_{sat}}}{h_{f,i}} = C_{sf} \left(\frac{q}{h_{f,i} \sqrt{g(\rho_i - \rho_g)}} \right)^r \left(\frac{C_{p,i}}{k_i} \right)^{1.7} \quad (2-23)$$

The surface-fluid coefficient, C_{sf} , and the bubble Reynolds modulus exponent, r , were 1.013 and 0.33, respectively. [The currently recommended^{33,34} value of the exponent on the Prandtl modulus for water is 1.0, and remains 1.7 for other fluids.]

At higher steam qualities (up to 55 to 65%), they found the mechanism characteristic of forced convection (i.e., the region of suppressed nucleation) and were able to correlate their data with a slight modification to the type of equation proposed by Guerrieri and Tait¹⁶. Bennett et al.'s correlating equation is given in dimensional form by

$$\frac{h_{TP}}{h_{conv,mod}} = 0.64 \left(\frac{1}{x_{tt}} \right)^{2.7} (q'')^{2.11} , \quad (2-24)$$

where q'' is in Btu/hr·ft² and $h_{conv,mod}$ is given by Eq. (2-8).

For qualities above 65%, the rate of heat transfer decreased and in the limit approached the value set by the dry steam coefficient.

Collier and Pulling¹⁷ used the same experimental equipment to extend the work of Bennett et al.¹³ to higher pressures and mass fluxes. Heated test sections of 48-in. length were also included in this series of experiments. The range of test pressures was 15.5 to 80.3 psia, with mass fluxes in the range of 2.87×10^4 to 7.94×10^5 lb/hr·ft². Exit quality ranged as high as 65.2%.

These authors confirmed the existence of the three regions of heat transfer found by the previous researchers. Further they were able to eliminate the slight heat flux dependence reported by Bennett et al. for the suppressed nucleation region. Some 1542 data points, including the 542 data points reported by Bennett et al.¹³ in the suppressed nucleation region were correlated by the expression:

$$\frac{h_{TP}}{h_{conv,mod}} = 2.16 \left(\frac{1}{x_{tt}} \right)^{2.639} \quad (2-25)$$

with a standard deviation of 26.9%. In this equation the liquid-only coefficient is given by Eq. (2-8) with the properties of the liquid evaluated at the film temperature, T_f , as defined by:

$$T_f = T_{sat} + 0.33 (T_w - T_{sat}) . \quad (2-26)$$

Their data for the nucleate boiling region were not correlated, although the pattern of the data followed that observed by Bennett et al.

and is discussed above. Similarly, no attempt was made to correlate the data obtained under liquid deficient conditions.

In addition to a thorough discussion of both Bennett et al.'s and their own extensive data, Collier and Pulling¹⁷ gave an excellent review of a number of proposed correlations published prior to 1963. These investigators tested the correlation of Schrock and Grossman on a representative fraction of their data and found that that correlation underpredicted the measured coefficients in the suppressed nucleation region by amounts up to 100% and overpredicted the measured coefficients in the nucleate boiling region by about 25%. Collier and Pulling commented that their data, unlike those of Schrock and Grossman, indicated a sharp transition from the nucleate boiling to the convective or suppressed nucleation region and that an equation in which the two processes are made additive cannot be expected to represent the data accurately in the vicinity of the transition. Since the two mechanisms of heat transfer are quite different in character, they considered it most unlikely that a simple additive-type equation can be made to span the transition region with the desired accuracy.

Collier and Pulling tested their data against the Dukler-Hewitt³⁵ analytical model for annular flow and found that the calculated heat transfer coefficients averaged about 50% (with considerable scatter) above the measured values.

Ten well-defined data points were used to estimate the transition from nucleate boiling to forced convection based on the method proposed by Hsu³⁶ for predicting the onset of nucleation in the boundary layer. Very good agreement was found between calculated and experimental results.

Chen

Chen²³ presented a correlation at the 1963 AIChE-ASME Heat Transfer Conference that has received considerable attention in the succeeding years. [Although Chen's correlation was available at the Conference as ASME Paper 63-HT-34, it was generally unavailable in the literature until published in 1966 as Ref. 24.] The proposed correlation is said to be applicable to saturated, two-phase, convective boiling of non-metallic fluids in the upflow mode. It is assumed that stable flow conditions exist, with the heat flux being less than the critical flux. Slug flow and liquid deficient conditions are excluded. According to Chen,²⁴ these conditions usually occur with annular or annular-mist flow in the quality range of approximately 1 to 70%. In other words, the range of applicability is the nucleate boiling and suppressed nucleation regions observed in the previously discussed investigations.

In developing his correlation, Chen²⁴ relied altogether on the data of Dengler and Addoms,¹⁵ Guerrieri and Talty,¹⁵ Bennett et al.,¹³ Schrock and Grossman,¹⁶ Sani,²⁰ and Wright.²¹ First, a thorough comparison was made of the correlations proposed by these investigators, using 665 reported data points covering the range of conditions tabulated in Table II-1. The results of the comparisons as reported by Chen²⁴ are summarized in Table II-2. He concluded the correlations were unsatisfactory for general use and turned his attention to developing a new correlation, using the above-mentioned data sources. The last column in Table II-2 shows how well his results predicted the various data.

Chen²⁴ postulated that two mechanisms are involved in convective flow boiling of saturated, nonmetallic fluids: the ordinary macro-convection mechanism normally operative with flowing fluids, and the

TABLE II-1

RANGE OF CONDITIONS FOR DATA USED IN TESTING CORRELATION^a

Ref. ^b	Fluid	Geometry	Flow	Pressure, psia	Flow Velocity, ft/sec (Liquid)	Quality, wt %	$q'' \times 10^{-4}$ Btu/hr·ft ²
15	Water	Tube	Up	8-40	0.2-4.8	15-71	2.8-20
18	Water	Tube	Up	42-505	0.8-14.7	3-50	6.5-76
20	Water	Tube	Down	16-31	0.8-2.7	2-14	1.4-5.0
13	Water	Annulus	Up	15-35	0.2-0.9	1-59	3.2-1.6
16	Methanol	Tube	Up	15	1.0-2.5	1-4	0.7-1.7
16	Cyclohexane	Tube	Up	15	1.3-2.8	2-10	0.3-1.3
16	Pentane	Tube	Up	15	0.9-2.2	2-12	0.3-1.2
16	Heptane	Tube	Up	15	1.0-2.4	2-10	0.2-0.9
16	Benzene	Tube	Up	15	1.0-2.4	2-9	0.4-1.3
21	Water	Tube	Down	16-68	1.8-11.2	1-19	1.3-8.8

^aFrom Chen.²⁴^bReference identification changed to agree with numbering system used in this thesis.

TABLE II-2
COMPARISON OF CORRELATIONS^a

Data	Average Percent Deviations for Correlations				
	Dengler and Addoms	Guerrieri and Talty	Bennett et al.	Schrock and Grossman	Chen
Dengler and Addoms (water)	30.5	62.3	20.0	20.3	14.7
Schrock and Grossman (water)	89.5	16.4	24.9	20.0	15.1
Sani (water)	26.9	70.3	26.5	48.6	8.5
Bennett et al. (water)	17.9	61.8	11.9	14.6	10.8
Guerrieri and Talty (methanol)	42.5	9.5	64.8	62.5	11.3
Guerrieri and Talty (cyclohexane)	39.8	11.1	65.9	50.7	13.6
Guerrieri and Talty (benzene)	65.1	8.6	56.4	40.1	6.3
Guerrieri and Talty (heptane)	61.2	12.3	58.0	31.8	11.0
Guerrieri and Talty (pentane)	66.6	9.4	59.2	35.8	11.9
Wright (water)	24.0	75.8	30.4	51.7	15.4
Combined average for all data	36.0	47.5	32.3	34.7	11.6

^aFrom Chen.²⁴

microconvective mechanism normally associated with nucleation and bubble growth. He further postulated that these two mechanisms were additive in their contribution to the total heat transfer, citing the suggestion of Rohsenow¹² that heat transfer by convection and boiling can be added and the success achieved by Gambill³⁷ in correlating subcooled burnout data with an additive-type equation. Chen reasoned that while the two mechanisms would be operative, their normal effects would be modified by the presence of quality and by mutual interactions.

Since the Dittus-Boelter correlation would be applicable in the limits of 0 and 100% quality, it was postulated that in the two-phase region a modified version of this correlation should be used to describe the macroconvection contribution, thus,

$$h_{\text{macro}} = 0.023 (k/D) N_{\text{Re}}^{0.8} N_{\text{Pr}}^{0.4} , \quad (2-27)$$

where the physical properties and the mass flux (N_{Re}) are effective values for the two-phase fluid. For ordinary fluids (excluding liquid metals), the Prandtl moduli of the liquid and vapor phases are normally of the same magnitude; the two-phase Prandtl modulus should therefore also be of the same magnitude. Furthermore, since the heat is transferred through an annular film at the wall, the liquid properties should be dominant. Chen assumed as a first approximation that the effective values could be replaced by the liquid-only values and that the macroconvection contribution could be expressed as:

$$h_{\text{macro}} = 0.023 (k_l/D) N_{\text{Re},l}^{0.8} N_{\text{Pr},l}^{0.4} F_{\text{TP}} , \quad (2-28)$$

where F_{TP} is defined as:

$$F_{\text{TP}} = (N_{\text{Re},\text{TP}}/N_{\text{Re},l})^{0.8} = \left[N_{\text{Re}} \cdot \frac{\mu_l}{DG (1-x)} \right]^{0.8} . \quad (2-29)$$

In principle the only unknown in Eq. (2-23) is F_{TP} ; this ratio is strictly a flow parameter and is related to the Martinelli parameter, X_{tt} .

The analysis of Forster and Zuber³⁸ was chosen as the basis for formulating the microconvection contribution to the total heat transfer. These investigators derived for the case of pool boiling from horizontal surfaces the following equations for the boiling Nusselt modulus:

$$N_{Nu} = 0.0015 N_{Re}^{0.62} N_{Pr}^{0.32} = h r_b / k_l, \quad (2-30)$$

where

$$r_b = \frac{\Delta T}{h_{fg} \rho_g} \left(\frac{2\pi k_l \rho_l C_{p,l} \sigma}{\Delta P} \right)^{0.5} \left(\frac{\rho_l}{\rho_g \Delta P} \right)^{0.25}, \quad (2-31)$$

and

$$N_{Re} = \left(\frac{\pi k_l C_{p,l}}{\mu_l} \right) \left(\frac{\rho_l \Delta T}{\rho_g h_{fg}} \right)^2. \quad (2-32)$$

One of the basic premises of Eq. (2-30) is that the Reynolds modulus for boiling is determined by the bubble growth rate as described by the Rayleigh equation. Forster and Zuber were able to solve the Rayleigh equation to show that the product of the bubble radius and the bubble growth rate is a constant for a specific superheat. Chen reasoned that the superheat across the boundary layer (through which bubble growth and detachment takes place) was not constant and, therefore, that the Reynolds modulus must be representative of some mean, effective superheat. According to Chen, the difference between the wall superheat and the effective superheat is small for pool boiling and could be neglected in the Forster-Zuber analysis. However, in flow boiling this difference cannot be neglected, since the radial temperature gradient is a function

of flow rate and vapor quality and is in general much steeper than for the case of pool boiling with the same degree of wall superheat.

Based on this postulate, Eq. (2-30) was then written by Chen in terms of physical properties and effective values of ΔT_e and ΔP_e as:

$$h_{\text{micro}} = 0.00122 \left(\frac{k_l^{0.79} C_{p,l}^{0.45} \rho_l^{0.49} \mu_l^{0.25}}{\sigma^{0.5} \mu_l^{0.29} h_{fg}^{0.24} \rho_g^{0.24}} \right) \Delta T_e^{0.24} \Delta P_e^{0.75} \quad (2-33)$$

A suppression factor, S , was defined as the 0.99 power of the ratio of the effective superheat to the total superheat of the wall; namely:

$$S = \left(\frac{\Delta T_e}{\Delta T} \right)^{0.99} \quad (2-34)$$

The choice of the power in Eq. (2-34) was arbitrarily taken as 0.99, so that S would appear to the first power in the final equation.

Writing the Clausius-Clapeyron equation in terms of the effective and the wall superheats:

$$\Delta T_e^{0.99} = \left(\frac{T_e}{h_{fg} \rho_g j} \right)^{0.75} \Delta T_e^{0.24} \Delta P_e^{0.75} \quad (2-35a)$$

and

$$\Delta T^{0.99} = \left(\frac{T}{h_{fg} \rho_g j} \right)^{0.75} \Delta T^{0.24} \Delta P^{0.75} \quad (2-35b)$$

For small temperature differences:

$$\left(\frac{T_e}{h_{fg} \rho_g j} \right)_{T_e} = \left(\frac{T}{h_{fg} \rho_g j} \right)_T \quad (2-36)$$

giving:

$$S = \left(\frac{\Delta T_e}{\Delta T} \right)^{0.24} \left(\frac{\Delta P_e}{\Delta P} \right)^{0.75} \quad (2-37)$$

Combining Eqs. (2-37) and (2-38) gives Chen's expression for the micro-convection coefficient in terms of the total superheat and the suppression factor:

$$h_{\text{micro}} = 0.00122 \left(\frac{k_l^{0.79} C_{p,l}^{0.45} \rho_l^{0.49} g_c^{0.25}}{\sigma^{0.5} u_l^{0.29} h_{fg}^{0.24} \rho_g^{0.24}} \right) \Delta T^{0.24} \Delta T^{0.75} S \quad (2-39)$$

Finally, the local two-phase convective boiling heat transfer coefficient is given by:

$$h_{\text{TP}} = h_{\text{macro}} + h_{\text{micro}} \quad (2-40)$$

where h_{macro} and h_{micro} are given by Eqs. (2-28) and (2-38), respectively.

In principle the only unknowns in the above equations are the convective enhancement factor, F_{TP} , and the nucleate boiling suppression factor, S . These were determined empirically by Chen from the data sources mentioned above by an iterative procedure and are presented in Figs. II-1 and II-2. The shaded area in the figures represent the data scatter around the two correlating functions. The convective enhancement factor, F_{TP} , increases smoothly with increasing values of the reciprocal Martinelli parameter, indicating the effect of increased turbulence with increasing vapor fraction. The nucleation suppression factor, S , tends to unity at low values of the two-phase Reynolds modulus and to zero at high values of the modulus. According to Chen, these trends appear reasonable in that at low flows and low vapor fraction the boiling contribution is relatively important; whereas at high flows and high vapor fractions, nucleation is suppressed and the convective mechanism becomes increasingly important. These trends are also consistent with experimental observations.

Fig. II-1. Chen's flow enhancement factor.

Fig. II-. . Chen's nucleation suppression factor.

As indicated in Table II-2, Chen's correlation shows a definite improvement over those proposed by the other investigators. Based on this success, standard references²⁹⁻³¹ on two-phase flow and boiling heat transfer recommend the use of Chen's correlation for convective boiling situations. More recently (1972), Tong³⁹ recommended the correlation for application to the thermal-hydraulic design of nuclear reactor cores.

Hughmark

Although Chen's correlation is accepted as being the best available, Hughmark⁴⁰ was unable to correlate his data with it. In particular, it appeared that a value of $S = 0.25$ better fit his data where nucleation apparently occurred. This seems to imply that S is independent of $N_{Re,TP}$. Hughmark also points out that Chen's microconvection expression neglects the effect of metal surface-fluid combination on nucleate boiling. This factor is known^{33,34,41-46} to be an important variable.

Stone

Stone,⁴⁷ in a thorough investigation of convective boiling of water over a wide range of conditions, tested his extensive data against the correlations of Dengler and Addoms,¹⁵ Schrock and Grossman,¹⁸ Wright,²¹ Collier and Pulling,¹⁷ and Chen²⁴ and found that all were inadequate in that χ_{tt} did not appear to be the proper correlating parameter. In particular, for constant heat flux, mass velocity, and quality, the data showed that h_{TP}/h_{conv} increased with increasing pressure; whereas the correlations predict the opposite, due to the effect of pressure on the physical properties in χ_{tt} . Stone⁴⁷ did not propose a correlating

technique, but he did speculate that a nonequilibrium model might be more successful.

Owhadi et al.

Owhadi et al.,⁴⁸ in a study of forced-convection boiling of water at atmospheric pressure in electrically heated, helically coiled tubes, found reasonably good agreement of their data with Chen's²⁴ correlation. The predicted coefficients were about 15% high at low qualities and about 15% low at high qualities. On the other hand, the correlation of Dengler and Addoms¹⁵ predicted values, on the average, only 10% less than the experimental values in the low and medium quality ranges. Both correlations predicted coefficients several times too high for the very high (>90%) quality range. This is to be expected since neither is applicable under liquid deficient conditions.

Pujol and Stenning

Pujol⁴⁹ studied the effect of flow direction on convective boiling heat transfer in an electrically heated, stainless steel pipe, using Freon-113. The test section was 0.622-in. nominal inside diameter pipe, having a total heated length of 19 ft, bent to a serpentine form to provide two upflow and two downflow sections of equal length. The mass flux was varied over the range 1.48×10^5 to 1.40×10^5 lb/hr·ft², with qualities in the range of 0 to 70%; pressures ranged from 17 to 64.2 psia. The derived Martinelli parameter, x_{tt} , showed a spread of 0.065 to 84; while the boiling modulus, N_{BO} , varied between 1.09×10^4 and 6.5×10^4 .

Pujol and Stenning²³ analyzed Pujol's data and found that the difference between upflow and downflow boiling heat transfer coefficients was small in the suppressed nucleation region but was substantial in the nucleate boiling region; higher coefficients were observed in the upflow mode.

The equation that best correlated ($\pm 15\%$) the suppressed nucleation data for both upflow and downflow modes was found to be of the type suggested by Dengler and Addoms:¹⁵

$$\frac{h_{TP}}{h_{conv}} = 4.0 \left(\frac{1}{x_{tt}} \right)^{0.37}, \quad (2-40)$$

where h_{conv} is as given by Eq. (2-2).

Correlations of the type suggested by Schrock and Grossman,¹⁸ applicable to both the nucleate boiling and the suppressed nucleation regions, were also developed. For upflow, Pujol and Stenning's best equation was:

$$\frac{h_{TP}}{h_{conv}} = 0.90 \left[N_{Bo} \times 10^4 + 4.45 \left(\frac{1}{x_{tt}} \right)^{0.37} \right]; \quad (2-41)$$

and for downflow:

$$\frac{h_{TP}}{h_{conv}} = 0.53 \left[N_{Bo} \times 10^4 + 7.55 \left(\frac{1}{x_{tt}} \right)^{0.37} \right]. \quad (2-42)$$

In the limit, as N_{Bo} tends to zero, both Eqs. (2-41) and (2-42) degenerate to give Eq. (2-40). The accuracy of Eqs. (2-41) and (2-42) is not as good as that of Eq. (2-40) in the suppressed nucleation region, since this region is not necessarily restricted to low values of N_{Bo} . The equations predict values about 10 to 20% high in this region.

In the discussion of Pujol and Stemring's paper,²³ these authors point out that they compared their suppressed nucleation data with Chen's²⁴ macroconvection term and found good agreement. Although the slope of Chen's enhancement factor (Fig. II-1) was larger than the authors, this was attributed to a slight difference in the definition of the single-phase forced-convection coefficient.

Rohsenow

Historically, Rohsenow¹² made one of the first attempts to correlate forced-convection boiling. He suggested a simple additive-type equation for the total heat transfer rate:

$$q''_{TP} = q''_{boil} + q''_{conv} , \quad (2-43)$$

where q''_{conv} was computed from a modified Dittus-Boelter equation in which a coefficient of 0.019 (rather than 0.023) was used, and q''_{boil} could be correlated with the Rohsenow pool boiling correlation,³² Eq. (2-23).

Piret and Isbin

A number of investigators have followed the Rohsenow method for correlating forced-convection boiling heat transfer data. Piret and Isbin⁵⁰ correlated natural-circulation boiling heat transfer data (at atmospheric pressure) for water, isopropyl and n-butyl alcohols, carbon tetrachloride, and potassium carbonate solutions in this manner with good results.

Beaver and Hughmark

Beaver and Hughmark⁵¹ also used this method to correlate the nucleate boiling region in a single tube thermosiphon investigation

under sub- and atmospheric pressure conditions for 12 organic fluids. Their data for the suppressed nucleation region did not correlate well with the correlation of Dengler and Addoms,¹⁵ but showed improved agreement when plotted against the reciprocal of the liquid volume fraction to the 0.8 power in place of the reciprocal Martinelli parameter.

Blatt and Adt

Blatt and Adt⁵² investigated forced-convection boiling heat transfer to Freon-11 and Freon-113 in horizontal tubes and correlated their results by the Rohsenow method. Using values of C_{sf} (the surface-fluid factor in Rohsenow's pool boiling correlation) determined experimentally, all their nucleate boiling data were correlated with an average deviation of about 30%.

Sato et al.

Sato et al.²⁶ recently (1972) reported on an extensive investigation of flow patterns and heat transfer in the (saturated) nucleate boiling region for upflowing water in a rectangular flow channel (15 x 10 mm), having a heated length of 1000 mm. As observed by many investigators, the transition between the various flow patterns was found to be complex and dependent on flow rate, heat flux, heated length, and channel diameter. In the bubbly flow regime, bubble diameter decreased with increased flow rate; the bubbles had a tendency to collect mainly near the wall. They observed that the liquid film on the wall adjacent to slugs of vapor was rather thick when noticeable nucleate boiling occurred. In the slug-to-annular transition flow regime, bubbles were observed to nucleate and grow to large diameters in the liquid film. Slug flow was

not observed in the higher flow range, instead the bubbly annular regime appeared. The transition between the flow patterns occurred at lower quality as the heat flux, the flow rate, or the heated length was increased and as the flow channel diameter decreased.

Based on the observed flow patterns, Sato et al.²⁵ correlated their saturated boiling heat transfer data for the nucleate boiling region (bubble flow range) by an additive correlation of the form suggested by Rohsenow¹² as:

$$q''_{TP} = q''_{boil} + q''_{conv} , \quad (2-44)$$

where

$$q''_{boil} = 4.5 e^{P/20} (G_L \times 10^{-6})^{-1/3} \Delta T_{sat}^{3.6} , \quad (2-45)$$

$$q''_{conv} = 0.0506 \frac{k}{D_{eq}} (G_L \times 10^{-6})^{-0.45} N_{Re,TP}^{0.8} N_{Pr,L}^{0.4} \Delta T_{sat} , \quad (2-46)$$

$$G_L = \frac{1-x}{1-f_d} G , \quad (2-47)$$

$$f_d = 1 - \left(\frac{(1-x) \rho_g}{x \rho_L + (1-x) \rho_g} \right)^{0.5} , \quad (2-48)$$

and

$$N_{Re,TP} = \frac{G_L D_{eq}}{\mu_L} . \quad (2-49)$$

Metric units are employed in the above equations; in particular, the pressure normalization factor in Eq. (2-45) is in kg/cm² and is applicable over the range 1 to 150 kg/cm² (Ref. 53). The pressure and two-phase flow dependencies given in the above equations were derived specifically from these authors' work and are not necessarily general.

However, they speculate that their method of correlating the flow patterns with the heat transfer equations may have general applicability.

Kutateladze, Domanskii, Andreyev, et al.

The Russian literature generally correlates forced-convection boiling heat transfer by an expression of the type:

$$\frac{h_{TP}}{h_{conv}} = \left[1 + \left(\frac{a h_{PB}}{h_{conv}} \right)^n \right]^{1/n}, \quad (2-50)$$

where a and n are empirically determined constants. Kutateladze⁵⁴ recommends $a = 0.7$ to 0.8 and $n = 2$. Domanskii⁵⁵ suggests $a = 0.5$ and $n = 3$. Andreyev et al.²⁶ indicate $n = 3$ and give a modified pool boiling expression for the term " ah_{PB} ."

Chawla

Chawla^{56,57} proposed a parameter different from the Martinelli parameter for correlating the forced-convection boiling heat transfer coefficient. He defines a two-phase flow parameter, ϵ , as the ratio of the mean liquid velocity to the mean vapor velocity, which he correlates^{58,59} (using measured two-phase flow pressure drop data) as a function of x , $N_{Re,l}$, $N_{Fr,l}$, ρ_l/ρ_g , μ_l/μ_g , and relative tube wall surface roughness, e/D . The results of this correlation are presented in Fig. II-3. Since the major portion of the heat is transferred to the liquid film on the wall, Chawla defines an equivalent hydraulic diameter for the liquid film as:

$$D_{eq,l} = D \left[1 - \left(1 + \frac{1-x}{x \epsilon \rho_l / \rho_g} \right)^{-0.5} \right], \quad (2-51)$$

for use in the definition of the two-phase Nusselt modulus, $N_{Nu,TP}$.

Fig. II-3. Chawla's two-phase flow parameter, ϵ .

Using the two-phase heat transfer data of Penman and Tait⁶⁰ for water, methanol, ethanol, 2-propanol, and chloroform, of Lavin and Young⁶¹ for Freon-12 and Freon-22, and of Chawla⁵⁷ for Freon-11, he was able to correlate the entire data set to within 30%. Chawla's set of recommended^{58,57} dimensionless equations for the forced-convection boiling heat transfer in the quality range $0 < x < 1.0$ is:

For $N_{Re,l} \cdot N_{Fr,l} < 109$:

$$N_{Nu,TP} = 0.0066 (N_{Re,l} N_{Fr,l})^{0.475} N_{Re,l}^{0.35} N_{Pr,l}^{0.42} \left(\frac{\mu_l}{\mu_g}\right)^{0.8} \left(\frac{\rho_l}{\rho_g}\right)^{0.3} \frac{x}{1-x} ; \quad (2-52)$$

and for $N_{Re,l} \cdot N_{Fr,l} > 109$:

$$N_{Nu,TP} = 0.015 (N_{Re,l} N_{Fr,l})^{0.3} N_{Re,l}^{0.35} N_{Pr,l}^{0.42} \left(\frac{\mu_l}{\mu_g}\right)^{0.8} \left(\frac{\rho_l}{\rho_g}\right)^{0.3} \frac{x}{1-x} , \quad (2-53)$$

where

$$N_{Nu,TP} = \frac{h_{TP}}{k_l} D \left[1 - \left(1 + \frac{1-x}{x \rho_l / \rho_g} \right)^{-0.5} \right], \quad (2-54)$$

and ϵ is as presented graphically in Fig. II-3.

Pool Boiling

Zuber and Fried

Zuber and Fried³ discussed nucleate pool boiling and compared in detail a number of proposed correlations with particular emphasis on their application to cryogenic liquids. In addition to a lucid discussion of the dimensionless moduli used to correlate pool boiling heat transfer, this review (published in 1962) provides a list of 61 references. In 1964 Zuber⁶² critically reviewed the state of the art for nucleate pool boiling (101 references are included). These two reviews are strongly recommended for an overview and detailed discussion of the world literature available at that time. According to Zuber,⁶² more than three dozen equations had been proposed for correlating nucleate pool boiling; a number of additional ones have been suggested since 1964. Some of the more widely known are given in Refs. 34, 38, 46, and 63 through 80.

Since the high heat transfer rates associated with nucleate boiling are attributed to bubbles that induce a strong localized agitation in the thermal boundary layer, most of the correlations are formulated in terms of a bubble Reynolds modulus and a bubble Nusselt modulus. The significance of these is discussed by Zuber and Fried.³ Most of the correlations result in an expression for the heat transfer coefficient of

the form:

$$h_{PB} = \text{const} (T_w - T_{\text{sat}})^b, \quad (2-55)$$

where b varies between 1 and 3, and the constant depends on the thermodynamic properties of the fluid. Rohsenow³² included a surface-fluid-combination factor in the constant to account for some of the properties thought to be important but not included in the correlation. Numerous investigations (e.g., Refs. 41 through 45, 70) have shown that different nucleating surface characteristics affect not only the constant in Eq. (2-55) but also the power of the temperature difference factor.

Nishikawa and Yamagata

Nishikawa and Yamagata⁶⁹ were among the first to propose that pool boiling data correlated well with a two parameter equation of the type:

$$q''_{PB} = \text{const} (T_w - T_{\text{sat}})^b (N/A)^c, \quad (2-56)$$

where N/A is the bubble population (or nucleating site) density. Subsequently, a number of investigations (e.g., Refs. 70 through 73, 75, 76) confirmed this observation; and several of the proposed correlations include this parameter. A difficulty lies in the fact that reliable methods are not available to predict the site density.

Mikic and Rohsenow

Mikic and Rohsenow⁷⁵ assumed that the experimental results of Brown⁸¹ on site density could be generalized to provide a relationship for the site density for use in their proposed correlation; insufficient evidence exists to substantiate this assumption. Thus, in the absence of a reliable method for quantifying the site density, the designer has

few choices available for predicting the performance of a given pool boiling system. Since the Rohsenow correlation³² purports to do this and has been shown to correlate experimental results with reasonable accuracy, wide use is made of it. In consequence, numerous investigations^{33,50,82-84} have been undertaken to broaden the range of surface-fluid factors for this correlation.

Nucleation Models

Zuber⁶² commented that there was no completely satisfactory theory of bubble nucleation from a heated surface. Experimental observations show that bubbles nucleate at superheat temperatures considerably below values predicted by classical theories. Clark *et al.*⁸⁵ and Griffith and Wallis⁸⁶ showed conclusively that bubbles nucleate from surface irregularities, such as pits, cavities, and scratches. The interaction of the boundary layer, the diameter of pits, and the heat flux density has been analyzed by a number of investigators, based on the assumption that the temperature profile is linear near the heated wall. This assumption appears well substantiated by the temperature measurements of Jiji and Clark,⁸⁷ Marcus and Dropkin,⁸⁸ and Bobst and Colver.⁸⁹ Hsu³⁶ proposed a criterion that has proved reasonably successful for determining the size range of effective cavities. Collier and Pelling¹⁷ were able, using Hsu's method, to estimate the transition from nucleate boiling to forced convection with very good agreement between predicted and measured values.

Rohsenow⁹⁰ reviewed nucleation phenomena and, using Brown's results,⁸¹ suggested that the position of the boiling curve for pool

boiling should be a strong function of the heating surface characteristics but that for forced-convection boiling the influence would be much less.

Heat Transport Models

Earlier fundamental nucleate boiling studies were based on the behavior of separate bubbles; i.e., the regime of isolated bubbles.⁶² In this regime, which is restricted to low heat flux densities and hence to low vapor flow rates, liquid is set in motion by growing and rising bubbles. Source flow (liquid displacement) accompanies bubble growth in the thermal boundary layer, and liquid entrainment takes place in the wake of rising bubbles. This source and wake flow induces liquid circulation in the boundary layer. It appears that heat transfer in this regime is primarily by localized convection, or microconvection as it is often referred to in the boiling literature.

As the heat flux increases, bubbles begin to interact to form vapor columns and large vapor patches; this has been termed the regime of interference.⁶² The transition from the first regime to the second takes place when the space between the two adjacent bubbles is about 1.5 to 2 times the bubble diameter.^{91,92} Gaertner⁹² noted that the change in flow regime corresponded to the inflection point on the boiling curve. He showed further that the transition was not only marked by a change in the flow in the vicinity of the heating surface but also by a change in the thermal boundary layer thickness and, therefore, in the heat transfer mechanism. The heat transfer mechanism in this regime is thought to be microlayer evaporation and latent heat transport.

Several models have been hypothesized to explain nucleate pool boiling, some of which are undoubtedly valid in part. However, experimental verification of the various hypotheses has been difficult to obtain due to the complexities of the boiling phenomena. A brief overview will be given of some of the more prominent models; more detail is found in the referenced sources.

Microconvection model. This model, depicted schematically in Fig. II-4a, assumes that random microconvection is induced in the superheated layer by bubble action. Heat transfer is considered to take place from the heated surface to the superheated liquid; augmentation results from the action of the induced microcurrents. Heat transport by vapor bubbles is considered negligible. This concept is generally disregarded in the current boiling literature.

Latent heat transport model. This concept (shown schematically in Fig. II-4b) assumes that each bubble as it nucleates, grows, and departs transports heat from the heated surface and deposits in the bulk fluid (by condensation and/or bubble collapse) an equivalent bubble volume of latent heat. The model is only concerned with the mechanism by which heat is removed from the superheated layer and does not consider the transfer of heat from the surface to the superheated layer. This model was highly regarded by early boiling researchers but fell into disfavor as other models were proposed. In 1964, Rollis and Jawurek⁹³ reintroduced it by reporting an experiment in which it could be shown that latent heat transfer and natural convection together accounted for the total heat transfer for saturated water boiling on a

Fig. II-4. Boiling heat transfer models.

horizontal wire. Once again, the latent heat transport model is considered a possible mechanism in the isolated bubble regime.

Vapor-liquid exchange model. This concept (see Fig. II-4c) assumes the vapor bubbles act as a piston pump. As the bubbles grow and depart from the heated surface, hot fluid is displaced from the superheated layer to the bulk liquid. Cooler fluid (from the bulk liquid) is drawn in to replace the void created by the departing bubbles. Latent heat transport is assumed negligible. The dominant mechanism in this model, as proposed by Forster and Grief,⁶⁸ is the transport of sensible heat associated with the volume of the vapor bubbles.

Enthalpy transport model. This concept, proposed by Han and Griffith,⁷⁴ is illustrated in Fig. II-4d; the transport of enthalpy from the superheated layer to the bulk liquid is accomplished by departing bubbles, somewhat like the vapor-liquid exchanger model of Forster and Grief. The quantity of heat transported by each bubble is assumed to equal the energy content of the boundary layer within the region of influence about a given nucleation site. This region is taken to be an approximately cylindrical volume of twice the bubble departure diameter and of height equal to the boundary layer thickness at the instant of departure. Transient conduction is considered to be the mechanism by which heat is transferred from the heated surface to the superheated layer. Although the model predicted Han and Griffith's experimental data with good accuracy, these investigators conceded that an extraordinary amount of data about the surface was required. Such detailed descriptions are not usually available for systems of practical interest.

Maróti,⁹⁴ in a recent (1973) publication, further develops the concept of enthalpy transport as the dominant mechanism for fully developed nucleate pool boiling of saturated fluids. He claims good agreement for his correlation with published experimental results. He also dismisses microclayer evaporation (see below) as being insignificant in most cases.

Source flow model. This model, shown schematically in Fig. II-4e, attributes the high boiling heat transfer rates to the combined action of microconvection induced by growth, detachment, and collapse of bubbles (bubble agitation) and transient conduction to the liquid flowing in to replace the collapsed bubbles (surface quenching). This model serves as the basis of Rohsenow's correlation.³² Bankoff⁹⁵ was able to express the model mathematically (without resorting to dimensional analysis) with a single arbitrary constant which must be obtained from experimental data.

Wake flow model. This concept, pictured in Fig. II-4f, was proposed by Tien⁷¹ and is based on the similarity of axisymmetric stagnation flow. It is assumed that the periodic detachment and rise of bubbles in the isolated bubble regime can be replaced with a continuous column moving upward through the boundary layer. The resultant upward flow pattern induced in the adjacent liquid in the wake of the vapor column is approximated by an inverted stagnation flow model. Boehm and Lienhard⁹⁶ modified Tien's model to include an additional term to account for heat transfer associated with the transient wake flow induced by departure of individual bubbles. According to Tien, this model can also be used for the region of bubble interference.

Enhanced natural-convection model. This model (Fig. II-4g) was proposed by Zuber⁷³ for the isolated bubble regime. It is based on the similarity between nucleate boiling and turbulent natural convection with respect to the induced circulation that governs heat transfer in both cases. The model assumes that thermally induced buoyancy in the liquid is augmented by vapor bubble buoyancy in the vicinity of the heated surface, enhancing the convective heat transfer mechanism much the same as turbulent eddies. Zuber⁷³ was able to show that the same predictive equation for the heat transfer coefficient under turbulent natural convection could also be used for the isolated bubble regime, provided the volumetric vapor fraction was taken into account. [Reference 73 also contains an excellent discussion of nucleate pool boiling and a list of 72 references.]

Microlayer vaporization model. This model, depicted in Fig. II-4h, was first suggested by Snyder and Edwards³⁷ with Moore and Mesler³⁸ being the first to present evidence in support of the model. Since this model appears promising, it is receiving wide-spread attention; and much of the current boiling literature is devoted to it (e.g., see Refs. 98 through 105).

The model assumes, based on the "no-slip hypothesis," that a bubble cannot wipe the surface dry of liquid as bubble growth takes place. Thus, a very thin sublayer (hence, the name microlayer) of liquid should be established between the growing bubble and the heated surface. Bubble growth takes place by evaporation of the thin layer (beneath the bubble) with simultaneous localized cooling of the heated surface. A significant consequence of this model is the hypothesis that the heating

surface area between bubbles plays a passive role in the process. In effect, the rising bubble column acts as a shunt through which most of the heat is transferred; and the problem of explaining the high surface-to-liquid heat transfer rate characteristic of nucleate boiling is eliminated. Optical techniques^{103,104} and surface temperature measurements (using fast response thermometry) have shown conclusively that a thin liquid film does, indeed, exist beneath a growing bubble.⁹⁸⁻¹⁰⁵ Analyses¹⁰⁶ based on these observations confirm the importance of this mechanism. However, Maróti⁹⁴ discounts the importance of this mechanism.

Thermocapillarity model. Brown⁸¹ suggested that thermocapillarity (resulting from surface tension gradients) was possibly the most important mechanism, overshadowing the others. Marangoni¹⁰⁷ was the first to report that motion can be introduced by a variation in surface tension at a liquid-vapor interface. This variation produces a force imbalance that tends to accelerate the interfacial liquid film. As a result of viscous forces, motion is induced in the adjacent fluid. Since temperature gradients exist near the heater, surface tension gradients will be present. These act to cause bubbles in nonuniform temperature fields to eject hot liquid in the cold direction. This phenomenon, known as "jetting," has been observed^{81,108} from electrically heated wires and surfaces. Based on his work, Brown concluded that latent heat transport and microlayer evaporation were not important heat removal mechanisms in pool boiling but that thermocapillarity might be the dominant mechanism. In general, thermocapillarity will remove more heat in flow boiling than in pool boiling.

Experimental Confirmation of Models

Figure II-5 is a plot of the heat transfer coefficient for nucleate pool boiling at atmospheric pressure versus the superheated film thickness as given by Bobst and Colver.⁸⁹ The data of Marcus¹⁰⁹ are also shown. Although the curves are displaced (attributed to different chemical and physical properties of the boiling surface), there is great similarity in the two data sets. The boiling coefficient is observed to increase as the film thickness decreases, with three distinct regions

Fig. II-5. Nucleate pool boiling heat transfer coefficient vs film thickness according to Bobst and Colver.

being indicated. The break in Marcus' data occurs at $h_{PB} = 700 \text{ Btu/hr}\cdot\text{ft}^2\cdot^\circ\text{F}$ ($q'' \approx 10,000 \text{ Btu/hr}\cdot\text{ft}^2$). Above this value h_{PB} increases faster with decreasing film thickness. Bobst and Colver postulate that the break corresponds to the laminar-turbulent flow transition in the boundary layer. They also observed another sharp break at $h_{PB} = 2000 \text{ Btu/hr}\cdot\text{ft}^2\cdot^\circ\text{F}$, corresponding to a heat flux of $80,000 \text{ Btu/hr}\cdot\text{ft}^2$. Above this break point, the coefficient increases at a lower rate than below the transition. They attribute this change to bubble interference, based on Gaertner and Westwater's findings¹¹⁰ that bubbles grow and depart with a constant diameter in the isolated bubble regime and that they depart with smaller diameters in the regime of interference. (Koumoutsos et al.¹¹¹ observed that the bubble departure diameter for saturated flow boiling was considerably less than for similar conditions of pool boiling and that the diameter decreased with increasing flow velocity.) Bobst and Colver⁸⁹ conclude that the effect of this change places a limitation on the fraction of heat that can be transferred by mechanisms other than those solely related to the liquid. They speculate that at the point where bubble interference begins, the boundary layer develops a condition that might be characterized as "saturated with turbulence." With further increases in the heat flux, bubble generation and agitation are not as effective and contribute less significantly to a reduction in the thermal resistance of the boundary layer. In other words, a condition of diminishing returns develops with respect to increased bubble generation.⁸⁹

Mention was made above to reviews and comparisons by Zuber and Fried³ and by Zuber⁶² of correlations proposed by a number of investigators; other reviews are also available. Kosky and Lyon⁴⁵ tested

several correlations against experimental pool boiling heat transfer data for a number of cryogenic liquids. They found that the correlations of McNelly⁶⁷ and Borishanskii et al.⁷⁷ were superior to those of Rohsenow,³² Forster and Grief,⁶⁸ Forster and Zuber,³⁶ Gilmour,⁶⁶ and Kutateladze.⁶³ It is noted that the equations having the strongest theoretical basis were found to give poorest agreement with their data.

Using their own data for water and alcohol, Borishanskii et al.⁷⁷ compared five Russian correlations over a pressure range of 1 to 60 atm. They also included the data of Cichelli and Bonilla¹¹² for heptane, benzene, propane, and pentane. They found poor agreement with the data and concluded that the correlations are only approximately valid for the conditions for which they were developed.

Rice and Calus⁴⁶ obtained pool boiling data for a number of hydrocarbons, water, and water-isopropanol azeotrope at atmospheric pressure. They tested their data against the correlations mentioned above; however, they omitted those involving nucleation site density. These investigators found that the Borishanskii et al. correlation gave the best agreement for each fluid although each curve was displaced from the others. However, by introducing an additional correlating factor relating the boiling points of the fluids, all the data could be correlated by a single line. The Cichelli and Bonilla and the Borishanskii et al. data were also correlated by this modified correlation. This does show quite clearly, based on the three independent data sets, that

- (1) the nature and structure of the heat transfer surface constitute the dominant component of the surface-fluid combination factor and
- (2) the modified correlation responds well to variations in pressure over the range of 1 to 60 atm.

Another noteworthy comparison of pool boiling correlations was published recently (1972) by Judd and Merte.¹¹³ They compared the heat flux predictions of six different correlating models to heat flux measurements of Freon-113 at the same subcooling and acceleration levels. The models tested (discussed above) are identified as:

1. latent heat transport model proposed by Rallis and Jawurek,⁹³
2. vapor-liquid exchange model proposed by Forster and Grief⁶⁸
(identical to the Forster-Zuber correlation³⁸),
3. enthalpy transport model proposed by Han and Griffith,⁷⁴
4. source flow model proposed by Bankoff⁹⁵ (this model is conceptually identical to the model of Rohsenow³²),
5. wake flow model proposed by Tien⁷¹ and modified by Boehm and Lienhard,⁹⁶ and
6. enhanced natural-convection model proposed by Zuber.⁷³

Judd and Merte¹¹³ found that only Tien's wake flow model (as modified by Boehm and Lienhard) was capable of predicting boiling heat transfer results for saturated Freon-113 on an oxide-coated glass heater surface at standard gravity. All the models overpredicted the measured results with increasing subcooling and acceleration level. A slightly modified version of Zuber's enhanced natural-convection model showed improvement at the higher levels of subcooling. They suggest that the tested models omit some mechanism that is inoperative (or insignificant) under saturated boiling conditions but that is very sensitive to the level of subcooling and acceleration. Mass transfer resulting from microlayer evaporation at the bubble base with subsequent condensation of the vapor at the vapor-liquid interface appears to be the most likely prospect.

Kenning and Kao¹¹⁴ tested Brown's⁶¹ hypothesis under simulated flow boiling conditions by injecting gas bubbles into the thermal boundary layer of a heated channel with upflow of water. They argue that thermocapillary effects should be stronger for gas bubbles with a near-adiabatic boundary condition than for vapor bubbles for which internal evaporation-condensation processes reduce the interfacial temperature (and therefore surface tension gradients). The effects should also be more pronounced at high heat flux levels than at low flux levels. It was found that significant (up to 50%) improvement in the heat transfer coefficient was obtained with small quantities of injected gas (volumetric flow ratios up to 0.07) and that the improvement was independent of heat flux level over the range of 20 to 300 kW/m². No evidence of a direct contribution to heat transfer by thermocapillarity was found in their investigation. They concede that this mechanism might exert an indirect effect by modifying the bubble distribution in the vicinity of the heated wall. In a related investigation, Kao and Kenning¹¹⁵ point out that thermocapillary flow, being dependent on surface tension effects, is very sensitive to surfactants (surface-active contaminants). Numerical analyses and experimental observations both indicate for water that thermocapillary motion may be suppressed by contamination that reduces the surface tension by as little as 0.1%. In the latter study,¹¹⁵ considerable difficulty was encountered in keeping the experimental equipment "clean" so that thermocapillary motion could be observed. Thus, it appears from these investigations that thermocapillarity effects are not likely to be encountered in practical systems. Larkin¹¹⁶ also performed a numerical analysis of thermocapillary flow around a

hemispherical bubble and concluded that the improvement in heat transfer was modest. These results tend to disprove Brown's hypothesis that thermocapillarity is an important mechanism in both pool and flow boiling.

CHAPTER III

EXPERIMENTAL PROGRAM

An experimental program was developed to investigate basic concepts believed to be significant in explaining the "nucleate boiling" and the "suppressed nucleate boiling" regions observed by many investigators in forced-convection boiling of saturated fluids. The experiments were designed to elucidate and to clarify the mechanisms involved in turbulent, forced-convection boiling of saturated fluids at atmospheric pressure.

The following experiments with water and Freon-11₂ at atmospheric pressure were conceived as important to a complete investigation in this area:

1. saturated pool boiling from horizontal surfaces facing upward,
2. saturated pool boiling from vertical surfaces,
3. two-phase, forced-convection boiling in a flow channel with upward flow,
4. single-phase, forced-convection heat transfer in a flow channel with upward flow,
5. two-phase, forced-convection boiling in a flow channel with downward flow, and
6. single-phase, forced-convection heat transfer in a flow channel with downward flow.

This present investigation concentrated on the pool boiling and forced-convection upflow experiments (Items 1., 2., 3., and 4.).

The experiments were built around the use of a heat flux meter to measure the heat-transfer coefficients under carefully controlled conditions. The heat flux meter, described in detail in Chapter IV, is quite simple in principle and permits a simultaneous determination of the test surface heat flux and temperature. A small electric heater supplies heat to the boiling surface via conduction through a copper cylinder having a number of thermocouples located along its axis at known distances from the boiling surface. From the thermal conductivity of the copper and the temperature as a function of position as indicated by the thermocouples, the heat flux through the boiling surface can be calculated. The boiling surface temperature can be determined simultaneously by extrapolation of the internal temperatures. Measurement of bulk fluid temperature provides the additional information needed to calculate heat-transfer coefficients. Two identical heat flux meters were installed in a mounting plate with the boiling surfaces smooth with the surface of the plate.

The following series of experiments were carried out with water:

Series HPBWC. This test series provided saturated pool boiling data from a horizontal copper surface for heat fluxes in the range of 10^3 to 1×10^4 Btu/hr·ft². The mounting plate with the two meters installed was placed in a large Pyrex glass container with the boiling surfaces facing upward. A bubble level was used to give assurance that the mounting plate was horizontal. Demineralized and deaerated water was used as the fluid. These data are considered the base for comparison with data reported in the literature and with other data obtained in this study.

Series VPBWC. This series of experiments provided saturated pool boiling data from vertical copper surfaces for the same range of heat fluxes used in the HPBWC series. The mounting plate with installed meters was positioned in the glass container as before. The plate was oriented so that the meters were one above the other with the copper boiling surfaces in the same vertical plane. A bubble level was used to give assurance that the plane of the boiling surfaces was vertical. Demineralized and deaerated water was used as the fluid. These data were collected to compare with Series HPBWC data (from horizontal surfaces). In addition, with the meters located one above the other as described, the lower meter could generate bubbles that flowed (by natural convection) past the upper boiling surface. In this manner, any enhancement of the heat transfer on the upper meter by bubbles in the boundary layer could be determined.

Series TPBWN and TPBWC. These experiments provided two-phase, forced-convection boiling heat transfer data for vertical upflow of demineralized and deaerated water on nickel and copper surfaces. The mounting plate (with heat flux meters installed) was inserted in a specially constructed, rectangular flow channel (described in Chapter IV). Metered flow to the channel produced flow velocities past the heat flux meters at nominal values of 0.5 to 4 ft/sec, corresponding to Reynolds moduli of $1.1 \times 10^4 \leq N_{Re} \leq 9.2 \times 10^5$. Similar to Series VPBW the upstream heat flux meter could be used to generate bubbles in the boundary layer flowing past the downstream meter. In this manner, the upstream meter could be used to measure the effect of bubble enhancement in forced-convection boiling. The range of heat fluxes was the same as in the previous series of experiments.

Series SPCWN and SPCWT. These experimental sets provided forced-convection heat transfer data (nonboiling) for vertical upflow of demineralized and deaerated water on nickel and copper surfaces. Under certain flow and boiling conditions, the pressure at the heat flux meters could be increased sufficiently to suppress nucleation and thus provide forced-convection heat transfer data at essentially the same conditions as the previous two-phase boiling data. The achievement of these conditions was dictated by structural strength requirements on the glass and Plexiglas members (cover plates) of the flow channel. The pressure could be increased by either of two techniques:

- (1) slightly closing a valve in the exit line of the flow channel, or
- (2) by slightly decreasing the cooling water flow in the exit line cooler (assuming the cooler was in operation) to increase the two-phase pressure drop in the exit line. Forced-convection heat transfer data were collected for representative ranges of the independent parameters (velocity and heat flux) as used in the previous series.

Freon-113 was selected as a second working fluid to provide experimental data similar to that obtained for water but for a fluid having physical properties quite different from those of water. A comparison of some of the significant properties of Freon-113 and water is given in Table III-1. Relevant differences to be noted are (1) the thermal conductivity of Freon-113 is about one-eighth that of water, indicating a tendency for poorer forced-convection heat transfer for the refrigerant relative to water, and (2) the latent heat of vaporization and the surface tension of Freon-113 are factors of about 15 and 4, respectively, lower than the values for water, indicating much lower superheat for

TABLE III-1
COMPARISON OF PHYSICAL PROPERTIES OF WATER AND FREON-113

	Water	Freon-113
Chemical formula	H ₂ O	(Cl ₂ F)C-C(ClF ₂)
Molecular weight	18	187.4
Boiling point at 1 atm, °F	212	117.6
Freezing point, °F	32	-21
Critical temperature, °F	705.3	417.4
Critical pressure, psia	3206.7	445
Liquid density (70°F), lb _m /ft ³	62.3	68.86
Vapor density (boiling point), lb _m /ft ³	0.0373	0.462
Liquid specific heat (70°F), Btu/lb _m ·°F	1.02	0.13
Vapor specific heat (86°F), Btu/lb _m ·°F	0.44	0.161
Latent heat of vaporization (atmospheric pressure), Btu/lb _m	970.3	69.1
Liquid viscosity (86°F), cP	0.828	0.61
Liquid thermal conductivity (70°F), Btu/hr·ft·°F	0.333	0.044
Surface tension (70°F), dyne/cm	72.1	17.3

bubble nucleation and larger sized bubbles for the refrigerant for the same boiling surface characteristics.

The above described set of water experiments was repeated with Freon-113 as the test fluid, except that the nickel-plated surface was not available. The same range of flows were tested; however, the maximum heat flux was limited to approximately 40,000 Btu/hr·ft² to avoid film boiling. Since contamination of the copper surface was not a problem with the freon, these data proved to be more consistent and reproducible than those obtained with water. Consequently, the freon data

form the basis for the testing of analytical models in Chapter V, with the water data serving to support the results of the freon tests.

A descriptive nomenclature, consisting of five letters and a numeral followed by a serial number (e.g., ABCDE2-001), was employed to identify the individual experiments. The first three letters characterize the general experimental arrangement, the fourth letter identifies the fluid, the fifth letter denotes the surface metal, and the numeral indicates on which heat flux meter the data were taken. Thus, for the first three letters HPB = horizontal pool boiling, VPB = vertical pool boiling, TPB = two-phase convective boiling, SPC = single-phase forced convection, and TPX = two-phase convective boiling with "high" void fraction flow. These descriptors will become apparent in the discussion of the results in Chapter V. As for the fluid identification, F = Freon, W = water, while the surface is denoted by N for nickel and C for copper. The heat flux meters are denoted by 1 (upstream) or 2 (downstream). For example, HPBWC2-09 identifies test run No. 9 with heat flux meter No. 2 for pool boiling of water on a horizontal copper surface.

CHAPTER IV

EQUIPMENT AND PROCEDURES

The equipment was designed to permit direct calculation of heat transfer coefficients from the experimental data obtained with the test configurations described in the previous chapter (Experimental Program). Central to the experiments was a heat flux meter that enabled simultaneous determination of the test surface heat flux and temperature. Details of this heat flux meter and of other test equipment are discussed in following sections; an outline of the procedures used in the experimental program follows the description of the equipment.

Flow Test FacilityTest Loop

A test loop was assembled from available components to provide a facility in which heat flux meters (described below) could be operated to obtain forced-convection and two-phase, convective boiling heat transfer coefficients in the upflow mode. Initially constructed of stainless steel, brass, and copper, the loop was subsequently modified on evidence of corrosion to eliminate all brass and copper surfaces exposed to the circulating hot water. A schematic flowsheet of the facility is shown in Fig. IV-1.

Major components in the system included a reservoir-head tank, a small reflux condenser (vented to the atmosphere), two centrifugal pumps connected in parallel, a preheater (in three sections), a trim-cooler, two turbine flow meters in parallel flow lines, a test flow channel with

Fig. IV-1. Schematic flowsheet of test loop.

two heat flux meters, an after-cooler, a pump bypass (with filter), and associated instrumentation. Valves permitted controlled variability in operating conditions at the test section. The piping system was designed for test flows up to 25 gpm; maximum system pressure was about 23 psig at

the pump discharge header. Individual components are described in detail in later sections.

The experimental system prior to application of thermal insulation is pictured in Fig. IV-2. Prominent features, as noted in the photograph, include (1) a pump control and monitoring cabinet to the extreme left, (2) the reservoir-head tank with reflux condenser on the framework in the center, (3) the two pumps and interconnecting piping located beneath the reservoir, (4) the test stand with flow channel installed for upflow, and (5) an instrument panel on the extreme right containing a potentiometer, thermocouple switches, flow measuring readout equipment, etc. Other control and readout instrumentation are not visible in this photograph.

Heat Flux Meter

The heat flux meters were constructed similar to those used by a number of investigators (Refs. 74, 84, 88, 117 through 120). Based on reported difficulties^{74,117,118} and experience gained from a number of earlier models fabricated at this Laboratory, the design selected for the current research was as shown in Fig. IV-3.

In principle, the heat flux meter is quite simple and permits simultaneous determination of the boiling surface heat flux and temperature. A small electric heater supplies heat to the boiling surface via conduction through a copper body having a number of thermocouples located along its axis at known positions relative to the boiling surface. From the known thermal conductivity of the copper and the temperature gradient indicated by the thermocouples, the heat flux is calculated. The

Fig. IV-2. Test loop.

ORNL DWG 73 7129

Fig. IV-3. Details of heat flux meter.

boiling-surface temperature is obtained by extrapolation of this same temperature gradient.

Reducing the conceptual design of the heat meter to practice was not simple, with the primary concerns being heat losses and accuracy of the temperature measurements. By careful attention to the design and construction details and to the calibrations, the meters described herein functioned exceptionally well.

The theory of the heat flux meter assumes a highly conductive material so that the isotherms are normal to the axis of the conductor and local perturbations in the heat flow path are quickly smoothed out. A good conductor also minimizes the influence of heat losses to the surroundings. Oxygen-free copper (ASTM B-187, Type OF, 100% electrical conductivity) was selected for this use. Sufficient stock from the same 1-in.-diam bar was obtained to fabricate the four meters prepared for this study. A coiled heating element was brazed into a machined recess in one end of the copper body. A thin-walled stainless steel tube was brazed (simultaneously with the heater brazing operation) to the heater end of the copper bar to serve as a rear guide and support to minimize bending stresses on the thin stainless steel flange at the boiling end. The length of this tube was initially a few inches longer than necessary both to facilitate handling and to serve as an anchor for the small diameter sheathed thermocouples during final fabrication; the tube was cut to proper length prior to installation of the meter in the mounting plate. Tubing was selected in preference to a solid rod for this function to reduce heat losses by axial conduction.

Approximately 10 in. of commercially available single-conductor, sheathed heater wire was tightly coiled around the rear support tube and brazed into the machined recess on the end of the copper cylinder. The heating conductor was No. 1- B&W Inconel 600 wire ($\sim 1.6 \text{ ohm/ft}$), and the sheath was 0.062-in.-OD stainless steel type 304. Compacted magnesium oxide electrically isolated the sheath from the central conductor. About 1 in. of additional length was necessary on each end of the coil to construct reliable electrical connections. Closely fitting copper tubing was brazed over the sheath of these free (connector) ends of the coils to conduct heat back to the copper cylinder and, thus, to prevent overheating and premature failure of the heater extensions. Small nickel connectors were used to join (by soldering) the Inconel heater wire to the electrical lead and the voltage tap wires. The joint was potted in a small machined sleeve fitted over the end of the sheath, using a ceramic cement, to provide mechanical strength and electrical isolation for the fragile connection.

Initially, ac electrical power was supplied independently to each heat meter, using two Variacs in series for control. The first Variac reduced the 120 V supply to about 12 V, and the second Variac provided fine adjustment of the heater voltage over the operating range of ~ 1 to 10 V. After preliminary operation disclosed inadequate control and undesirable voltage fluctuations, this method was discarded in favor of individually regulated dc power supplies for each meter. Two Kepco power supplies (Model SM-75-EXM, 0-75 Vdc, 0-3 A) were used for the high flux ($q'' \approx 10,000 \text{ Btu/hr}\cdot\text{ft}^2$) range. Since these units would not control

at voltage levels below about 4.5 V, two MJE power supplies (Model QR-10-10C, 0-10 Vdc, 0-10 A) were used for the low heat flux range. These units successfully eliminated the operational difficulties with voltage control and stability.

The copper cylinder was reduced in diameter to 0.750 in. nominal in the temperature measuring region to increase the heat flux, and thus, provide sufficient temperature differences for accurate measurement at low heat flux levels. Based on the maximum design heat flux of 50,000 Btu/hr·ft² in the copper cylinder, the temperature gradient was ~18.5°F/in. With the thermocouples spaced nominally at 0.375 in., the difference between adjacent thermocouples was about 6.9°F, corresponding to an emf of 160 μV for Chromel-Alumel thermocouples. Similarly, at the minimum design heat flux of 1000 Btu/hr·ft², the temperature gradient was about 0.37°F/in. (0.14°F between adjacent thermocouples). This latter difference corresponded to an emf of about 3.2 μV, which was considered acceptable for calibrated thermocouples in conjunction with use of a Leeds and Northrup K-3 potentiometer as a readout instrument (0.5 μV scale subdivisions).

Five 0.040-in.-OD stainless steel sheathed, Chromel-Alumel grounded junction thermocouples were soldered in holes drilled to the center line of the copper cylinder. [While most investigators have used only three thermocouples to determine the gradient in similar devices, it was decided that the improved statistics derived from five thermocouples justified their use.] The thermocouples were nominally spaced 0.375 in. apart; however, their exact location relative to the boiling surface

was determined by accurate measurements of axial position and perpendicularity relative to the axis of the cylinder. These measurements (Table IV-1), made with a 100:1 Shadowgraph projector, were used to calculate the position of the junctions for subsequent data analysis.

All thermocouples were calibrated simultaneously by submerging the heat flux meters (without insulation) in an oil bath. These calibrations were performed by the Oak Ridge National Laboratory's Standard Laboratory against standards traceable to the National Bureau of Standards. The calibration included the heat flux meters, the thermocouples, the selector switch, the ice bath reference, and the copper leads up to the point where they were connected to the readout potentiometer. Results of the calibrations are given in Appendix A-3 in 1968 IPTS units.¹²¹

A section of the copper bar stock from which the heat flux meters were machined was used to determine the thermal conductivity at room temperature, using very accurate measurements of electrical properties and appropriate lattice corrections. This determination was performed by Metals and Ceramics Division personnel¹²² of Oak Ridge National Laboratory. From the data obtained at room temperature, a best-estimate curve was produced for the temperature range of $80 \leq T \leq 450^\circ\text{F}$, based on accurate ($\pm 0.7\%$) thermal conductivity data reported¹²³ for very pure copper. The best-estimate curve¹²³ for the copper used in the heat flux meters is:

$$k = 230.1755 - 0.018617 T \quad (4-1)$$

The estimated accuracy¹²² of the above equation is $\pm 2.0\%$ over the full temperature range.

TABLE IV-1

AXIAL POSITION OF THERMOCOUPLE JUNCTIONS RELATIVE TO BOILING SURFACE

Heat Flux Meter No. 1^a

Thermocouple (see Fig. IV-3)	TC-1	TC-2	TC-3	TC-4	TC-5
Distance from Boiling Surface, in.	0.4579	0.8336	1.2066	1.5781	1.9576
Spacing between Thermocouples, in.	0.3757	0.3730	0.3715	0.3795	

Heat Flux Meter No. 2^b

Thermocouple (see Fig. IV-3)	TC-1	TC-2	TC-3	TC-4	TC-5
Distance from Boiling Surface, in.	0.4508	0.8151	1.2003	1.5784	1.9532
Spacing between Thermocouples, in.	0.3643	0.3852	0.3781	0.3748	

^aDiameter of heat meter body = 0.7493 in. (meter serial No. 2).

^bDiameter of heat meter body = 0.7495 in. (meter serial No. 3).

Bubble nucleation at the periphery of the boiling surface and/or on the flange surface has been reported by a number of investigators;^{74,117,118} this was also a problem with earlier designs at this laboratory. As shown in Fig. IV-3, a thin stainless steel flange was brazed to the end of the copper cylinder to alleviate this problem. Fabrication proved difficult (from both brazing and finish machining viewpoints) but was successfully accomplished by special brazing and machining techniques. The thickness of the flange was rough machined to 0.125 in. for the vacuum furnace brazing operation. A gold-nickel alloy (82 w/o Au, 18 w/o Ni, melting point: 1742°F) was used without flux to eliminate objectionable voids and/or inclusions and to provide a ductile transition alloy in the joint. Following the brazing operation, the stainless steel flange and copper cylinder were finish machined to a final thickness of 0.010 in. The surface was then hand lapped until flat with 400-grit aluminum oxide compound. This was followed by a finish lap, using 1200-grit aluminum oxide compound to produce a bright mirror-like surface with a 3 to 4 RMS finish. The surfaces exposed to the test fluids were then nickel plated by a sputtering process to provide a thickness of 0.0001 in. of nickel on meter No. 1 and 0.0003 in. on meter No. 2. Since this plating process faithfully reproduces the surface finish of the substrate for the finished surface, no further treatment (other than cleaning with alcohol and air drying) was given to the boiling surface. Figure IV-4 is a photograph of the completed meters; the bright surface finish and the thin section (made apparent by slightly different reflective characteristics) are evident. Nickel plating was selected for the surface exposed to the test fluid, since it is known

PHOTO 1659-73

Fig. IV-4. Instrumented heat flux meter.

to be less prone to oxidation and, hence, gives reproducible results, particularly after an initial "aging" period. Also, data reported for pool boiling on nickel¹¹⁹ and nickel-plated^{109,124,125} surfaces under similar conditions could be used as a check against the data generated in this research. Surface aging and nonreproducible nucleation behavior during operation made necessary subsequent surface treatment; this is considered in detail in Chapter V, Discussion of Results.

Extensive calculations were performed, using the HEATING-3 computer program,¹²⁶ to assess the suitability of the final design for alleviating the bothersome nucleating problems resulting from heat losses through the stainless steel flange to the test fluid. The very thin nickel plating was ignored in the computer studies.

HEATING-3,¹²⁶ a modified version of the HEATING program,¹²⁷ is designed to solve a wide variety of steady-state and/or transient heat conduction problems in one-, two-, or three-dimensional Cartesian or cylindrical coordinates. Input values of thermal conductivity, density, and specific heat may be dependent on temperature at each lattice point. Specified heat generation rates may be time dependent. The boundary conditions may be fixed temperatures or any combination of constant heat flux, forced convection, natural convection, and radiation and can be from-surface-to-boundary, from-surface-to-surface, or from-surface-to-surface plus conduction. The mesh spacing can vary along each axis, with as many as 1750 lattice points, 100 regions, 50 materials, and 50 boundary conditions being accommodated.

Figure IV-5 presents the results of a typical design calculation for the meter assembly submerged in 212°F water with pool boiling taking place on the boiling surface and natural convection on all other exposed

Fig. IV-5. Plot of temperature and heat flux distribution on boiling surface based on computer calculations.

surfaces. This figure shows the computed radial distribution in the heat flux to the fluid and the surface temperature. As indicated, the heat flux and temperature are quite uniform over the central boiling region and decrease sharply at the inner edge of the stainless steel flange. For the example, the heat flux is 50,000 Btu/hr·ft² over the boiling surface ($h_{PB} = 2300 \text{ Btu/hr·ft}^2 \cdot ^\circ\text{F}$) and decreases precipitously to about 4000 Btu/hr·ft² in 0.010 in., while the temperature decreases from 233.75°F to about 230.98°F in the same radial distance. At 0.050-in. radial distance, the temperature is decreased to 223.68°F. If higher coefficients (enhanced natural convection) had been specified for the stainless steel flange, the abrupt decrease at the inner edge of the flange would be more pronounced. Based on these results and others discussed in Appendix B, it was concluded that the design selected would eliminate nucleation on the stainless steel flange.

A thermocouple, identical to those installed in the copper cylinder, was soldered into the stainless steel flange as indicated in Fig. IV-3 to measure the flange temperature for comparison with the computed distribution. This thermocouple was calibrated simultaneously with the ones in the copper; the results are reported in Appendix A-3.

The heat meters were calibrated for heat losses prior to use in the pool boiling and the flow experiments; the results are given in Appendix A-4.

Mounting Plate Assembly

A brass mounting plate was designed to retain the two heat flux meters in fixed relative positions as shown in Fig. IV-6. The mounting plate could be inserted through an opening in the back side of the flow

Fig. IV-6. Mounting plate and heat flux meter assembly construction details.

channel so that the flow boundary was smooth and continuous (the two surfaces were machined and polished simultaneously to minimize boundary layer disturbances). A neoprene O-ring was used to seal the mounting plate to the flow channel. The mounting plate assembly was also used as a fixture for positioning the heat flux meters in the pool boiling experiments.

The stainless steel flanges on the meters were tightly fitted into machined recesses in the mounting plate. Special machining precautions were taken to give assurance of minimum boundary layer disturbance. The meters were sealed to the mounting plate as indicated in Fig. IV-6, using Buna-N O-rings. Small Buna-N O-rings with retainers were used under the head of the mounting screws to prevent inleakage when the assembly was submerged in the pool boiling experiments. Flanged cylinders were soldered on the back side of the mounting plate to provide a leak-proof protective housing for the meters. A flanged cover, with Buna-N O-ring seal and a tube fitting for egress of the thermocouples and electrical leads, completed the water-tight enclosure.

Two layers of "Min-K" flexible, thermal insulating tape ($1/4$ -in. thick, $k = 0.20 \text{ Btu}\cdot\text{in.}/\text{hr}\cdot\text{ft}^2\cdot^\circ\text{F}$) filled the space between the heat meters and the cylindrical housing. A rear support spider, machined from "Mykroy-1100" ($k = 2.8 \text{ Btu}\cdot\text{in.}/\text{hr}\cdot\text{ft}^2\cdot^\circ\text{F}$), was centered on the thin-walled stainless steel tube to minimize heat losses from the heater via axial conduction.

Figure IV-7 is a photograph of the mounting plate and heat flux meters in the final stage of assembly. One meter is shown in place on the upstream end of the plate. In the completed assembly, the leads exit from the cylindrical housing through the side tubing connection of

Fig. IV-7. Mounting plate and heat flux meter assembly.

the end closure flange. This tubing (not shown) extends a sufficient distance to terminate above the top closure of the pool boiling apparatus. The leads are sealed to the open end of the protective tube, using an epoxy sealant to permit operation with a slight buffer gas pressure.

A sharp-edged pressure tap (0.032-in. drilled hole) for local pressure measurements is located midway between the heat flux meters and slightly to one side of the center line (to avoid flow disturbances) as seen in the photograph. Locating dimensions of the tap are indicated in Fig. IV-6.

Shakedown operation of the flow facility revealed significant corrosion of the brass surfaces exposed to the hot ($\sim 215^{\circ}\text{F}$) demineralized and deaerated water. This raised the possibility that, if not corrected, corrosion products might be plated out on the hot surface of the heat flux meters. Consequently, the exposed brass surfaces were nickel plated to preclude this possibility.

Flow Channel

A rectangular flow channel was machined from brass stock. Pertinent dimensions of the channel are given in Fig. IV-8. To provide the actual flow channel, a cavity 3.000 in. in width by 0.500 in. in depth ($D_{eq} = 0.857$ in.) was milled in the brass stock with the heat flux meter mounting plate (with excess thickness at this stage) bolted in position. This technique gave assurance that the mounting plate finished surface was precisely the same as the flow channel. The channel-mounting plate assembly was polished after machining to give a surface finish better than 32 rms. Brass couplings, 1 1/2-in. nominal pipe size,

Fig. IV-8. Flow channel construction details.

were soldered on the back side of the channel at each end to provide inlet and exit connections.

Originally, a 5-in.-long section of "honeycomb" aluminum was machined to fit the flow channel and pinned in place near the inlet to serve as a flow straightener as shown in Fig. IV-8. Severe galvanic corrosion of the aluminum and brass flow channel occurred during the shake-down operation; the brass flow channel was nickel plated and the aluminum flow straightener was replaced with a 5-in.-long section of stainless steel "mist separator" ("Yorkmesh"; similar to steel wool). Although the mesh introduced considerably higher pressure drop relative to the "honeycomb" section, it was very effective in dispersing the large turbulent eddies generated by the inlet design.

The center line of the first heat flux meter was located 19.125 in. ($L/D_{eq} \approx 22$) downstream of the flow straightener. The center line of the second meter was 2.875 in. ($L/D_{eq} = 3.4$) downstream of the first.

Sharp-edged pressure taps were located 1 in. upstream and downstream of the flow straightener, midway between the meters, and 1 in. upstream of the exit. The pressure taps were made by drilling a 0.032-in. hole through the backside of the channel and then counterdrilling to 0.250-in. diameter to a depth of 0.375 in. A short section of copper tubing was soldered in the counterdrilled hole to facilitate external tubing connections.

Compression-type seals were provided at the exit end to permit insertion of two calibrated thermocouples for measuring the bulk fluid temperature in the vicinity of each heat flux meter. The thermocouples were 0.040-in.-diam stainless steel sheathed, Chromel-Alumel, with grounded junctions.

Since visual observation was the only technique available in this investigation for detection of nucleation on the boiling surface, the flow channel was fitted with a 0.75-in.-thick by 6.0-in.-wide piece of Plexiglas. This also had the advantage that the entire flow field was visible. A groove was machined in the brass channel for a neoprene O-ring seal between the cover and the channel. Prolonged exposure of the Plexiglas to the hot ($\sim 215^{\circ}\text{F}$) demineralized water caused the Plexiglas to become opaque. This was subsequently circumvented by inserting a 5-mm-thick section of tempered Pyrex glass between the Plexiglas and the coolant. Careful control of the seal bolting and of the rate of heatup prevented breakage of the Pyrex insert. The insert also served (in the tests with refrigerant) to prevent Freon from contacting the Plexiglas. It has been reported^{12a} that long-term exposure of Plexiglas to Freon-113 causes "crazing" and loss of optical qualities. [Our preliminary compatibility tests of Plexiglas in Freon-113 did not confirm the reported observations.] Figure IV-9 is a photograph of the flow channel assembly and the mounting plate (before machining the holes for the heat flux meters). The "honeycomb" aluminum flow straightener (replaced later with stainless steel "mist separator" to overcome a severe corrosion problem) and the bulk stream thermocouples are also shown in place.

The flow channel was mounted in a section of 8-in. structural steel channel to prevent thermal distortion. A 1-in.-thick layer of Fiberglas insulation was placed between the backup structure and the flow channel. Provisions were made to permit mounting the channel upside down for future downflow experiments.

PHOTO 4110-73

Fig. IV-9. Flow channel assembly.

Although photographing of the heat flux meter surfaces during boiling was not a necessary element of this research, provisions were made to take high quality, selected photographs of boiling for future reference. A Polaroid camera with strobe flash (1/20,000-sec duration) was used for this purpose.

Reservoir-Head Tank

The reservoir-head tank, made from a 55-gal stainless steel drum, was fitted with a 4.7 kW immersion-type electric heater to maintain the fluid at saturation temperature. The tank was vented to atmosphere through a small reflux condenser, connected to the top of the reservoir by a section of glass pipe. Assurance that saturated conditions were being maintained was obtained from visual observation of the reflux condensate flow rate. A sight glass provided indication of the liquid level. A calibrated thermocouple immersed in the reservoir measured the fluid temperature.

The reservoir-head tank was elevated 4 ft to provide sufficient net positive suction head (NPSH) to avoid cavitation in the pumps. A 2-in. pipe was used for the pump suction line to minimize friction losses.

Return flow to the reservoir was discharged beneath the free surface to eliminate entrainment of noncondensable gases. The pump bypass entered the reservoir tangentially to promote mixing and to eliminate thermal stratification in the tank.

Pumping Station

Two small centrifugal pumps were piped in a parallel flow configuration to provide the desired range of test channel flow rates. The pumps, Chempump Model SK-2/4-2/4S, were rated for 10 gpm at a developed

head of 40 ft of fluid with very low NPSH requirements. Valves were installed in the pump discharge lines to prevent recirculation when the test flow range required only one of the pumps to be operated. Bourdon pressure gages were also connected to the pump discharge lines to provide an indication of the pumping rate (from the pump performance curves). Suction pressure and temperature indicators were installed to maintain a check on incipient cavitation.

A bypass line was connected from the pump discharge header to the reservoir to divert the excess flow and to improve flow stability. The bypass line contained a turbine flow meter and a 10-micron filter station to assure particle-free coolant.

Preheater

A preheater section, consisting of three separately controlled, immersion-type electric heaters (each rated at 4.7 kW) inserted in sections of 1 1/4-in. nominal pipe, provided sufficient heat input to maintain saturation conditions at the heat flux meters. By locating the major flow restriction (the flow control valves) downstream of the preheater, the absolute pressure in the preheater was sufficient to prevent boiling.

The heaters were manually controlled to provide saturated, bulk fluid conditions at the heat flux meters.

Trim-Cooler

A trim-cooler was located downstream of the preheater in the event that stable conditions could not be maintained by manual control of the preheater. This proved not to be the case, however, over the range of test conditions.

Since Freon-113 boils at 117.6°F at atmospheric pressure and the pumps supplied more than adequate heat to the circulating refrigerant to maintain saturation conditions, this cooler was available to remove the excess pump heat and, thus, minimize boiloff and loss of the refrigerant from the reservoir.

The cooler consisted of 10 ft of 1/4-in.-OD copper tubing (nickel plated) coiled inside a section of 1 1/2-in. pipe. Connections for a Heise pressure gage and a calibrated thermocouple were located near the exit of the cooler to determine the thermodynamic state conditions of the coolant at that point.

After-Cooler

At the higher flow rates, the two-phase pressure drop in the return line from the flow channel to the reservoir was such that fluctuating pressures were encountered at the heat flux meters. An after-cooler, consisting of 10 ft of 3/8-in.-OD copper tubing (nickel plated) coiled inside the 2-in. pipe, circumvented this operating difficulty. A calibrated thermocouple was located downstream of the cooler to monitor the return flow so that subcooled conditions could be avoided.

Flow Measuring Station

The desired range of flow velocities past the heat flux meters (1 to 5 ft/sec) and the dimensions of the rectangular flow channel (0.500 in. x 3.000 in.) dictated a flow measuring capability over the range of about 5 to 25 gpm. Since a single flow measuring device capable of accurate measurement over this range was not readily available, two suitably sized turbine flow meters were installed in a parallel flow

configuration. The flow control valves were located downstream of the meters to avoid flashing flow as a result of head losses in the meters.

The meters selected for use were an In-Val-Co Model W3/750 ($K_o = 585$ pulses per gallon) covering a flow range of 1.8 to 18 gpm, and an In-Val-Co Model W3/1000 ($K_o = 908$ pulses per gallon) covering a flow range of 6 to 60 gpm. The meters have an accuracy of $\pm 0.5\%$ in their flow ranges and $\pm 0.1\%$ repeatability. Recalibration runs were performed in a weigh tank system with the upstream and downstream piping connected to include the entrance-exit effects in the calibration. The results of this recalibration are given in Appendix A-2. The meters were installed (and calibrated) in vertical runs of pipe to minimize gravitational effects at low flow rates.

Instrumentation

Pressure. Pressure measurements were made with Bourdon-type gage at the pump suction, on the discharge lines of the two pumps, and downstream of the trim-cooler. Since the pressure downstream of the cooler was used to establish the thermodynamic state of the coolant, a 0 to 100 psi Heise gage (10 psi scale subdivisions, $\pm 0.1\%$ full-scale accuracy, temperature compensated over the range -25 to 125°F) was used at this position. The gage was certified for accuracy prior to use. Pressure tap connections were located at four places along the flow channel (Fig. IV-4): these were valved to a manifold which was connected to a 12-in. mercury manometer (12-in. span) and to a 100-in. water column referenced to atmosphere. By proper manipulation of the valves, the pressure could be read at the desired location in either inches of water or mercury,

depending on range. A temperature compensated barometer was used to measure atmospheric pressure.

Temperature. Temperature measurements used in the data analysis were obtained from 0.040-in.-OD stainless steel sheathed, Chromel-Alumel, grounded junction, thermocouples referenced to the ice point. A Leeds and Northrup, Model 7553, Type K-3, Universal Potentiometer (0.5 μ V sub-division, $\pm 0.015\%$ at 0.5 μ V accuracy) was used as the readout device. The potentiometer was certified for accuracy prior to use. A selector switch was used to connect the individual thermocouples in sequence to the potentiometer. Calibration of the thermocouples was performed against standards traceable to the National Bureau of Standards. Results of the calibration are given in Appendix A-3. The calibration included all elements of the temperature sensing instrumentation up to the terminals of the potentiometer. The calibration data were fitted to cubic equations, using the CRDEAL program,¹²⁹ to facilitate data reduction.

A few uncalibrated thermocouples, with a Brown recorder readout potentiometer, were used for monitoring loop operation.

Flow Rate. Calibrated turbine meters were used to measure the test channel flow rate. This was accomplished by counting the pulses generated per unit time interval by the meter. A Transistor Specialties, Inc., Model 401-R Universal Counter (1 count accuracy, 1 msec minimum time interval between pulses) was used to count the pulses. The volumetric flow rate (turbine meters are volume flow and not mass flow devices) was obtained from calibration data (see Appendix A-4), corrected for temperature effects on viscosity and meter housing thermal

expansion.¹³⁰ Mass flow rates were calculated from the measured volumetric flows.

A continuous readout flow indicator was also used for monitoring flow trends during startup and stabilization of conditions prior to each test run.

Liquid Level. An ordinary sight glass was used to indicate the coolant level in reservoir-head tank.

Heat Flux Meter Power. Input power to the heat flux meters was computed from voltage and current measurements. A DANA Model 4800 digital voltmeter [10 μ V to 1000 V dc full-scale input ranges, accuracy $\pm(0.005\%$ of reading + 0.005% of full scale)] was used to measure the dc voltage drop across the heater. [Voltage taps were attached to the ends of the heater.] With the aid of a selector switch, the digital voltmeter could be switched to measure the voltage drop across a calibrated shunt (10 amp, 0 to 100 mV, $\pm 0.5\%$ accuracy on current) in the dc power lead to the heater. The voltmeter and shunt were certified for accuracy by the Standards Laboratory prior to use. The results of the calibration are reported in Appendix A-1.

Electrical power measurements, corrected for heat losses, were used as a check on power measurements obtained by the temperature gradient method.

Pool Boiling Facility

The pool boiling tests were performed in a Pyrex glass boiler (11.5 in. ID by 18 in. high) fitted with a Plexiglas cover. A 2.5 kW heating element was incorporated in the support structure and served as the

primary heat source. The test fluid was maintained at saturation temperature by proper adjustment of a Variac control coupled to the heating element. Additional heating capacity (for faster heatup) was provided by a Variac-controlled, immersion heater. Fiber glass insulation minimized heat losses through the wall of the cylindrical boiler. The Plexiglas cover was constructed in three layers (with insulation between the lower two layers) to reduce heat losses through the top. A reflux condenser, fabricated in the form of a "pancake" from copper tubing, was suspended beneath the cover to condense the vapor and to prevent loss of the test fluid. Access was provided through the cover for four calibrated thermocouples; two were located approximately 1/2 in. below the vapor-liquid interface and the other two were located 0.18 in. above and slightly to the side of the test surfaces.

For the horizontal boiling surface tests, the mounting plate assembly (see Fig. IV-7) was supported and carefully leveled in the boiler. Figure IV-10 shows a photograph of the setup. Pyrex glass skirts (2.3 in. ID by 5.5 in. high) were positioned concentric with the 0.75-in.-diam test surfaces to minimize the effect of external convection currents generated by the hot plate. In addition to the internal details, the photograph shows the manner by which access was gained for the heat meter instrumentation and power through the cover plate assembly. A slight overpressure of buffer gas (nitrogen) was applied to the housings at all times to prevent inleakage of test fluid. The fluid level was maintained 5 to 6 in. above the test surfaces.

A dark glass background was placed behind the test surfaces to aid photography. The strobe light was positioned to the left for side

Fig. IV-10. Horizontal pool boiling arrangement.

lighting. [Back lighting was impractical due to the access tubes and optical distortion.]

For the vertical surface boiling tests, the mounting plate assembly was carefully aligned (in the same relative position as was used in the flow test facility) by means of a support frame as shown by the photograph in Fig. IV-11. A rectangular isolation chamber was attached to the front of the mounting plate to minimize the influence of external convection currents. The chamber, clearly shown in the photograph, had a cross section of 3 in. by 1.5 in. and was closed at the bottom. Thermocouples for local fluid temperature measurements were located at the center line of the heat meters; the center line distance between the two meters was 2.875 in. The fluid level was maintained 5 to 6 in. above the center line of the upper heat flux meter.

Procedures

Flow Tests

The flow tests (boiling and forced convection) were performed before the pool boiling experiments; water data collection preceded the Freon-113 experiments. The following discussion is primarily related to the water tests; the tests with Freon-113 deviated slightly due to the peculiarities associated with the use of a low vapor pressure (and expensive) fluid.

The reservoir was maintained at sufficient level (~40 gal total loop inventory) to permit operation for several days without addition of makeup. Boiling conditions were maintained in the reservoir continuously (except during prolonged shutdowns) by the reservoir heater

Fig. IV-11. Vertical pool boiling arrangement.

to provide deaeration. Heat input was adjusted to the point that a slight amount of condensate was refluxed from the vent condenser.

The pressure was set at the upstream heat flux meter (HM-1) to correspond to a specific saturation temperature (215°F for water, 122°F for Freon-113) by adjusting the throttling valve downstream of the test channel and/or by controlling the coolant flow to the aftercooler. The barometer was read every 3 or 4 hr (corrected for temperature and local gravity) prior to making adjustments. The coolant flow rate was maintained constant (indicated by a frequency meter monitoring the turbine output signal) throughout a given series of tests. Local pressure at the heat meters was indicated on 100-in. water (or Freon) manometers or by a 12-in. mercury manometer, depending on pressure range. The suppressed boiling tests required that the mercury manometer be used. The voltage to the line heaters was adjusted as required to maintain the saturation temperature corresponding to the specified pressure. The bulk stream thermocouple (TC-37) was used to indicate that the temperature was at the desired value. Occasional generation of bubbles at local flow disturbances provided a second indication of saturation conditions at the heat flux meters.

The taps for measuring the local pressure before and after the channel flow straightener and at the channel exit were unfortunately located in regions of hydrodynamic disturbances, thus negating their usefulness. Since these measurements were superfluous to the objective of this research, no attempt was made to relocate the taps; neither was any use made of these data. The tap at the heat meters, being in a hydrodynamically "smooth" region, provided data consistent with the measured temperatures.

A series of experiments showed that equilibrium was achieved in about 15 minutes after a step change was made in the heat meter power level, provided stable boiling conditions were developed. The heat meter thermocouples were monitored to determine when stable conditions prevailed: 30 to 60 minutes were usually allowed after a change before data were taken. Upon ascertaining that suitable conditions existed, a complete data set was taken. Data recorded included, in addition to the pressures, temperatures, and flows indicated on the flow sheet (Fig. IV-1), the power measurements to the heat meters and the auxiliary loop heaters, the heat flux meter thermocouple emfs, and when possible a sketch of boiling sites. Typical data sheets are included in Appendix D. Photographs were taken of most of the boiling tests for future reference. The raw data were reduced with the aid of computer routines. Usually, immediate data reduction was made by typing the data into a time-sharing computer terminal set up at the experiment for that purpose. This was a most beneficial arrangement, since the results of the previous experiment were available before the conditions of the next experiment were set.

It was also found that the boiling data were more consistent if a high heat flux was first applied to the heat meters to activate as many nucleation sites as possible and then the heat flux was reduced to the desired level for the particular data point. This is consistent with the observations of many investigators of boiling heat transfer, particularly with pool boiling experiments. Consequently, this pattern was followed in setting the conditions for each boiling data point.

The high void fraction Freon flow boiling tests were conducted by initiating bulk or homogeneous boiling (as the result of the pressure drop across the mist separator) at the channel inlet. This technique provided stable, two-phase-flow conditions and produced uniform bubbles in the stream. This was accomplished at a predetermined flow rate by adjusting the inlet fluid temperature to the desired level and then momentarily decreasing the flow rate to initiate nucleation, followed by restoration of the flow rate. The range of thermodynamic quality obtainable (2 to 8%) was limited by the maximum heat flux on the line heaters to avoid film boiling.

After establishing the stream conditions, the same procedures were followed to obtain heat transfer data outlined above for the zero-quality tests.

Pool Boiling Tests

The pool boiling tests with both horizontal and vertical surfaces were much easier to perform. Using the arrangement previously described, the boiler was filled to the proper level, heated to saturation temperature, and maintained at this temperature until the series was complete (two or three days). Preliminary tests showed that 30 minutes was adequate to equilibrate after a change in heat meter power level. Thus, the boiling tests were conducted by starting at the maximum power level and reducing in steps to the next lowest level. Data obtained at each power setting included barometric pressure (with corrections for local gravity and temperature), pool depth (to permit calculation of the local pressure and saturation temperature at the boiling surface), four bulk fluid temperatures, and the heat meter temperatures and power parameters.

The natural-convection data points were obtained by setting the desired power level in the evening and recording the data the next morning; thus 12 to 16 hr were available for the meters to equilibrate under these low heat flux conditions.

Boiling Surface Preparation

Originally, the test surfaces were plated with a 0.0001-in. thickness of pure nickel to prevent oxidation (often referred to as "aging" in the boiling literature) of the copper surface during the water tests. This nickel surface was then finished, by lapping with 1-micron (1200-grit) aluminum oxide compound, to give a mirror finish. (Photographs of the surfaces were shown in Figs. IV-4 and IV-7.) Since a moderately large number of nucleation sites was essential to insure stable and reproducible boiling, it appeared that the nickel-plated surface would be ideally suited. However, difficulties with surface contamination (caused by corrosion of brass fittings in the system piping) and non-reproducible nucleation in the water flow boiling tests necessitated further surface treatments to improve the nucleating characteristics. Accordingly, the surfaces were successively lapped with a 600- and 400-grit corundum (naturally occurring aluminum oxide) lapping compound in an effort to reactivate and to improve the performance of the nickel-plated surfaces; this was only marginally successful. [Lapping was selected over the use of emery paper to provide a more uniform surface finish.] In a final resort, the nickel plating was removed completely by polishing with 300-grit silicon carbide paper, followed by 600-grit paper, to expose the copper as the boiling surface. The copper surface was then lapped with 400-grit corundum to remove all scratches. This

preparation gave a surface that performed admirably; results are discussed in the next chapter. Thus, the final convective flow tests with water, all the Freon flow tests, and all the pool boiling tests (both water and Freon) were carried out with this surface preparation.

Prior to each series of tests, the copper surface was relapped (using a small cast iron lapping disk and 400-grit corundum compound), washed with Freon, and air dried. Examination of the lapped surfaces under a microscope (40X) showed that the mat finish was reproducible.

As will be discussed later, the lapped copper surface yielded pool boiling data indicative of a fairly rough surface when referenced to the bulk of the data reported for copper finished with dry 4/0 emery paper. Possible explanations for this somewhat surprising finding involve the type of surface produced on the relatively soft copper by the lapping operation and the possibility that aluminum oxide particles (from the lapping compound) remained embedded in the copper. This prompted some further examination of surfaces finished with dry emery paper and with 400-grit lapping compound.

One of the prototype meters used in perfecting the fabrication procedures (the copper came from the same bar as that used in the test meters) was given the same surface finishing treatment as used for the boiling test surfaces. Examination of this surface under the microscope indicated the same surface finish as had been produced numerous times on the test meters. The surface was then submitted for metallographic examination and surface roughness measurements. Following this, the surface was lapped and (reexamined) with 600-grit compound, and then successively with 400-grit silicon carbide paper (wet), 2/0, 3/0, 4/0

dry emery paper and 600-grit silicon carbide paper (wet), with the surface being examined between each treatment. This procedure is in order of decreasing particle size, with wet (water lubricated) paper producing smoother surfaces than dry emery paper. [Water cannot be used as a lubricant with emery paper as it dissolves the adhesive bonding the particles to the support paper.] The surface finish was generated by placing the paper on a surface plate and rubbing the prototype meter over the paper in parallel strokes. About 10 strokes were sufficient to remove completely the marks from the previous polishing (perpendicular to the direction of the strokes); each stroke was made on a new area of the paper, since worn paper burnishes the surface and produces a smoother finish. The above is the standard technique for preparing metallographic specimens for examination.

Photomicrographs were taken of the surfaces; full-sized photographic reproductions (100X) are shown in Fig. IV-12 for the 400-grit lapped surface, the 2/0 and 3/0 dry emery finished surface, and the 600-grit wet silicon carbide finished surface. The photomicrograph of the 400-grit lapped surface confirms the absence of scratches in the prototype meter and, by inference, from the meters used in the boiling tests. The surface exhibits a mat-like finish with no confirmed embedded aluminum oxide particles within the area of view.

After the photomicrographs were taken, the arithmetic average surface roughness was measured a number of times (perpendicular and parallel to the direction of polishing) for each surface finish with a calibrated Brush Surfindicator, Model B-110 (0.0006-in. radius diamond stylus). The arithmetic average roughness in microinches (RMS values would be 11%

(a) 400-grit lapped

(b) 2/0 emery polished (dry)

(c) 3/0 emery polished (dry)

(d) 600-grit silicon carbide polished (wet)

Fig. IV-12. Photomicrographs of lapped and polished surfaces (100X).

greater) were:

<u>Lapped Surfaces</u>		<u>Emery Polished Surfaces</u>		
<u>400 grit</u>	<u>600 grit</u>	<u>2/0</u>	<u>3/0</u>	<u>4/0</u>
20	13	11	5	7

The surface roughness measurements were highly reproducible and essentially independent of polishing direction. Although the roughness of the 3/0 and 4/0 finishes appear reversed, both were rechecked and confirmed. It is noted that the 2/0, 3/0, 4/0 paper each came from a different manufacturer; this probably accounts for the inversion in standings for the last two listings.

Two specimens of the same copper bar were prepared as before (one by lapping with 400-grit compound; the other by polishing with dry 2/0 emery) and cast in epoxy for sectioning to obtain profiles of the surface finish. The specimens were handled in accord with standard metallographic techniques to preserve the surface contour. Full-sized reproductions of photomicrographs (500X) of the surface contours are shown in Fig. IV-13; the dark area is the epoxy and the light area is the copper. These photomicrographs show the high irregularity of the nucleating surface with numerous cavities within which vapor can be trapped.

To probe the possibility that aluminum oxide particles embedded in the relatively soft copper might be the cause of the outstanding nucleating performance of the test meters, another specimen was lapped with the 400-grit compound as before and examined by an electron scanning microscope (ESM). This device has a very large range of magnification and depth of field for focusing compared to an ordinary optical microscope. The specimen surface was scanned across two diameters in a

(a) 400-grit lapped surface

(b) 2/0 emery polished surface

Fig. IV-13. Profiles of lapped and polished surfaces (500X).

0.035-in.-wide band (12% of the total area) to determine if particles were present and, if so, the approximate number per unit area. Several particles (about 50 in each scan) were found and identified by spectral analysis to be aluminum oxide. The particles were mostly on one side of the copper area, possibly where the lapping stone was first placed or where it was removed. Full-size reproductions of the micrographs of one interesting cluster of particles are shown in Fig. IV-14. The area shown in Fig. IV-14a is approximately 0.03×0.03 in.; this micrograph can be contrasted to the one shown in Fig. IV-12a, since they are of identically prepared surfaces. The depth of field within focus in the 100X magnification of the ESM scan clearly shows the presence of four aluminum oxide particles. Figures IV-14b, c, and d, showing enlarged views of the "large" central particle of Fig. IV-12a, also clearly exhibit the character of the surface roughness. Figure IV-14b (at 300X) is consistent with the surface profile shown (at 500X) in Fig. IV-12a. The particle shown in Fig. IV-14c appears to be about 0.0017 in. across the longest dimension.

Figure IV-15 shows full-size reproductions of a set of ESM images of the 400-grit lapped copper surface. The two traversing scans (each covering a strip 0.035 in. wide by 0.750 in. long) indicated these areas to be far more representative of the general surface condition than that shown in Fig. IV-14a. Thus, it was concluded that embedded aluminum oxide particles had an insignificant effect on the boiling performance relative to the effect of surface roughness produced by the lapping.

By way of comparison, the ESM image of Fig. IV-15a (100X) shows far more detail of the surface than that given by optical means (see

(a) 100X

(b) 300X

(c) 1000X

(d) 3000X

Fig. IV-14. ESM micrographs showing aluminum oxide particles on lapped surface.

PHOTO 0254-74

(a) 100X

(b) 500X

Fig. IV-15. ESM micrographs of 400-grit lapped copper surface.

Fig. IV-12a); the surface is vividly portrayed at 500X in Fig. IV-15b. The profile view shown at 500X in Fig. IV-13a can be compared to the surface image (at 500X) in Fig. IV-15b.

These findings strongly suggest that the use of a surface roughness measurement to characterize a boiling surface finish is somewhat questionable. This undoubtedly partially explains the cause of differences in boiling performance as reported by various investigators on surfaces ostensibly prepared alike. Views of a surface by the electron scanning microscope clearly display the character of the surface. The use of this instrument may thus provide an improved technique for studying the relationship between the boiling surface and the boiling performance.

CHAPTER V

RESULTS AND DISCUSSION

Heat Flux Meter Thermal Performance

It is of interest to compare the thermal performance of the heat flux meter with the expected behavior on which the design was based (see Chapter IV and Appendix B). The results of a typical flow boiling experiment were used as input parameters to the HEATING-3 computer program¹²⁷ to determine how well the computational model would predict the experimental data. Run TPBWC2-01 (upflowing water boiling on a copper surface) was selected from the flow boiling experiments for this purpose, since the heat flux and the boiling heat transfer coefficient were comparable to those used in the design analysis (Appendix B).

The measured electrical power and stream temperature and the boiling coefficient calculated from the heat flux and the temperature difference measurements were input to the HEATING-3 computer program¹²⁷ to describe the experiment in terms of the calculational model. Natural-convection heat transfer to water [Eq. (B-1) of Appendix B with $T_b = 214.81^\circ\text{F}$] and to air [Eq. (B-2) of Appendix B with $T_b = 80^\circ\text{F}$] was specified for the stainless steel flange surrounding the copper surface area and the exposed surface area of the meter housing, respectively. The resulting calculated center-line temperature within the meter is compared in Fig. V-1 with the five experimentally determined temperatures and a least-squares linear fit to these experimental values. The agreement between the predicted and measured temperature profiles is extraordinarily good.

Fig. V-1. Comparison of HM-2 performance with predicted behavior.

A further comparison of the HEATING calculation and the experimental data is given in Table V-1. As noted from the table, the calculated boiling surface temperature is in remarkable agreement with the extrapolated temperature obtained from a linear least-squares fit of the measured temperatures (corrected for thermocouple calibration); the fitting function is:

$$T = 296.18 + 17.6 X \quad , \quad (5-1)$$

with a standard deviation of 10.1 °F. The deviation in the intercept (surface temperature) is 10.1 °F and in the gradient, 17.6 °F/in. The measured flange temperature is also in excellent agreement with the calculated value, indicating the effectiveness of the thin stainless steel flange and of the insulation as heat barriers. This is supported further by the agreement in the heat balance comparisons given in the table. As shown by such results, the design selected for the heat flux meters proved to be capable of providing highly reliable data.

Boiling Surface Performance

In attempting to separate nucleation and forced-convection effects in flow boiling, the convective studies were to be made on a surface having known pool boiling characteristics. It was recognized that good nucleating performance with a nonoxidizing surface would be required to avoid the "aging" phenomena observed by a number of previous researchers. Based on the experience of others,^{112,113,124,125} a highly polished nickel-plated surface was initially selected to achieve the desired results. However, corrosion of a few brass pipe fittings in contact with saturated water in the early flow boiling tests caused contamination

TABLE V-1

VERIFICATION OF HM-2 HEAT METER DESIGN USING RUN TPBWC2-017 RESULTS

Parameter	HEATING Results	Experimental Results
Electrical power (W)	49.4 ^a	49.399
Boiling heat transfer coefficient (Btu/hr·ft ² ·°F)	2361 ^a	2361
Stream temperature (°F)	214.81 ^a	214.81
Average boiling heat flux (Btu/hr·ft ²)	50,755	50,451
Test surface temperature (°F)	236.38	236.18 ^b
Temperature at TC-1 (°F) ^c	245.07	244.42 (244.57) ^d
Temperature at TC-2 (°F) ^c	252.17	251.51 (251.36) ^d
Temperature at TC-3 (°F) ^c	259.70	258.72 (258.53) ^d
Temperature at TC-4 (°F) ^c	267.10	265.33 (265.57) ^d
Temperature at TC-5 (°F) ^c	274.36	272.60 (272.55) ^d
Flange temperature at TC-6 (°F) ^c	215.15	215.09
Fraction of heat through test surface (%)	92.38	91.71
Fraction of heat through SS flange (%)	2.78	e
Heat loss to surroundings (%)	4.84	4.75 ^f
Unaccounted heat loss (%)	0.0	3.54

^a Input parameter to HEATING model.^b $T_{x=0}$ from fitted equation: $T = 236.1793 + 18.6213 X$.^c See Fig. IV-3 and Table IV-1 for location.^d Temperature in parentheses from fitted equation.^e Not determined; included in unaccounted heat loss.^f Determined from heat loss calibration equation [see Appendix A, Eq. (A-12)].

of the nickel-plated boiling surface; and efforts to overcome the effects of this corrosion were insufficient to produce a surface with the requisite nucleating characteristics. As discussed in detail in Chapter IV, the nickel plating was removed to reexpose the underlying copper surface. The copper was then finished by lapping with 400-grit corundum, yielding a surface with excellent nucleating characteristics (as discussed later).

The nickel-plated surface (after treatment to overcome the contamination problem) performed erratically in water with no more than 10 to 12 nucleating sites even at the highest heat fluxes and, consequently, showed considerable scatter and poor reproducibility in the data. In contrast the copper surface provided copious sites (at much lower heat fluxes) and very good data reproducibility. With copper it was difficult to count the active boiling sites, except from photographs taken under stroboscopic lighting conditions during boiling operation. Although the copper surface thus prepared fulfilled the requirement for a surface of good nucleation characteristics for investigating convective boiling, performance during pool boiling indicated that a smoother surface would have produced data in better agreement with the bulk of the published literature. Even so, the data were consistent and agreed well with published data for surfaces of comparable finish. [Metallographic examination of identically prepared surfaces after completion of the boiling experiments provided exact details of the surface finish. See discussion and Figs. IV-13, IV-14, and IV-15 in Chapter IV.]

Pool Boiling

Data were taken with Freon-113 and with demineralized, deaerated water for pool boiling on both horizontal and vertical copper surfaces at atmospheric pressure, using the apparatus described earlier. Heat transfer regimes ranged from natural convection to fully developed nucleate boiling. The reduced data are tabulated in Tables D-1 through D-16 in Appendix D. In reducing the data, the measured (and corrected for calibration) temperatures were fitted with a linear least-squares routine from which the boiling surface temperature was obtained by extrapolation. A typical example was shown in Fig. V-1. The boiling heat flux was computed from the gradient of the fitted function. In general, the boiling flux accounted for about 90 to 98% of the measured electrical power input; at the very low heat fluxes, the boiling heat flux decreased to about 75 to 80% of the input. Nucleation was never observed on the stainless steel flange surrounding the copper boiling surface.

Typical photographs of vertical pool boiling at very low heat fluxes are shown in Fig. V-2a for Freon-113 (experiment VPEFC2-15, $q'' = 2359$ Btu/hr·ft², $\Delta T = 17.31^\circ\text{F}$) and Fig. V-2b for water (experiment VPBWC2-10, $q'' = 11,345$ Btu/hr·ft², $\Delta T = 10.71^\circ\text{F}$). The heat flow through the boiling surface for Fig. V-2a was 80.0% of the total input and for Fig. V-2b, 90.8%. The bubble column to the right side of the meters in Fig. V-2b arises from in-leakage of buffer gas (nitrogen) through the O-ring seal at the periphery of the lower meter. Note that the spherical shape and bubble diameter remain nearly constant for this stream while rising

(a) Freon-113, $q'' = 2359$, $\Delta T = 17.31$

(b) Water, $q'' = 11,345$, $\Delta T = 10.71$

Fig. V-2. Typical vertical pool boiling photographs.

(Note: Reference diameter of thermocouples is 0.040 in.)

through the saturated fluid; in contrast, the steam bubbles rising from the upper boiling surface grow in size and become rather distorted.

Results for pool boiling of Freon-113 at atmospheric pressure are shown in Fig. V-3 for boiling on vertical copper surfaces, and in Fig. V-4 for the same surfaces in horizontal orientation. In the vertical arrangement, the data for the upper meter (HM-2) were taken with no power on the lower meter (HM-1). The figures indicate little, if any, difference in the characteristics of the two meters. Further, Figs. V-5 and V-6, depicting the boiling coefficient as a function of the heat flux, indicate for this study insignificant effects of surface orientation. Linear least-squares regression analysis¹²⁹ of the fully developed nucleate boiling data (on a log-log plot) gave values tabulated below for $\underline{m}$ in the relationship $q'' \propto \Delta T_{\text{sat}}^{\underline{m}}$.

Values of $\underline{m}$ for pool boiling of Freon-113

<u>Orientation</u>	<u>HM-1 Data</u>	<u>HM-2 Data</u>	<u>Combined Data</u>
Vertical	7.43	10.20	7.73
Horizontal	9.06	9.20	9.07

These values are considerably higher than normally associated with pool boiling of ordinary fluids but are in apparent agreement with published data;¹²⁵ it is known that $\underline{m}$ can be altered from 3 to 24 by increasing the surface roughness.

Relatively few data were found in the literature for pool boiling of Freon-113, the oft referenced data of Corty^{42,125} being the exception. By way of comparison, the data of Fig. V-4 are replotted in Fig. V-7 with the three data sets Corty obtained for a horizontal copper boiling

Fig. V-2. Pool boiling with Freon-113 on vertical surfaces.

Fig. V-4. Pool boiling with Freon-113 on horizontal surfaces.

surface finished with 4/0 emery. Clearly, the functional relationship between q'' and ΔT_{sat} is nearly the same for the two sets of data. The noticeable difference is the displacement of the present data toward lower values of ΔT_{sat} . This is a direct manifestation of the "rough"

Fig. V-7. Comparison of Corty's Freon-113 data with data obtained in this work.

of about 13,000 Btu/hr-ft²; the levels were increased on the second and third days to the maximum permitted by the available power supplies. The data acquired on the first day were at slightly lower temperature differences than the subsequent data. While "aging" is a common occurrence with many fluid-metal combinations, this limited data was the only evidence of such found in this work with Freon-113. The meters had been used for many weeks in the flow boiling experiments that preceded the pool boiling tests without any indication of surface aging. Further, the several days of operation in the vertical pool boiling arrangement provide no additional evidence of aging.

Except for the single instance discussed above, the pool boiling performance of the copper surface exposed to Freon-113 was exceptionally stable and reproducible; the same cannot be said for the horizontal pool and the flow boiling experiments with water. Although the water was boiled continuously during the pool boiling experiments and should have been thoroughly deaerated (most investigators consider two hours of vigorous boiling adequate to deaerate water prior to data collection), the surfaces shown in the horizontal pool arrangement definite indications of the "aging" (surface oxidation) phenomena encountered by previous researchers.

Pool boiling data collected for water boiling on vertical copper surfaces are graphed in Fig. V-7; again the data for EM-1 were taken with EM-1 off to eliminate the effect of downstream bubbles. These data, taken over a period of three days with saturation conditions maintained continuously, show no effect of surface aging. This is to be contrasted with the data of Fig. V-7 for pool boiling of water on a freshly lapped,

Fig. V-8. Pool boiling with water on vertical surfaces.

horizontal copper surface taken over a two-day interval where a definite aging effect appears present. [Note the definite displacement to the right in Fig. V-2 for the second day's data.]

The data for the vertical orientation were taken after completion of the Freon tests (with vertical surfaces) without relapping the surfaces. Thus, the surfaces may have been "conditioned" sufficiently to

Fig. V-6. Pool boiling with water on horizontal surfaces.

prevent aging. Marcus¹² reported that for pool boiling of water on a freshly prepared copper surface, the boiling coefficient decreased over an initial period of 1 to 2 hr, gradually recovered over the next several hours, and reached an essentially stable value in about 10 to 15 hr. Although 10 hr of vigorous boiling (in the container) elapsed in the

present study before the first data point was taken in the horizontal orientation, this was apparently insufficient for stable conditions to be established.

As indicated with Freon, there is evidently little difference in the performance of the two meters. Slopes obtained by linear regression analysis of the horizontal pool boiling data taken on the first day were tested to determine if a statistically significant difference existed (at the 95% confidence level) between the data sets. This test showed that the data collected from the two meters could be considered the same.

Similarly to the analysis of the Freon data, least squares fits to the fully developed nucleate pool boiling data for water gave the values tabulated below for n in the relation $q'' \propto \Delta T_{\text{sat}}^n$.

Values of n for pool boiling of water

<u>Orientation</u>	<u>HM-1 Data</u>	<u>HM-2 Data</u>	<u>Combined Data</u>
Vertical	3.65	6.34	4.69
Horizontal	6.88	6.16	6.14

These values are somewhat high, compared to the usually reported values of 3.5 to 4.5 for pool boiling of water. Further discussion of this is deferred until a few paragraphs later.

Figures V-10 and V-11, showing boiling heat transfer coefficients for water as a function of heat flux for the two meters, indicate somewhat higher ($\approx 10\%$) boiling coefficients on the vertical surfaces than on the horizontal surfaces; this contrasts with the results for Freon-113. Marcus¹⁰⁹ reported pool boiling coefficients on a vertical surface to be

Fig. V-10. Water pool boiling coefficients for HM-1 in horizontal and vertical planes.

about 20 to 25% higher than the corresponding coefficient for water boiling on a horizontal copper surface; his data, shown in the figures for comparison, are lower than those determined in this research. Marcus¹⁰⁹ did not report details of his surface preparation other than "... freshly lapped, polished, and cleaned," and "... the boiling surface was

Fig. V-11. Water pool boiling coefficients for HM-2 in horizontal and vertical planes.

polished and cleaned with acetone, methyl alcohol, and distilled water." However, his data indicate a smoother surface than used for the experiments reported herein.

Kurihara^{70,131} carried out a thorough investigation of the effects of surface finish with water (and other fluids) in saturated pool boiling at atmospheric pressure on a horizontal copper surface for a full

range of surface finishes (4/0, 3/0, 2/0, 1/0, No. 1 and No. 2 emery, and 140-mesh carborundum, in order of increasing roughness). The horizontal pool boiling data for water collected in this work can be compared with Kurihara's data to establish a relationship between the 400-grit-lapped and emery-polished surfaces, since both experiments employed the same general design concepts. To perform this comparison, equations were fitted (using a least-squares routine) to Kurihara's fully developed nucleate boiling data; plots of the fitted equations are given in Fig. V-12, with the comparable data of this research. By inspection the data acquired on the second day (presumably after stable conditions were established) compare favorably with Kurihara's data for a 2/0 emery finish; for this reason the 2/0 emery finish was compared to the 400-grit lapped finish in detail in the metallographic investigation discussed in the preceding chapter. Further data comparisons are also repeated in the figure: Marcus¹¹⁰⁹ horizontal data of uncertain surface finish are shown along with the data of Nagarajan^{84, 124} for a surface polished with 600-grit silicon carbide paper and buffed. [600-grit is approximately equivalent to 4/0 emery particle size and for this reason Nagarajan's data are in good agreement with Kurihara's 4/0 data.]

From the pool boiling data it is concluded that the steep q' vs ΔT_{sat} curves (indicated by the previously tabulated values of $\underline{m}$) result directly from the roughness of the boiling surface. This conclusion is also supported by the metallographic investigation of the boiling surface discussed in Chapter IV with the rough surface (on a microscopic scale) providing superior nucleating characteristics for both Freon and water.

Fig. v-12. Comparison of data obtained by several investigators with pool boiling of water on horizontal copper surfaces.

The experimental boiling surfaces were arranged to permit the lower heat flux meter (in the vertical orientation tests) to be used as a bubble generator while measuring on the upper meter the effect of downstream bubbles on the upper surface's boiling performance. It was thought that this would provide insight into the mechanisms of convective flow boiling.

The results of such an experiment are graphed in Fig. V-13, wherein the vertical surface data of Fig. V-3 for Freon boiling on HM-2 (with HM-1 off) are replotted for comparison with data obtained for HM-2 with constant HM-1 flux levels of 14,850 and 32,100 Btu/hr-ft². [The bubble flow rate, or void fraction, is a direct function of HM-1 flux.] The figure indicates a slight increase in the temperature difference (decrease in boiling coefficient) at the higher heat fluxes to sustain a given heat flux on HM-2 in the presence of bubbles generated upstream and an appreciable decrease (increase in coefficient) at the lower flux levels as a result of enhanced convection.

Related data are given in Fig. V-14, wherein constant flux performance of HM-2 is graphed as a function HM-1 flux, or, in other words, as a function of upstream bubble generation rate. Two flux levels are shown for each of the fluids tested. For Freon, there appears to be no effect on the measured HM-2 boiling coefficient of bubbles (generated upstream) in the fluid layers near the wall. On the other hand, there is a small but discernable decrease in the measured coefficient for water. This difference in behavior is believed to be directly related to the size of the bubbles. Although measurement of bubble diameters was outside the scope of this investigation, qualitative observations indicated the water vapor bubble diameters were two to three times larger than the Freon vapor bubbles (see Fig. V-2). It appears plausible that the larger bubbles passing the boiling surface tend to sweep the growing bubbles away before they complete their growth cycle and, thus, reduce the amount of heat removed by each bubble. This will be discussed further with the flow boiling results.

Fig. V-14. Effect of HM-1 flux level on the constant flux performance of HM-2 in vertical pool boiling.

Since the Rohsenow³² and the Forster-Zuber³⁰ pool boiling correlations [Eqs. (2-23) and (2-30), respectively] are widely used in additive correlations^{12,24} for convective boiling, the fully developed nucleate boiling data were examined to determine the applicability of these correlations. Figures V-15 and V-16 show the results (Tables D-9 through D-16 in Appendix D tabulate the data) for the Rohsenow correlation for Freon and water, respectively. Values determined by least-squares analysis for the two correlating factors in Eq. (2-23), the surface-fluid combination factor, C_{sf} , and the bubble Reynolds modulus exponent, r , are tabulated in the figures. All the Freon data are well correlated in the least-squares sense (13.8% maximum error) with a C_{sf} of 0.00272 and an r value of 0.092; values for pool boiling of Freon were not found in the literature for comparison. The data for water exhibit considerably more scatter, reflecting differences between the two meters arising from surface finish, orientation (horizontal vs vertical), and aging. Considering all the water data as one set, a least-squares analysis gave values of 0.00898 for C_{sf} and 0.114 for r with a maximum error of 29.3%. The above values of C_{sf} and r were obtained by the procedure customarily followed in the boiling literature; alternative values obtained using the technically correct statistical procedures are discussed and compared in Appendix F.

Although it was originally suggested³² that r might have a value of 0.33 for most fluids, Vachon et al.³³ performed an extensive evaluation of reported values of these two parameters; they suggest better results might be obtained if r is permitted to take on different values. By way of comparison, Vachon et al. suggest $C_{sf} = 0.0097$ and $r = 0.10$ for water boiling on a copper surface finished with 2/0 emery. The line obtained

Fig. V-15. Rohsenow pool boiling correlation for Freon-11?

using these values is seen in Fig. V-16 to correlate the horizontal pool boiling data quite well also.

While the data collected in this research were correlated by the Rohsenow correlation,³² the Forster-Zuber relation³⁸ failed; this is

Fig. V-16. Rohsenow pool boiling correlation for water.

indicated by the graphed functions in Figs. V-17 and V-18 for Freon and water, respectively. Rohsenow et al.^{1,22} in the discussion of a paper authored by Zuber^{1,22} gave a lucid explanation of certain inherent shortcomings of the Forster-Zuber correlation and questioned its applicability.

Fig. V-17. Forster-Zuber pool boiling correlation for Freon-113.

Fig. V-18. Forster-Zuber pool boiling correlation for water.

Zuber determined the leading coefficient and the exponents in Eq. (2-30) from experimental data at maximum heat flux (near or at burnout) conditions; use of the parameters over the full range of nucleate boiling is therefore subject to question. Secondly, Rohsenow *et al.*¹²⁸ pointed out that $q'' \propto T_{sat}^2$ is implicit in Eq. (2-23), whereas the exponent on ΔT_{sat} in Eq. (2-30) is approximately 2. Then, using the data on which

Eq. (2-23) was originally based, they proceeded to determine new constants (dependent on surface-fluid combination) for Eq. (2-30). The equation with the new constants was shown to correlate nonpolar fluids but apparently would not correlate water and other polar fluids.

With this background and the fact that the exponent n in $q'' \propto \Delta T_{\text{sat}}^n$ was found to be of the order 4 to 6 for the boiling data collected in this investigation, it is not surprising that the Zuber expression failed to correlate the Freon data or, particularly, the water data.

Forced Convection

Most correlations of convective boiling heat transfer data require a knowledge of the single-phase forced-convective heat transfer coefficient. The experimental plan for this investigation (outlined in Chapter III) envisioned experimental determination of this coefficient for subsequent use in the data analysis. A considerable amount of such data, collected for various flow rates and heat flux levels with Freon-113 and water, are tabulated in Tables D-17 and D-18 of Appendix D. The Reynolds modulus ranged in the interval $1.1 \times 10^4 \leq N_{\text{Re}} \leq 9 \times 10^4$, and the heat flux was varied from 1000 to 35,000 Btu/hr-ft². The data at the higher heat fluxes were taken under suppressed boiling conditions to preclude nucleation. Plots of the data in terms of heat transfer coefficient versus heat flux are deferred for convenience until the discussion of flow boiling (see below).

The heated length of the heat flux meters was short in terms of L/D_{eq} (≈ 1.5), resulting in the forced-convection data being in the thermal entrance region rather than the fully developed region. Thus, the local Nusselt modulus was considerably greater than that predicted

by the Dittus-Boelter correlation [Eq. (2.2)]; this is in agreement with published data.¹³³ Eissenberg et al.¹³⁴ studied the Reynolds and Prandtl moduli dependence of the convective heat transfer coefficient in the turbulent thermal entrance region, using heat meters almost identical to those employed in this study. [The design of the meters used herein are second generation developments derived from the Eissenberg meters.] They studied demineralized water and aqueous-glycerol solutions up to 30 w/o glycerol, giving a range for the Prandtl modulus of 1.9 to 11.4 and a Reynolds modulus range of 10^4 to 2×10^5 . It was concluded that, in agreement with previous studies,¹³³ the exponent on the Reynolds modulus was reduced somewhat from the value for fully developed thermal conditions for heat transfer in the thermal entrance region and the exponent on the Prandtl modulus was essentially the same. The significant difference appears in the leading constant, being much greater for thermal entrance region effects. The Eissenberg correlating equation is given as:

$$N_{Nu} = 0.122 N_{Re}^{0.70} N_{Pr}^{0.36} \quad (5-1)$$

The unenhanced forced-convection data collected in this study are graphed in Fig. V-19; the correlating equation of Eissenberg et al. is shown for comparison. Note that the Prandtl modulus for water at the temperatures tested was about 1.7 and for Freon-113 about 6.9, well within the range investigated by Eissenberg et al. The individual surface-fluid data and the entire data set (excluding the enhanced convection data) were fitted with least-squares equations to determine the best-fit constants for use in equations of the form of Eq. (5-1); the constants determined are listed in the figure. The fitted equation for

Fig. V-19. Effect of Reynolds modulus on forced-convection heat transfer.

the total data set is also graphed in the figure. Although the experimentally determined constants differ slightly from Eissenberg's constants, it was concluded that an equation of the form (5-1) represented the data adequately for the intended use. Thus, in subsequent data analysis the forced-convection heat transfer coefficient was computed from the

expression ($\pm 16\%$ maximum error; 85% of data $\leq \pm 10\%$ error):

$$N_{Nu,conv} = 0.274 N_{Re}^{0.62} N_{Pr}^{0.35} \quad (5.2)$$

Flow Boiling

Convective flow boiling data were collected with both Freon-113 and demineralized (and deaerated) water under saturation conditions at atmospheric pressure. The heat meters were used with the flow test facility described earlier (Chapter IV) to obtain local two-phase boiling heat transfer coefficients in the vertical upflow mode over a range of flow and heat flux conditions. All the data reported and discussed herein were taken with the copper boiling surface prepared by lapping with 400-grit compound. Most of the data were collected under conditions of essentially zero percent thermodynamic quality; thus, the value of the reciprocal Martinelli parameter, $1/\chi_{tt}$, approached zero for these tests. A limited number of annular flow boiling tests were carried out with Freon over a range in the thermodynamic quality of 2 to 8% ($0.25 \leq 1/\chi_{tt} \leq 1.06$). Thermodynamic quality is defined as:

$$x = \frac{h - h_f}{h_{fg}} \quad (5-3)$$

where h is the local two-phase enthalpy, assuming thermodynamic equilibrium exists.

The reduced flow boiling data are tabulated in Tables D-19 through D-24 in Appendix D. Two-phase flow boiling data for Freon-113 are shown in Fig. V-20, where the local boiling coefficient for heat meter HM-2 is graphed as a function of the heat flux with flow velocity as a parameter. The data presented in the figure were obtained with zero

Fig. V-20. Results for flow boiling of Freon-113 without upstream bubbles.

heat flux on the lower meter (HM-1) to preclude the effect of upstream bubbles. It can be seen that, for heat fluxes greater than 8000 Btu/hr·ft², coefficients for the four velocities tested (note: 1 fps ≈ 325,000 lb/hr·ft²) all lie along the same line. At fluxes below 8000 Btu/hr·ft², incipient boiling and forced-convection prevail; and in this region, a minor dependence of the convective coefficient on heat flux was observed. Sufficient convective data were obtained at 2 and 4 fps to establish slopes for the curves in this region; since these were the same, surface roughness can be ruled out as the cause of the observed flux dependence. If roughness were the cause, the slope for the 4 fps data should have been greater than for the 2 fps data. Curves having the same slope as the 2 fps data were drawn through the 1 and 3 fps data in the convective region.

Data taken on the upper meter (HM-2) with bubbles in the stream (generated by constant flux operation of HM-1) are graphed in Fig. V-21; the enhanced convection data are included in the data tabulation of Table D-18 and the fully developed nucleation data are included in Table D-20 of Appendix D. The curves (without data points) from Fig. V-20 are shown also for reference in Fig. V-21 by the dashed lines. For heat flux values above 8000 Btu/hr·ft², the presence of bubbles in the stream has little, if any, effect on the local boiling coefficient; at lower heat flux values, the bubbles enhance appreciably (as expected) the convective heat transfer coefficient. These observations are consistent with the results obtained in the Freon pool boiling experiments with vertical orientation (see Fig. V-14).

Since the data graphed in Figs. V-20 and V-21 are for essentially zero thermodynamic quality, Chen's correlation²⁴ is not applicable; on

Fig. V-21. Results for flow boiling of Freon-113 with upstream bubbles.

the other hand, Rohsenow's additive technique¹² has no such restriction.

This correlation is given as:

$$q''_{TP} = q''_{NB} + q''_{conv} \quad , \quad (5-4)$$

where q''_{conv} is given by the Dittus-Boelter expression [Eq. (2-2)], and q''_{NB} is correlated with the Rohsenow pool boiling relation [Eq. (2.23)].

In analyzing the fully developed nucleate boiling data, an appropriate convection coefficient was calculated using Eq. (5-2). [The enhanced convection data points were not included in that correlation.] Next, the quantity of heat attributed to forced convection (q''_{conv}) was evaluated, based on $\Delta T_{sat, meas}$. Following this, Eq. (5-4) was used to obtain the nucleate boiling heat flux, q''_{NB} . The results of this analysis are graphed in Fig. V-22 (and tabulated in Table D-21 of Appendix D); clearly a high degree of correlation exists. Figure V-23 presents the data in terms of the Rohsenow pool boiling correlation. Using a linear regression routine, values of $C_{sf} = 0.00262$ and $r = 0.144$ were found to correlate the entire data set with 10.5% maximum error (27.7% if the one point for not fully developed nucleate boiling is included); these are to be compared with $C_{sf} = 0.00272$ and $r = 0.092$ (Fig. V-15) obtained for the entire Freon pool boiling data set. Blatt and Adt,⁵² using the Rohsenow additive method, correlated their Freon-113 flow boiling data (horizontal flow in 0.25-in.-ID copper tube, exit quality in the range 2 to 90%, mass flux in the range 28,800 to 54,000 lb/hr·ft²) with $C_{sf} = 0.007$ and $r = 0.33$. [See Appendix F for alternative C_{sf} and r values.]

Flow system design permitted operation with Freon under limited conditions of annular flow as described in the Procedures Section of Chapter IV. A series of such tests were performed in which the local (calculated) thermodynamic quality at HM-2 ranged from 2 to 8% vapor by weight;

Fig. V-22. Nucleate boiling heat flux contribution in Freon flow boiling.

Fig. V-23. Rohsenow correlation of Freon flow boiling results.

corresponding values of x_{tt} ranged from 4.1 to 0.94. In reducing the data, the Reynolds modulus was based on the liquid phase only; the results are tabulated in Table D-21 of Appendix D and graphed in Fig. V-24. Separation of the individual points in the plot according to the local quality did not appear justified, since the measured coefficient was not sensitive to this parameter in the range tested.

Chen's correlation²⁴ should be applicable under the conditions of this series of tests. Accordingly, the data were analyzed to determine if this method adequately predicted the experimental data. In applying the method of Chen, use was made of Eq. (2-39) and Figs. II-1 and II-2; the equation is repeated here for convenience:

$$h_{TP} = h_{macro} + h_{micro} \quad , \quad (2-39)$$

where h_{macro} is related to the Dittus-Boelter expression [Eq. 2-2)] and h_{micro} is based on the Zuber pool boiling correlation,³⁸ as discussed in Chapter II. As before, a convective heat transfer coefficient was computed, using Eq. (5-2), and multiplied by the proper F_{TP} value from Fig. II-1. The value for h_{micro} was obtained from the Zuber expression [Eq. (2-30)] and the appropriate suppression factor from Fig. II-2. The sum of these two coefficients, according to Eq. (2-39), is the local two-phase boiling coefficient as given by Chen's correlation and should compare favorably with the experimental values. Coefficients thus calculated are graphed in Fig. V-24 for comparison with the experimental data. In these calculations, qualities were varied at four levels between 2 and 8% (to correspond with the measured levels) at the lowest flow rate; at the highest mass flow, system limitations prevented operation with quality above the single value (2.1%) shown. Clearly, the correlation of Chen does not represent the data. This could have

Fig. V-24. Results for two-phase boiling of Freon in annular flow.

been anticipated, since the Zuber pool boiling expression did not correlate the Freon (and water) pool boiling data obtained in this research (see Figs. V-17 and V-18). This result suggests the need for a modification of the Chen correlation to include a more suitable nucleate boiling formulation.

As discussed in Chapter II, a number of investigators have divided flow boiling into two regimes: the nucleate boiling and the suppressed nucleation regimes, depending on the general behavior. That is to say, if the heat transfer phenomena is primarily characteristic of forced convection, it is referred to as suppressed nucleation; whereas if the heat transfer coefficient increases with flux (as in pool boiling), it is classified as nucleate boiling. These regimes can be recognized conveniently in a dimensionless plot of h_{TP}/h_{conv} versus the boiling modulus N_{Bo} [Eq. (2-12)]. On such a plot, in the region where the coefficient is independent of N_{Bo} , nucleation is suppressed. Since the heat transfer process in this regime is characteristic of forced convection, the dimensionless plot is quite similar to an ordinary plot of h versus q'' . In the region for which the coefficient shows a dependence on N_{Bo} , the heat transfer process is characteristic of nucleate boiling and appears so in the dimensionless plot. For constant flow, N_{Bo} varies approximately linearly with q'' , and the similarity of the dimensionless plot with the ordinary plot reflects the behavior of nucleate boiling. The annular flow data are graphed in this manner in Fig. V-25, where h_{conv} was calculated by Eq. (5-2). For N_{Bo} less than 1.6×10^{-4} , the dimensionless coefficient is insensitive to N_{Bo} (or flux), and nucleation was suppressed; above this value the coefficient depends strongly on flux

Fig. V-25. Boiling modulus vs h_{TP}/h_{conv} for Freon-113 in annular flow.

and, hence, is presumed to correspond to the nucleate boiling regime. Thus, the figure shows that most of the data of this study were collected under nucleate boiling conditions and that very few data points were in the suppressed nucleation region. This may provide an alternative reason for failure of Chen's correlation to predict the measured coefficient.

Pujol⁴⁵ investigated flow boiling of Freon-113 in the upflow mode (discussed in Chapter II) and correlated his results in terms of the Martinelli parameter, x_{tt} , and N_{Bo} [defined in Eqs. (2-3) and (2-12), respectively]. The results of this study are given in Fig. V-26 in a form similar to Fig. 19 of Ref. 49; the suppressed nucleation correlation of Pujol is shown also for reference. Plotted in this manner, the nucleate boiling regime is independent of x_{tt} (as indicated by the horizontal lines) and if extended to lower values of x_{tt} (higher qualities), would fair into the suppressed nucleation line. Except for the data at the lowest values of N_{Bo} , this figure also demonstrated that the annular flow results were obtained in the nucleate boiling regime. Too few data were collected under suppressed nucleation conditions to attempt a correlation; it is noted, however, that the group of data points with the lowest boiling moduli agree with Pujol's correlation.

The annular flow data were also analyzed using the Rohsenow additive technique in the same manner as described above for the very low ($\leq 0.2\%$ quality) flow boiling data; the results are tabulated in Table D-24 of Appendix D. The contribution to the total heat flux attributed to nucleate boiling is plotted in Fig. V-27 as a function of ΔT_{sat} ; again the correlation is good. Correlation of the fully developed nucleate

Fig. V-26. Ratio of h_{TP}/h_{conv} vs Martinelli parameter.

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Fig. V-27. Nucleate boiling heat flux contribution for boiling Freon in annular flow.

boiling data of Fig. V-27 ($q' \geq 6000$) in terms of the Rohsenow pool boiling expression is indicated in Fig. V-28. The data were correlated (5.8% maximum error) with $C_{sf} = 0.00276$ and $r = 0.192$ (see Appendix F), compared to values of 0.00262 and 0.144, respectively, for the very low quality flow boiling results. It was concluded that the Rohsenow additive technique also best described the annular flow boiling results.

The preceding discussion of Freon flow boiling was marked by the consistency and reproducibility of the data; the same was not the case with the water flow boiling experiments. While considerable effort was expended in deaerating the circulating system and maintaining it in that condition by continuous boiling in the large reservoir, the data still showed evidence of aging similar to that observed in the pool boiling experiments with water.

The reduced data for the water flow boiling experiments are tabulated in Tables D-20 and D-23 of Appendix D; the enhanced convection results are tabulated with the forced-convection data in Tables D-17 and D-18. The flow boiling data collected with meter HM-2 under conditions of very low local thermodynamic quality ($\leq 0.2\%$ vapor) and without upstream bubbles are graphed in Fig. V-29; when compared with a similar plot for Freon (see Fig. V-20), the data show the same general behavior but are less consistent. Thus, in the nucleate boiling region, significant separation of the data with velocity exists. While this contrasts with the Freon results, no consistent trend is observed. Data at 2 fps (corresponding to $\sim 430,000$ lb/hr·ft²) and 4 fps lie below and to the right of the values for 1 and 3 fps; limited data at 0.5 fps fall intermediate to the higher results. These discrepancies appear to be associated with surface aging. Indeed, it was difficult to operate for

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Fig. V-28. Rohsenow correlation of Freon annular flow boiling.

Fig. V-29. Results for flow boiling of water without upstream bubbles.

more than three successive days before severe degradation of the boiling performance necessitated reconditioning of the surface. In fact, the two groups of data points shown for the 2 fps line (distinguished in Fig. V-29 only by the two data points at each flux level) were actually obtained on successive days, with the higher values being recorded on the second day. After collecting these particular data, the surfaces were relapped, and the data points marked as "A" and "B" (at 2 fps) were recorded shortly after operation resumed. These values agree reasonably with the 1/2, 1, and 3 fps data collected subsequently.

Data taken with HM-1 operating at constant heat flux levels of 75,000 and 50,000 Btu/hr·ft² to produce bubbles in the stream flowing (at 2 fps) past HM-2 are graphed in Fig. V-30; the dashed curves are reproduced from Fig. V-29 for reference. With the exception of the points noted by "C" and "D", these data were taken alternately with the 2 fps data of Fig. V-29; i.e., a run with HM-1 at zero flux was followed immediately by runs with HM-1 at the higher fluxes. This should have minimized the effects of aging. Points "C" and "D" followed points "A" and "B" of the preceding figure. The data suggest for heat flux values above 40,000 Btu/hr·ft² that the bubbles cause a slight degradation in the local two-phase boiling coefficient; this result is consistent with the findings of the vertical pool boiling experiments (see Fig. V-14).

Of particular interest are the three data points recorded for a velocity of 1/2 fps with HM-2 operating at a constant flux level of 35,000 Btu/hr·ft² and HM-1 at three different flux levels. The sequence of this series of experiments was as follows: (1) a flux of 35,000 Btu/hr·ft² was established on HM-2 with zero flux on HM-1 to obtain a reference value, and (2) subsequent data were taken with HM-1 at successive

Fig. V-30. Results for flow boiling of water with upstream bubbles.

flux levels of 50,000, 35,000, and 65,000 Btu/hr·ft² to determine the effect of bubbles in the stream (from HM-1) flowing past HM-2 on the boiling performance of this (upper) meter. As noted in Table V-2 and graphed in Fig. V-31, the boiling heat transfer coefficient degraded as the number of bubbles in the stream increased. Photographs of the boiling during this sequence are presented in Fig. V-32. As observed in the photographs, moderate nucleation exists on HM-2 in the absence of upstream bubbles; then as the flux level on HM-1 increases, the number of nucleation sites on HM-2 decreases and essentially disappears at the highest flux value. These data are consistent with the pool boiling results for water on vertical surfaces discussed earlier (see Fig. V-14).

Based on the observed coefficients, this series of photographs might be interpreted as evidence that the contribution to the total heat removal process by bubble growth was relatively small; since the fractional loss

TABLE V-2
EFFECT OF UPSTREAM BUBBLES ON HM-2 BOILING PERFORMANCE

HM-2 Experiment No.	HM-1 Flux (Btu/hr·ft)	HM-2 Flux (Btu/hr·ft ²)	HM-2 Coefficient (Btu/hr·ft ² ·°F)	Heat ^a Balance
TPBWC2-002	0	34,905	1902	0.904
TPBWC2-032	34,984	34,880	1856	0.904
TPBWC2-037	50,209	34,767	1864	0.902
TPBWC2-033	65,228	34,832	1867	0.903

^a Fraction of heat input accounted for by the HM-2 heat flux.

Fig. V-31. Effect of upstream bubbles on constant flux performance on HM-2.

in coefficient was small. However, alternatively, the loss in heat removal due to bubble growth (while evading the question of how much) might be taken to be compensated by enhanced convection associated with the bubbles in the flowing stream. Since the study of bubble dynamics was outside the scope of this investigation, the information necessary to establish the correct interpretation is not now available.

Since the Rohsenow additive technique was shown to correlate the Freon flow boiling data well, the water flow boiling data were treated in a similar manner. Figure V-33 graphs the contribution of the total

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(a) $q''_{HM-1} = 0$; $h_{TP, HM-2} = 1902$ (b) $q''_{HM-1} = 35,000$; $h_{TP, HM-2} = 1856$ (c) $q''_{HM-1} = 50,200$; $h_{TP, HM-2} = 1864$ (d) $q''_{HM-1} = 65,200$; $h_{TP, HM-2} = 1867$

Fig. V-32. Boiling on HM-2 at 35,000 flux in the presence of upstream bubbles.

Fig. V-33. Nucleate boiling heat flux contribution in water flow boiling.

heat flux attributed to nucleate boiling as a function of ΔT_{sat} ; considerable scatter exists in the plot, unlike that obtained for Freon (see Fig. V-22). Careful study of the figure reveals fair consistency for the 0.5, 1, and 3 fps data collected with HM-2 and the 0.5 fps data from HM-1. It may be observed in Fig. V-33 that the data collected with HM-2 for 0.5, 1, and 3 fps and the 0.5 fps data from HM-1 are fairly well grouped and lie on approximately a (nearly vertical) straight line, indicating a degree of correlation between these particular data. Reference to Fig. V-29 shows that these data lie on approximately the same line in that figure, indicating a degree of correlation should be expected. Further, reference to the data for pool boiling of water on vertical surfaces (see Fig. V-8) shows good agreement between the data obtained from the two meters and a well-defined boiling curve. These data (Fig. V-8) did not indicate an aging effect and were found to be represented rather well by the Rohsenow pool boiling correlation. Thus, at least a portion of the scatter exhibited in Fig. V-33 is attributable to the lack of a good correlation (with flow rate) in Figs. V-29 and V-30, presumably a direct result of aging.

In summary, failure of the Rohsenow additive method to correlate the water flow boiling data adequately is attributed to the oxidation of the boiling surface, more commonly known as aging, under long-term operation. It is believed, based on the success of the additive technique with the Freon data and the ability of the Rohsenow pool boiling expression to correlate the water pool boiling data, that the additive technique would also be shown applicable to flow boiling of water provided consistent data were available.

System limitations precluded annular flow tests with water.

Careful attention was given to design, construction, and calibration of the heat flux meters so that reliable data, having both precision and accuracy, would be obtained. As discussed in the first section of this chapter, the meters performed exceptionally well; the quantity of heat that passed through the test surface, as determined from the gradient (evaluated at the wall) of the linear least-squares equation fitted to the five interior temperature measurements, in general, amounted to 90 to 96% of the measured input at the higher fluxes and 75 to 80% at the lower fluxes. The standard deviation for the slope of the fitted temperature distribution was generally in the range ± 0.30 to $0.35^\circ\text{F}/\text{in.}$ for the higher heat fluxes and for the lower fluxes, ± 0.03 to $0.07^\circ\text{F}/\text{in.}$; heat meter HM-1, being slightly better insulated (see Appendix A), showed slightly lower standard deviations. Similarly, standard deviations for the boiling surface temperature (obtained also from the fitted temperature distribution) were in the range ± 0.25 to 0.35°F and ± 0.03 to 0.10°F for the higher and lower flux experiments, respectively.

An error analysis (see Appendix E) was performed to estimate the range of the maximum percentage error expected for the heat flux, the temperature difference, and the heat transfer coefficient for high and low flux operation with both water and Freon. The results (tabulated in Tables E-1 and E-2 of Appendix E) indicate that the flux could be in error as much as 15.1% at the lower fluxes, decreasing to 3.3% at the higher fluxes. Upper and lower limits of the errors in ΔT_{sat} were estimated to be 0.49 and 0.13°F , respectively, depending on flux level and temperature difference. Correspondingly, errors in the coefficient could

be 17.3% for the lower flux range, decreasing to 6.7% for the higher range. These estimates assumed maximum errors in all the parameters occurred simultaneously and were such to produce maximum errors in flux, temperature difference, and coefficient. Certainly, this is an unlikely situation; the most probable error is expected to be lower.

CHAPTER VI

SUMMARY, CONCLUSIONS, AND RECOMMENDATIONS

Two well-known correlations are widely used to predict convective boiling heat transfer results: (1) the method of Chen based on the Zuber pool boiling expression, and (2) an additive technique suggested by Rohsenow that makes use of the pool boiling relation proposed also by him. Since these two methods differ rather significantly, this research was undertaken to obtain an internally consistent set of data from which flow boiling could be separated into its constituent pool boiling and forced-convection components and which could be used to test the experimental results against the commonly used predictive techniques. To this end pool and flow boiling tests were performed with saturated Freon-113 and water at atmospheric pressure, using a copper boiling surface prepared by lapping with 400-grit compound. Based on the experimental findings and study of the published literature, the following conclusions may be drawn:

1. Convective flow boiling at atmospheric pressure with saturated Freon-113 was correlated with the Rohsenow additive method over a mass flux range of 325,000 to 1,325,000 lb/hr·ft² and a thermodynamic quality range of 0 to 8% ($0 \lesssim 1/x_{tt} \lesssim 1.06$). The experimental results were not predicted by the relationship suggested by Chen. Failure of the latter method was attributed primarily to the inability of the Zuber expression to provide adequate correlation of the pool boiling results. A possible contributing factor may be the fact that most of the flow boiling tests

in this investigation were performed in the nucleate boiling region and very few in the suppressed nucleation region.

2. Convective flow boiling with saturated water at essentially zero percent quality, spanning a mass flux range of 106,000 to 852,000 lb/hr-ft², was not correlated adequately by the Rohsenow additive technique, presumably due to excessive scatter in the data as a result of surface aging. However, the same general trends were exhibited by the water boiling data as observed for the Freon data; and the water pool boiling results were correlated reasonably well by the Rohsenow pool boiling relation. Based on these and other observations discussed, it is believed that the additive technique would be shown also applicable to flow boiling with water, provided consistent and reproducible data were available. Chen's method is not applicable to the quality range used (due to system limitations) in the water flow boiling tests.

3. Too few Freon flow boiling data were collected under suppressed nucleation conditions to attempt correlation; however, the data were shown to be in agreement with the results of Pujol.

4. The presence of thermally generated bubbles (produced by the lower heat meter) in the stream flowing past the upper test surface did not alter the boiling performance in the fully developed nucleate boiling region using Freon as the test fluid but appeared to degrade slightly the performance with water as the test fluid. In both cases, considerable enhancement was observed (as expected) for the convective and incipient boiling regions. These observations are consistent with the pool boiling results. Degradation of the boiling performance with water appears to be related to bubble size. Photographs of the water boiling process showed

that as the number of bubbles in the stream increased, the number of nucleation sites on the upper test surface decreased. It was conjectured that the free stream bubbles sweep the growing bubbles off the boiling surface before their growth cycle is completed, thus lowering the contribution of heat removal by bubble growth. Simultaneously, the free stream bubbles enhance the convective coefficient to partially compensate the loss suffered as a result of decreased (or eliminated) bubble nucleation. Since study of bubble dynamics was outside the scope of this research, sufficient information is presently unavailable to resolve the observation.

5. Saturated pool boiling with Freon was investigated with the same test surfaces in both horizontal and vertical orientations; the results showed no effect of surface orientation and were well correlated by the Rohsenow relation. Linear least-squares analysis of the fully developed nucleate boiling data gave a value of 0.00272 for the surface-fluid parameter (C_{sf}) and 0.092 for the bubble Reynolds modulus exponent (r). Published values for pool boiling with Freon were not found for comparison. The data were not correlated with the expression recommended by Zuber. The boiling heat flux was determined to be proportional to ΔT_{sat} raised to the seventh to the ninth power, higher than ordinarily encountered, but consistent with the oft-referenced data of Corty. Vapor bubbles generated on the lower meter and rising by natural convection (buoyancy) past the test surface produced little, if any, effect on the upper meter boiling performance in the fully developed nucleate boiling region but appreciably enhanced performance in the convective and incipient boiling regions.

6. Saturated pool boiling with water was investigated also with the same test surfaces in both horizontal and vertical orientation; in contrast with the Freon results, coefficients for vertical surfaces were about 10% greater than those observed for horizontal surfaces. This finding is in agreement with Marcus' results. The data were reasonably well correlated by the Rohsenow relationship, although surface aging effects in the horizontal pool boiling data caused some scatter. Linear least-squares analysis of the combined, fully developed nucleate boiling, data set gave a value of 0.0898 for the surface-fluid combination factor and 0.114 for the bubble Reynolds modulus exponent; these are in good agreement with currently recommended values for comparable surfaces. Zuber's recommended expression failed to describe the results. The boiling heat flux dependence on ΔT_{sat} was determined to be proportional to the 4.7 to 6.1 power; this range is somewhat higher than normally reported for very smooth surfaces but is in agreement with published data for comparable surface characteristics. Vapor bubbles buoyantly streaming past the test surface caused a slight degradation in boiling performance of the upper meter; this same effect was observed in the water flow boiling tests, in contrast with the Freon pool and flow boiling observations. Qualitative observation of bubble size indicated that the diameter of the water vapor bubbles was a factor of 2 or 3 greater than the diameter of the Freon vapor bubbles. [The Zuber pool boiling expression indicated a factor of 6 to 7.] This difference in bubble size is believed to be related to the deleterious effect on boiling performance observed in the water tests.

7. Based on the above results, the Rohsenow additive technique is recommended for use with Freon-113 under convective flow boiling conditions in the mass flux and thermodynamic quality ranges tested. Its use

at higher mass fluxes and qualities is probably justified. Although application of the Chen correlation is somewhat advantageous because of its direct calculational approach, the results discussed above suggest modification is needed to provide a more suitable nucleate boiling formulation. It appears particularly appropriate to include a surface-fluid combination effect, since this parameter has been shown by numerous investigators to be a significant factor in pool boiling. Another, perhaps less objectionable, deficiency relates to the fact that q'' is proportional to approximately the second power of ΔT_{sat} in the Zuber nucleate boiling formulation, whereas experience has shown that the third to fourth power is more commonly observed.

8. Due to the small relative heated length ($L/D_{\text{eq}} \approx 1.5$) of the heat meters, single-phase convective heat transfer was shown to be in the thermal entrance region. Measured (unenhanced) convective coefficients were in good agreement with published results for similar heat flux meters; however, they were represented slightly better by the relation:

$$N_{\text{Nu,conv}} = 0.274 N_{\text{Re}}^{0.82} N_{\text{Pr}}^{0.35} .$$

9. The surface finish produced by lapping with the 400-grit compound provided excellent nucleating characteristics. Although this was considered a necessary requisite to permit separation of nucleate boiling effects from convective heat transfer effects, in retrospect, a smoother surface would have produced pool boiling data in better agreement with the bulk of the published literature. Boiling performance measurements indicated the surface finish to be comparable to a 2/0 emery polished surface. Metallographic examination completed after conclusion of the boiling tests confirmed the surface characteristics and raised the question that surface roughness may not be the best indicator of boiling

performance. Micrographs obtained with an electron scanning microscope showed vivid details of the boiling surface; this device may offer an improved technique for detailed study of cavity size, shape, and distribution. To the knowledge of this investigator, this is the first time the electron scanning microscope has been used to characterize boiling surfaces.

Based on the experience gained and the findings of this research, the following recommendations are suggested for future research:

1. Although Chen's correlation has been recommended by a number of investigators as being the best available for convective boiling, it is based strongly on a pool boiling correlation of questionable applicability. The results of this research suggest that modification should be initiated to resolve certain inherent deficiencies.

2. There appears to be evidence that water vapor bubbles in the stream flowing past the boiling test surface affect the boiling performance deleteriously. This investigation should be expanded, perhaps in connection with the previous recommendation, since it could alter the present concepts of convective boiling.

3. It now appears possible to study cavity size, shape and statistical distribution on typical boiling surfaces, using the electron scanning microscope and surface profiles obtained by ordinary metallographic techniques. A fundamental study of these parameters in conjunction with measured boiling performance of surfaces of specified preparation should lead to a better understanding of the boiling phenomena as related to surface finish.

4. With regard to the equipment used and the data generated in this research, those data collected with water as the test fluid were more scattered than desired, due to aging effects. It is believed that the nickel-plated boiling surface offers the best opportunity for improved data collection with water. An improved flow system, permitting operation over a wide range of quality and incorporating techniques for ascertaining the local void fraction, would permit a more thorough evaluation of the existing correlations over the full range of two-phase boiling conditions, or alternatively, provide definitive data on which more realistic correlations could be based.

5. The heat flux meter used in this study has been demonstrated to be a well developed and highly reliable device for measuring local heat transfer coefficients. The conceptual design offers many opportunities for use in unusual situations. While this study used the same meters in both pool and flow test facilities, the design was somewhat complicated by the need for interchangeability. Experience suggests the use of separate meters for the different purposes. This would lead to simplification and eliminate some of the bothersome details required for multipurpose installations.

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APPENDICES

APPENDIX A

CALIBRATION DATA

A-1. Current Measuring Shunts

Calibration of the 0 to 100 mV shunts used to measure current to the heat flux meters was performed in the Y-12 Plant Standards Laboratory. The shunts were connected to a standard test console, and the dc current was set to known ($\pm 0.5\%$) values. A digital voltmeter (certified for accuracy) was used to indicate the potential across the shunt.

The data for shunt No. 1 are shown in Fig. A-1 and were fitted, using a least-squares routine, to the equation:

$$I = 0.097624 E , \quad (A-1)$$

with a standard deviation of 0.00475 amperes.

Similarly, the data for shunt No. 2, shown in Fig. A-2, were fitted to the equation:

$$I = 0.099759 E , \quad (A-2)$$

with a standard deviation of 0.00377 amperes. In spite of the good fit obtained, the accuracy of the fitted equations can be no better than that for the input current. Thus, the accuracy on current was taken as $\pm 0.5\%$. These equations were used to compute the heater current from the measured voltage drop across the calibrated shunts.

The DANA Model 4800 digital voltmeter used to measure (to the nearest 10 μ V) the voltage drop across the meter heaters and the calibrated shunts was certified to conform to the manufacturer's specifications [$\pm(0.005\%$ of the reading + 0.005% of full scale) on a five voltage

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Fig. A-1. Shunt No. 1 calibration.

Fig. A-2. Shunt No. 2 calibration.

scales (0.1 V, 1.0 V, 10 V, 100 V, and 1000 V) for 24 hr] by Le Oak Ridge National Laboratory Instrument Standards Laboratory.

A-2. Calibration of Turbine Flowmeters

Two turbine flowmeters, installed in a parallel flow path configuration, were used to measure the flow rate through the test channel. An In-Val-Co, Model W3/750, meter (nominal frequency range of 100 to 1000 Hz) was used for the lower range of flows, and a Model W3/1000 (nominal frequency range of 100 to 1000 Hz) was used for the higher flow range. Manufacturer's data indicate 50% overranging is acceptable. They were installed in vertical sections of pipe with 20 pipe diameters of straight pipe upstream to smooth the velocity profile entering the meters. The meters were guaranteed by the manufacturer to be accurate within $\pm 0.5\%$ in the "linear" flow range with $\pm 0.1\%$ repeatability. Since the meters had experienced considerable use prior to this research, they were refurbished and recalibrated, using a weigh tank system ($\pm 0.5\%$ accuracy) with the upstream and downstream piping attached. In this manner, the effects of the piping was included in the calibration.

Turbine meters measure velocity rather than mass flow. The output signal (voltage pulses) is proportional to flow velocity regardless of fluid, and for a given meter this proportionality is expressed as K_0 in pulses/gal. The coefficient must be determined by calibration. The data are usually plotted as K_0 in pulses/gal versus the logarithm of the ratio of measured frequency in Hertz to the viscosity in centistokes. Plotted this way the coefficient is universal (with respect to fluid) over the "linear" range. The "linear" range is defined as that range

over which the measured curve can be represented ($\pm 0.5\%$) by a constant value of K_0 . For greater accuracy, one can use the plotted curve to determine K_0 at the appropriate value of f/v . Fitting the calibration curve with a suitable function facilitates data reduction by computer. This approach was followed in this research.

The calibrations were performed with water at 52°F ($\nu = 1.239 \text{ cSt}$). The reduced data were fitted with linear functions, using a least-squares routine. Figure A-3 shows the calibration data for the W3/1000 meter, plotted on rectangular coordinates. For the "linear" range determined in this investigation, $150 \leq f/v \leq 600$ ($210 \leq f \leq 765$), the data fit the equation:

$$K_0 = 912.855 - 0.00978185 f/v , \quad (\text{A-3})$$

with a standard deviation of 1.282 pulses/gal. Since the frequency covered by the calibration is less than the nominal maximum guaranteed by the manufacturer (765 vs 1000 Hz) and since the value of K_0 is nearly independent of f/v over the "linear" range, it was assumed that Eq. (A-3) was applicable over the full range of frequencies. This is consistent with the manufacturer's data furnished with the meter.

Figure A-4 is a similar plot for the W3/750 meter. The "linear" range for this meter was determined from calibration data to be $120 \leq f/v \leq 1265$ ($150 \leq f \leq 1600$). The data were fitted over the range $120 \leq f/v \leq 485$ with the equation:

$$K_0 = 4650.33 - 0.179528 f/v , \quad (\text{A-4})$$

with a standard deviation of 10.79 pulses/gal, and for the range

Fig. A-3. Calibration data for flowmeter F-1.

$485 \leq f/v \leq 1265$ with the equation:

$$K_O = 4576.40 - 0.0268513 f/v, \quad (A-5)$$

with a standard deviation of 7.40 pulses/gal. The frequency range covered in this calibration confirmed the manufacturer's data that the meter is "linear" over the nominal frequency range (100 to 1000 Hz). In fact, it was demonstrated to be "linear" at 60% overrange.

In order to use the above equations at temperatures other than that at which the calibration is made, the viscosity (in centistokes) at the temperature of the fluid must be used to form the frequency-to-viscosity ratio. This automatically corrects for drag effects in the meter. A

Fig. A-4. Calibration data for flowmeter F-2.

geometric correction must be made also to account for the thermal expansion of the meter housing. This is simply a volumetric correction¹³⁰ and is given as:

$$F_T = \frac{1}{1 + 3\alpha (T - T_0)} \quad (A-6)$$

Substituting the appropriate values, Eq. (A-6) becomes for the temperature range $52 \leq T \leq 300^\circ\text{F}$:

$$F_T = \frac{1}{1 + 2.4876 \times 10^{-5} (T - 52)} \quad (A-7)$$

Using this correction, the appropriate turbine meter coefficient at operating temperature is given as:

$$K_T = F_T K_0 \quad (A-8)$$

where K_0 is obtained from Eq. (A-3), (A-4), or (A-5), using the temperature-corrected viscosity in the f/v ratio. An accuracy of 0.5% was assigned to the meters.

A-3. Thermocouple Calibrations

The thermocouples used for data analysis were calibrated, using standard techniques, by the Oak Ridge National Laboratory's Instrument Standards Laboratory. All 18 thermocouples (6 each installed in the finished heat flux meters and 6 individual thermocouples for use in various loop locations) were calibrated simultaneously against a reference standard traceable to the National Bureau of Standards. The entire circuit (i.e., thermocouple, ice-point reference junction, switches, and leads up to the potentiometer) were included in the calibration. The data are tabulated in Tables A-1, A-2, and A-3. Column 1

TABLE A-1

HM-1 THERMOCOUPLE CALIBRATION DATA (METER SER. NO. 2)

TC No.	Meter Position	Actual Temperature (°F) ^a	Tabulated ^b EMF (mV)	Measured EMF (mV)	EMF Error (mV)
7	TC-1	99.0061	1.497807	1.4948	0.003007
		149.7695	2.660598	2.6700	-0.009402
		200.8239	3.838130	3.8657	-0.027570
		251.3638	4.995476	5.0426	-0.047124
		301.9554	6.136223	6.2018	-0.065577
8	TC-2	99.0061	1.497807	1.4952	0.002607
		149.7695	2.660598	2.6694	-0.008802
		200.8239	3.838130	3.8636	-0.025470
		251.3638	4.995476	5.0406	-0.045124
		301.9554	6.136223	6.1985	-0.062277
9	TC-3	99.0061	1.497807	1.4957	0.002107
		149.7695	2.660598	2.6695	-0.008902
		200.8239	3.838130	3.8632	-0.025070
		251.3638	4.995476	5.0406	-0.045124
		301.9554	6.136223	6.1981	-0.061877
10	TC-4	99.0061	1.497807	1.5001	-0.002293
		149.7695	2.660598	2.6745	-0.013902
		200.8239	3.838130	3.8680	-0.029870
		251.3638	4.995476	5.0456	-0.050124
		301.9554	6.136223	6.2029	-0.066677
11	TC-5	99.0061	1.497807	1.4956	0.002207
		149.7695	2.660598	2.6694	-0.008802
		200.8239	3.838130	3.8631	-0.024970
		251.3638	4.995476	5.0393	-0.043824
		301.9554	6.136223	6.1983	-0.062077
19	TC-6	99.0061	1.497807	1.4994	-0.001593
		149.7695	2.660598	2.6749	-0.014302
		200.8239	3.838130	3.8699	-0.031770
		251.3638	4.995476	5.0475	-0.052024
		301.9554	6.136223	6.2057	-0.069477

^a1968 IPTS.^bNBS smoothed data for type K thermocouples.¹²¹

TABLE A-2

HM-2 THERMOCOUPLE CALIBRATION DATA (METER SER. NO. 3)

TC No.	Meter Position	Actual Temperature (°F) ^a	Tabulated ^b EMF (mV)	Measured EMF (mV)	EMF Error (mV)
21	TC-1	99.0061	1.497807	1.5001	-0.002293
		149.7695	2.650598	2.6741	-0.013502
		200.8239	3.838130	3.8676	-0.029470
		251.3638	4.995476	5.0448	-0.049324
		301.9554	6.136223	6.2017	-0.065477
22	TC-2	99.0061	1.497807	1.5007	-0.002893
		149.7695	2.660598	2.6761	-0.015502
		200.8239	3.838130	3.8709	-0.032770
		251.3638	4.995476	5.0491	-0.053624
		301.9554	6.136223	6.2070	-0.070777
23	TC-3	99.0061	1.497807	1.5011	-0.003293
		149.7695	2.660598	2.6755	-0.014902
		200.8239	3.838130	3.8692	-0.031070
		251.3638	4.995476	5.0467	-0.051224
		301.9554	6.136223	6.2041	-0.067877
24	TC-4	99.0061	1.497807	1.5016	-0.003793
		149.7695	2.660598	2.6774	-0.016802
		200.8239	3.838130	3.8725	-0.034370
		251.3638	4.995476	5.0512	-0.055724
		301.9554	6.136223	6.2096	-0.073377
25	TC-5	99.0061	1.497807	1.4960	0.001807
		149.7695	2.660598	2.6715	-0.010902
		200.8239	3.838130	3.8660	-0.027870
		251.3638	4.995476	5.0442	-0.048724
		301.9554	6.136223	6.2021	-0.065877
33	TC-6	99.0061	1.497807	1.4973	0.000507
		149.7695	2.660598	2.6728	-0.012202
		200.8239	3.838130	3.8674	-0.029270
		251.3638	4.995476	5.0456	-0.050124
		301.9554	6.136223	6.2035	-0.067277

^a1968 IPTS.^bNBS smoothed data for type K thermocouples.¹²¹

TABLE A-3

MISCELLANEOUS THERMOCOUPLE CALIBRATION DATA

TC No.	Loop Location	Actual Temperature (°F) ^a	Tabulated ^b EMF (mV)	Measured EMF (mV)	EMF Error (mV)
34	Reservoir	99.0061	1.497807	1.4959	0.001907
		149.7695	2.660598	2.6696	-0.009002
		200.8239	3.838130	3.8617	-0.023570
		251.3638	4.995476	5.0393	-0.043824
		301.9554	6.136223	6.1973	-0.061077
35	Pump Suction	99.0061	1.497807	1.4992	-0.001 93
		149.7695	2.660598	2.6708	-0.010202
		200.8239	3.838130	3.8614	-0.023270
		251.3638	4.995476	5.0375	-0.042024
		301.9554	6.136223	6.1967	-0.060477
36	Upstream of Flow Meters	99.0061	1.497807	1.4992	-0.001393
		149.7695	2.660598	2.6711	-0.010502
		200.8239	3.838130	3.8618	-0.023670
		251.3638	4.995476	5.0375	-0.042024
		301.9554	6.136223	6.1990	-0.062777
37	HM-1 Bulk Stream	99.0061	1.497807	1.5026	-0.004793
		149.7695	2.660598	2.6805	-0.019902
		200.8239	3.838130	3.8770	-0.038870
		251.3638	4.995476	5.0578	-0.062324
		301.9554	6.136223	6.2191	-0.082877
38	HM-2 Bulk Stream	99.0061	1.497807	1.5003	-0.002493
		149.7695	2.660598	2.6753	-0.014702
		200.8239	3.838130	3.8703	-0.032170
		251.3638	4.995476	5.0500	-0.054524
		301.9554	6.136223	6.2104	-0.074177
39	Downstream of After-Cooler	99.0061	1.497807	1.5005	-0.002693
		149.7695	2.660598	2.6765	-0.015902
		200.8239	3.838130	3.8723	-0.034170
		251.3638	4.995476	5.0533	-0.057824
		301.9554	6.136223	6.2162	-0.079977

^a1968 IPTS.^bNBS smoothed data for type K thermocouples.¹²¹

gives a unique identification number, whereas Column 2 gives an identification related to the heat flux meter geometry (see Fig. IV-3) or the loop flow scheme (see Fig. IV-1). Column 3 tabulates the known (determined independently) temperature ($\pm 0.006^\circ\text{F}$) at which the calibrations were performed. Column 4 tabulates the thermal emf corresponding to the known temperature as determined from the smoothed data of NBS for Type K thermocouples,¹²¹ whereas the emf measured ($\pm 0.1 \mu\text{V}$) at that temperature is given in Column 5. The difference ($\text{EMF}_{\text{tab}} - \text{EMF}_{\text{meas}}$) is identified as the error and is tabulated in Column 6.

For purposes of analysis, the errors for the individual thermocouples were fitted to a cubic equation using a least-squares fitting and plotting routine¹²⁹ as:

$$E_{\text{err}} = A_1 + A_2 E_{\text{meas}} + A_3 E_{\text{meas}}^2 + A_4 E_{\text{meas}}^3 \quad (\text{A-9})$$

Coefficients of the fitted equations are tabulated in Table A-4, along with standard errors and F-ratio coefficients. Figure A-5 shows a plot of the fitted error equation for thermocouple No. 7, which is typical of the plots of all the fitted functions. As indicated in the figure, the fitted function is "well behaved" over the calibration range. The fitted error functions and the NBS functional representation¹²¹ for Type K thermocouples were used in a computer routine to obtain corrected temperatures from the measured emf values.

Upon completion of assembly and initial heat flux meter performance (exploratory) tests in the flow loop, a reliability check was made on the meter and bulk stream thermocouples. Since the loop had been drained for several days, it was assumed that equilibrium existed (except for the

TABLE A-4

RESULTS OF LEAST-SQUARES CUBIC FIT TO THERMOCOUPLE CALIBRATION DATA

Thermocouple	Coefficients ^a	Fitted Value	Standard Error	F Ratio
TC-7	A ₁	0.31888451D-03	0.24566296D-04	3.32612592D 04
	A ₂	-0.44198359D-02	0.28521870D-03	
	A ₃	0.35134326D-02	0.10032835D-02	
	A ₄	0.65840493D-02	0.10337306D-02	
TC-8	A ₁	0.36542548D-03	0.72156487D-04	3.49974102D 03
	A ₂	-0.50066705D-02	0.83740056D-03	
	A ₃	0.64801467D-02	0.29446443D-02	
	A ₄	0.28360161D-02	0.30334729D-02	
TC-9	A ₁	0.40072468D-03	0.12363130D-03	1.16783504D 03
	A ₂	-0.54604958D-02	0.14347723D-02	
	A ₃	0.83988491D-02	0.50454177D-02	
	A ₄	0.32829870D-03	0.51982328D-02	
TC-10	A ₁	0.38005063D-03	0.16193348D-03	6.86168542D 02
	A ₂	-0.51659302D-02	0.18814765D-02	
	A ₃	0.70873204D-02	0.66253804D-02	
	A ₄	-0.27024748D-02	0.58375784D-02	
TC-11	A ₁	0.26516578D-03	0.24972309D-04	2.84132498D 04
	A ₂	-0.39859718D-02	0.28982339D-03	
	A ₃	0.37593063D-02	0.10191977D-02	
	A ₄	0.45949266D-02	0.10500539D-02	
TC-19	A ₁	0.35360131D-03	0.80467711D-04	3.05231031D 03
	A ₂	-0.47647814D-02	0.93522738D-03	
	A ₃	0.45859994D-02	0.32940009D-02	
	A ₄	0.99136766D-03	0.33997765D-02	

TABLE A-4 (Continued)

Thermocouple	Coefficient ^a	Fitted Value	Standard Error	F Ratio
TC-21	A ₁	0.40129563D-03	0.13925294D-03	8.98208216D 02
	A ₂	-0.53901503D-02	0.16177157D-02	
	A ₃	0.80036580D-02	0.56958646D-02	
	A ₄	-0.36273872D-02	0.58779754D-02	
TC-22	A ₁	0.39327840D-03	0.13013268D-03	1.17314885D 03
	A ₂	-0.52459603D-02	0.15129314D-02	
	A ₃	0.63086933D-02	0.53307973D-02	
	A ₄	-0.19716181D-02	0.55047721D-02	
TC-23	A ₁	0.38010758D-03	0.14289179D-03	8.86249225D 02
	A ₂	-0.51626983D-02	0.16607156D-02	
	A ₃	0.70192506D-02	0.58499686D-02	
	A ₄	-0.35878542D-02	0.60400851D-02	
TC-24	A ₁	0.38697072D-03	0.13956218D-03	1.06742230D 03
	A ₂	-0.51958200D-02	0.16232887D-02	
	A ₃	0.58832508D-02	0.57223067D-02	
	A ₄	-0.23254753D-02	0.59120096D-02	
TC-25	A ₁	0.37311332D-03	0.14896259D-03	8.88260328D 02
	A ₂	-0.50123276D-02	0.17297023D-02	
	A ₃	0.55532920D-02	0.60856156D-02	
	A ₄	0.33574024D-02	0.62725358D-02	
TC-33	A ₁	0.37814021D-03	0.14332256D-03	9.62718818D 02
	A ₂	-0.50668759D-02	0.16647936D-02	
	A ₃	0.57028416D-02	0.58596656D-02	
	A ₄	0.19520930D-02	0.60427822D-02	

TABLE A-4 (Continued)

Thermocouple	Coefficient ^a	Fitted Value	Standard Error	F Ratio
TC-34	A ₁	0.30947810D-03	0.21394117D-03	3.75301796D 02
	A ₂	-0.45505379D-02	0.24825565D-02	
	A ₃	0.62319028D-02	0.87290742D-02	
	A ₄	0.15933892D-02	0.89931339D-02	
TC-35	A ₁	0.20585750D-03	0.14882845D-03	6.84781477D 02
	A ₂	-0.37008185D-02	0.17276134D-02	
	A ₃	0.56229068D-02	0.60780607D-02	
	A ₄	-0.23084913D-02	0.62680282D-02	
TC-36	A ₁	0.58684119D-04	0.80610320D-04	2.47207294D 03
	A ₂	-0.21789712D-02	0.93603297D-03	
	A ₃	0.78051219D-03	0.32940265D-02	
	A ₄	0.20769406D-02	0.33976703D-02	
TC-37	A ₁	0.30608224D-03	0.16328110D-03	9.58799672D 02
	A ₂	-0.43826008D-02	0.19017449D-02	
	A ₃	0.18950134D-02	0.67123473D-02	
	A ₄	0.10944321D-02	0.69426931D-02	
TC-38	A ₁	0.36988775D-03	0.14742595D-03	1.01369322D 03
	A ₂	-0.53385370D-02	0.17146450D-02	
	A ₃	0.73987322D-02	0.60433596D-02	
	A ₄	-0.29356793D-02	0.62419213D-02	
TC-39	A ₁	0.29631852D-03	0.14639847D-03	1.17541667D 03
	A ₂	-0.46667777D-02	0.17040242D-02	
	A ₃	0.47427048D-02	0.60100928D-02	
	A ₄	-0.41119370D-03	0.62111318D-02	

^aCoefficients for use in Eq. (A-9).

Fig. A-5. Calibration data for TC-7.

normal diurnal temperature fluctuations present in the laboratory area). Table A-5 shows the results of the check. [The number of significant figures indicated in the table are reported for consistency and are not intended to reflect the absolute accuracy.] The excellent agreement between the temperatures calculated from the fitted equations and the measured values gave assurance that the thermocouples were reliable. It is noted that the downstream bulk fluid thermocouple (TC-38 at HM-2) indicated about 0.02°F higher than that indicated by the upstream thermocouple (TC-37 at HM-1). Similarly, the extrapolated surface temperature of HM-2 is indicated to be 0.04°F higher than the corresponding value for HM-1. Although agreement between these values is excellent, the discrepancies are attributed to the fitted calibration error functions and the fitted temperature distributions within the heat flux meters. Periodic rechecks of the above test continued to give assurance of thermocouple reliability.

Another comparison was made between the bulk stream thermocouples (TC-37 and TC-38) by measuring the boiling temperature in an open beaker of demineralized water at a barometric pressure of 29.10 in. of Hg. The average of seven determinations gave the following results:

<u>Thermocouple</u>	<u>Temperature ($^{\circ}\text{F}$)</u>	<u>Standard Deviation ($^{\circ}\text{F}$)</u>
TC-37	210.66	0.029
TC-38	210.76	0.015

These are in excellent agreement with the interpolated value of 210.60°F obtained from the 1967 ASME Steam Tables at 29.10 in. of Hg.

TABLE A-5

ROOM TEMPERATURE CHECK OF HEAT FLUX METER THERMOCOUPLES

HM-1 ResultsLeast-Squares Equation:^a $T = 81.9681 - 0.0151097 X$ Std. Dev. = 0.0321, Dev. of Slope = 0.027, Dev. of $T_{x=0} = 0.033$

Gradient F-Ratio = 0.310, Stream Temperature = 81.86

TC No.	Position ^a (in.)	Meas Temp (°F)	Calc Temp (°F)	Meas-Calc (°F)
7	0.4579	81.9904	81.9612	0.0292
8	0.8336	81.9452	81.9555	-0.0103
9	1.2066	81.9056	81.9499	-0.0443
10	1.5781	81.9471	81.9443	0.0028
11	1.9576	81.9611	81.9385	0.0226

HM-2 ResultsLeast-Squares Equation:^a $T = 82.0143 - 0.0299858 X$ Std. Dev. = 0.015, Dev. of Slope = 0.013, Dev. of $T_{x=0} = 0.015$

Gradient F-Ratio = 5.66, Stream Temperature = 81.88

TC No.	Position ^a (in.)	Meas Temp (°F)	Calc Temp (°F)	Meas-Calc (°F)
21	0.4508	82.0022	82.0007	0.0015
22	0.8151	81.9773	81.9898	-0.0125
23	1.2003	81.9887	81.9783	0.0104
24	1.5784	81.9783	81.9669	0.0113
25	1.9532	81.9450	81.9557	-0.0107

^aT = degrees Fahrenheit, X = distance from boiling surface, in.

It was observed in the experimental portion of this research that TC-38 nearly always indicated a slightly higher (0.05 to 0.25°F) temperature than TC-37 for the same location, with TC-37 agreeing more closely with the temperature calculated from the measured pressure. For this reason, the temperature indicated by TC-37 was used as the reference for computing saturation pressure and heat transfer coefficients. An exception was made in the horizontal pool boiling experiments (the heat flux meters were isolated from each other; see Chapter IV for details) for which the temperature indicated by TC-38 was used as the reference for computing the h_{M-1} heat transfer coefficient.

A-4. System Heat Losses

Since the flow boiling tests with Freon under high void fraction conditions depended on a calculated heat balance between the state point station upstream of the flow meters and the heat flux meters, a series of tests were made to ascertain the enthalpy loss per pound of flowing fluid for use in the analysis. Although the system was well insulated and the experimentally determined losses were small (probably negligible), they were taken into account in the heat balance.

The system was operated at saturation conditions in the reservoir to provide a stable temperature distribution (none of the line heaters were on for these tests) throughout the loop, while the flow rate was varied over the range of interest. The two thermocouples at the heat meters were positioned at the same stream point to provide duplicate temperature measurements from which the temperature loss experienced by

the fluid between the state point station and the meter could be determined. This loss was transformed into an enthalpy loss assuming saturated liquid conditions existed. [In reality the fluids were slightly subcooled at the end points, but this introduced negligible error.] Five flow velocities (0.5, 1, 2, 3, and 4 ft/sec) were employed for this purpose. The reduced data could be represented reasonably well with a hyperbolic function as shown in Fig. A-6, wherein the enthalpy loss (Btu/lb) is plotted as a function of the reciprocal of the mass flow (lb/hr)⁻¹. The ORFAL data analysis program¹²³ was used to determine a least-squares fit to the data as:

$$\Delta H = A_1/W + A_2 \quad . \quad (A-10)$$

The coefficient $A_1 = 214.851$ with a standard error of ± 6.413 , and $A_2 = 0.230 \times 10^{-3}$ with a standard error of $\pm 0.185 \times 10^{-2}$. The fitted equation has a standard deviation of ± 0.0037 Btu/lb.

Since the Freon temperature was varied over a narrow range (118 to 138°F) in the high void fraction tests, it was judged acceptable to use Eq. (A-10) for the entire range without including a temperature effect.

A-5. Heat Flux Meter Heat Loss Calibration

Flow Test Calibration

A number of tests were made to ascertain the equilibrium heat loss of the individual heat flux meters as installed in the flow loop. Three temperature levels, spanning the range of fluid temperatures expected in the experimental program, were used with water flowing at 2 ft/sec.

The condition of zero temperature gradient (within the meter) at the boiling surface indicates no transfer of heat across the surface.

Fig. A-6. System heat loss as a function of mass flow rate.

The electrical input power required to achieve this condition was determined indirectly at the three temperature levels in the following manner. The temperature of the fluid flowing past the meters was adjusted to the desired value and a series of runs made at various low levels of meter heater power. Temperature data were recorded for each meter as a function of measured electrical power. The temperature data were fitted to a linear least-squares equation to provide the gradient as a function of heater power at each temperature level. The gradients thus determined for a number of power levels were then fitted (using a linear least-squares routine) to an equation relating the gradient to the power level at that particular temperature level. The root of the resulting equation is the power level required to produce a zero temperature gradient at the boiling surface for that temperature level. The zero-gradient-power levels were then fitted to a linear least-squares function of the temperature for each of the tests. Under these conditions the indicated temperature, as well as the average of the indicated values, within the heat flux meter is the same as the surface temperature, at least within a standard deviation of the fitted temperature distribution.

Figure A-7 shows the results of the calculated gradient versus the measured input electrical power for each of the heat flux meters for fluid temperatures in the range of 213 to 216°F. Similar results were obtained for fluid temperature values of 125 and 180°F. Figure A-8 shows the roots of the equations, i.e., the input electrical power required to produce zero temperature gradient within the meters, as a function of surface temperature (essentially the same as the average of the indicated values).

Fig. A-7. Meter temperature gradient vs input power.

The equilibrium heat loss functions were determined to be linear over the range of interest and are given for HM-1 as:

$$P_{HL-1} = -0.614404 + 0.00781815 \bar{T} , \quad (A-11)$$

with a standard deviation of 0.001145 watt; and for HM-2 as:

$$P_{HL-2} = -0.815729 + 0.0122395 \bar{T} , \quad (A-12)$$

with a standard deviation of 0.04276 watt.

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Fig. A-8. Input power required to produce zero temperature gradient.

These equations were used to estimate the equilibrium heat loss for the flow tests, using the average of the measured temperatures within the heat flux meters. This is known to be a conservative estimate of the minimum equilibrium heat loss for two reasons:

1. The average value of the indicated temperature is less than the average temperature of the entire copper body of the meter. Reference

to Fig. IV-3 shows that considerable surface area will be at a temperature higher than the highest value included in the averaging process. Since no estimate was available for the true average temperature, the average of the measured values was used to calculate the equilibrium heat loss associated with each data point.

2. The manner by which the electrical leads were joined to the meter heater (as described in Chapter IV) necessitated about 1 in. of heater on each end not being brazed to the copper body. (About 10 in. of heater was brazed to the body.) Copper sleeves were brazed over the heater extensions to conduct heat to the body, but no estimate of this heat flow could be made. At the low power levels used to determine the heat loss functions, this loss was included in the overall loss; however, at the higher power levels associated with the data runs, this contribution to the total would be underestimated by the fitted heat loss equations. Thus, an unestimable fraction of the electrical power input never reaches the useful portion of the heat flux meter.

The heat loss calculated from the above equations will not include the heat transferred to the test fluid through the stainless steel flange surrounding the boiling surface of the meter. Based on design calculations discussed in Appendix B, this is shown to be a small fraction of the total heat input and is included as unaccounted heat losses. Therefore, the calculated heat balance, consisting of the heat flow through the boiling surface (determined from the Fourier heat conduction equation evaluated at the boiling surface) and the loss calculated from

Eq. (A-11) or Eq. (A-12), does not include the small heat losses discussed above; for these reasons, the calculated heat balance should be less than unity.

Checks were made prior to the Freon-113 flow boiling tests to determine if the meter heat loss calibrations made with water flowing at 2 ft/sec were applicable (as expected) under similar conditions to the refrigerant. For an average meter temperature of 118.5°F, the experimentally determined heat loss for meter HM-2 was 0.577 W, whereas Eq. (A-12) predicted 0.635 W. Similarly for HM-1 at an average temperature of 121.9°F, the heat loss was determined to be 0.375 W, which is to be compared to a value of 0.312 W computed from Eq. (A-11). Since these checks agreed well with the predictive expressions, the equations were also used to calculate the meter heat losses in the Freon-113 tests.

It may be noted from these results that meter HM-1 was somewhat better insulated against heat losses. This was also apparent in the pool boiling experiments in which the entire assembly was submerged in the boiling fluid (see Chapter 4 for a description of the physical arrangement).

APPENDIX B

COMPUTER ANALYSIS OF HEAT METER DESIGN

It was recognized at the outset of this research that much would depend on successful elimination of bubble nucleation at the periphery of the boiling surface and/or on the flange surface surrounding the test surface. This bothersome problem plagued a number of investigators^{74,117,118} both at this laboratory and elsewhere. Aside from the annoyance, definition of the boiling area was uncertain and varied with the heat flux level. As a result heat loss corrections, generally based on the fin equations, were necessary.

Based on physical considerations of thermal conductivity, thermal expansion, and material compatibility (for brazing), it was concluded that a thin stainless steel flange brazed to the end of a copper cylinder would most likely be the best meter design concept. Obviously the thinner the stainless steel flange, the better the performance. However, increasingly more difficult fabrication problems accompany the increased performance; and a compromise must be made between these two conflicting aspects.

Conceptual designs based on a flange thickness in the range of 0.010 to 0.015 in. was judged feasible by shop personnel. A number of design calculations were performed, using the HEATING-3 computer program,¹²⁶ to determine the expected performance characteristics.

HEATING-3 (Ref. 126), a modified version of the HEATING program,¹²⁷ is designed to solve a wide variety of steady-state and/or transient heat conduction problems in one-, two-, or three-dimensional Cartesian or

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cylindrical coordinates. Input values of thermal conductivity, density, and specific heat may be dependent on temperature at each lattice point. Specified heat generation rates may be time dependent. The boundary conditions may be fixed temperatures or any combination of constant heat flux, forced convection, natural convection, and radiation and can be from-surface-to-boundary, from-surface-to-surface, or from-surface-to-surface plus conduction. The mesh spacing can vary along each axis, with as many as 1750 lattice points, 100 regions, 50 materials, and 50 boundary conditions being accommodated.

For purposes of analysis, the design shown earlier (see Fig. IV-3) was modeled conceptually as depicted in Fig. B-1. Fourteen regions (for a total of 1073 node points) were necessary to avoid numerical instabilities in the calculation. The thin plating of nickel on the surface exposed to the fluid was neglected. The calculations were performed for R-Z geometry under steady-state conditions.

In order to use the HEATING program, boundary conditions, a volumetric heat generation function, and an initial temperature distribution must be specified. A relaxation technique is used to calculate a new temperature distribution; the calculation continues until the specified convergence criterion is satisfied (10^{-7} was generally used) or the specified computing time limit is reached. After the final temperature iteration, the heat fluxes across all the external boundaries and the temperature at each node is printed. Thus, since the boiling heat flux is a derived result, it is necessary to repeat the calculation, making an adjustment in the input heat generation function, a number of times to match the boiling heat flux to the specified boiling heat transfer

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Fig. B-1. Calculational model of heat flux meter.

coefficient. This, of course, is costly in computing time and should be avoided. This was accomplished by making survey calculations on a simplified model.

With the exception of natural-convection heat transfer coefficients, temperature independent input parameters were used. In the case where the meter was considered submerged in 212°F water (to simulate the pool boiling test setup), natural-convection coefficients were specified as:

$$h_{\text{nat conv}} = 0.46 \Delta T^{1/3}, \quad (\text{B-1})$$

for all the external boundaries except the boiling surface. A constant value for the pool boiling coefficient was specified to match the desired boiling heat flux. The data of Marcus¹⁰⁹ were used for this purpose. In those cases where the model simulated the flow boiling test setup, the surfaces exposed to air at 80°F were specified to have a natural-convection coefficient given by:

$$h_{\text{nat conv}} = 0.001319 \Delta T^{1/3} \quad (\text{B-2})$$

Calculations were performed for both 10-mil and 15-mil flange thicknesses. As would be expected, the temperature and heat flux distribution on the face of the 15-mil flange was not as flat as those for the 10-mil flange.

The surface temperature and heat flux distributions calculated for the meter (0.010-in.-thick flange) submerged in water at 212°F were shown earlier in Fig. IV-5. A pool boiling coefficient of 2304 Btu/hr·ft²·°F (at 50,000 Btu/hr·ft² heat flux) was used for the boiling surface, and Eq. (B-1) was specified for the remainder of the external surfaces. Figure B-2 shows the corresponding calculated axial (center

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Fig. B-2. Calculated axial temperature distribution.

line) temperature for the model. Excellent linearity in the reduced diameter portion of the meter is readily apparent in the figure. The corresponding radial temperature profile at the middle thermocouple is depicted in Fig. B-3. The calculated distribution indicates negligible radial heat loss.

For this case, the analysis shows that 89.9% of the electrical power input is transferred to the fluid by boiling on the test surface, 2.8% is transferred by natural convection from the stainless steel flange, and 7.3% by natural convection from the remaining exposed surfaces. It was concluded from these results that the stainless steel flange thickness should be specified as 0.010 in.

Fig. B-3. Calculated radial temperature distribution.

APPENDIX C

THERMODYNAMIC AND PHYSICAL PROPERTIES OF TEST FLUIDS

C-1. Water

The thermodynamic and transport properties of water were calculated with a computer routine written (unpublished) at Oak Ridge National Laboratory, using the equations and constants available in the 1967 ASME Steam Tables.

C-2. Freon-113

The thermodynamic and physical properties of Freon-113 are not known to the same accuracy and are not as readily available as for water. For the purposes of this research, published¹³⁵ and unpublished¹³⁶ data were fitted over the range of interest to facilitate data reduction and analysis by computer routines. The ORDEAL¹²⁹ least-squares fitting and plotting routines were used for this purpose. It was determined for the limited temperature range that first, second, or third degree polynomials (depending on the property) gave lower standard errors for the coefficients than did fourth, fifth, or sixth degree fits.

The fitted equations, used for data reduction, are given below in skeletal format; Table C-1 lists the coefficients and pertinent statistical parameters.

1. Saturation temperature as a function of absolute pressure.

$$T_{\text{sat}} = f(P); 100 \leq T_{\text{sat}} (^{\circ}\text{F}) \leq 160, 10.48 \leq P (\text{psia}) \leq 30.44.$$

$$T_{\text{sat}} = A_1 P^3 + A_2 P^2 + A_3 P + A_4. \quad (\text{C-1})$$

Data source: Bulletin T-113A (Ref. 135).

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TABLE C-1

RESULTS OF LEAST-SQUARES FITTED EQUATIONS FOR FREON-113 PROPERTIES

Equation	Coefficient	Fitted Value	Standard Error	F Ratio
C-1	A ₁	0.18971518D-02	0.66927096D-04	6.63460074D 05
	A ₂	-0.17542512D 00	0.40739514D-02	
	A ₃	0.76127221E 01	0.79013288D-01	
	A ₄	0.37424943D 02	0.48532009D 00	
C-2	A ₁	0.68132186D-05	0.22817187E-06	8.48753553D 06
	A ₂	-0.41520714D-03	0.89058609D-04	
	A ₃	0.88963580D-01	0.11474814D-01	
	A ₄	-0.10740069D 01	0.48789346D 00	
C-3	A ₁	0.12810952D-03	0.23385419D-05	1.25233786D 07
	A ₂	0.19566733D 00	0.60916762D-03	
	A ₃	0.81494590D 01	0.39081710D-01	
C-4	A ₁	-0.14631156D-03	0.23824005D-05	1.53830295D 00
	A ₂	-0.43684801E-01	0.62059239D-03	
	A ₃	0.70286251D 02	0.39814685D-01	
C-7	A ₁	-0.24791887D-05	0.66920961D-07	8.55586863D 05
	A ₂	0.12765348D-02	0.26120164D-04	
	A ₃	-0.23531746D 00	0.33654689D-02	
	A ₄	0.16217747D 02	0.14309514D 00	
C-8	A ₁	0.21326364D 04	0.71259749D 00	4.47831942D 06
	A ₂	-0.34891142D 01	0.11847024D-02	
C-10	A ₁	0.13888889D-08	0.28350566D-09	3.69246429D 03
	A ₂	-0.86607143D-06	0.15336286D-06	
	A ₃	0.21093254D-03	0.26964204D-04	
	A ₄	0.10450000D-01	0.15373565D-02	
C-11	A ₁	-0.65714286D-04	0.41239304D-06	1.26960000D 04
	A ₂	0.48461905D-01	0.47499852D-04	
C-12	A ₁	0.13857143D-03	0.38685876D-05	6.41522727D 02
	A ₂	0.21792381D 00	0.44558803D-03	
C-13	A ₁	-0.60019841D-01	0.34436025D-03	1.51891599D 04
	A ₂	0.21901349D 02	0.47294084D-01	

2. Saturation pressure (absolute) as a function of temperature.

$$P_{\text{sat}} = f(T); 10.48 \leq P_{\text{sat}} \text{ (psia)} \leq 30.44, 100 \leq T \text{ (°F)} \leq 160.$$

$$P_{\text{sat}} = A_1 T^3 + A_2 T^2 + A_3 T + A_4. \quad (\text{C-2})$$

Data source: Bulletin T-113A (Ref. 135).

3. Saturated liquid enthalpy as a function of temperature.

$$h_f = f(T_{\text{sat}}); 28.99 \leq h_f \text{ (Btu/lb}_m\text{)} \leq 42.74, 100 \leq T_{\text{sat}} \text{ (°F)} \leq 160.$$

$$h_f = A_1 T_{\text{sat}}^3 + A_2 T_{\text{sat}}^2 + A_3 T_{\text{sat}} + A_4. \quad (\text{C-3})$$

Data source: Bulletin T-113A (Ref. 135).

4. Latent heat of vaporization as a function temperature.

$$h_{fg} = f(T_{\text{sat}}); 64.46 \leq h_{fg} \text{ (Btu/lb}_m\text{)} \leq 59.55, 100 \leq T_{\text{sat}} \text{ (°F)} \leq 160.$$

$$h_{fg} = A_1 T_{\text{sat}}^2 + A_2 T_{\text{sat}} + A_3. \quad (\text{C-4})$$

Data source: Bulletin T-113A (Ref. 135).

5. Saturated liquid density as a function of temperature.

$$\rho_l = f(T_{\text{sat}}); -30 \leq T_{\text{sat}} \text{ (°F)} \leq 220, 105.64 \leq \rho_l \text{ (lb}_m\text{/ft}^3\text{)} \leq 84.80.$$

$$\rho_l = 103.555 - 0.07126 T_{\text{sat}} - 0.0000636 T_{\text{sat}}^2. \quad (\text{C-5})$$

Equation given in Bulletin T-113A (Ref. 135).

6. Saturated liquid specific volume as a function of temperature.

$$v_l = f(T_{\text{sat}}); 0.00947 \leq v_l \text{ (ft}^3\text{/lb}_m\text{)} \leq 0.01179, -30 \leq T_{\text{sat}} \text{ (°F)} \leq 220.$$

$$v_l = 1/\rho_l, \quad (\text{C-6})$$

where ρ_l is given by Eq. (C-5).

7. Saturated vapor specific volume as a function of temperature.

$$v_g = f(T_{\text{sat}}); 2.976 \leq v_g \text{ (ft}^3\text{/lb}_m\text{)} \leq 1.094, 100 \leq T_{\text{sat}} \text{ (°F)} \leq 160.$$

$$v_g = A_1 T_{\text{sat}}^3 + A_2 T_{\text{sat}}^2 + A_3 T_{\text{sat}} + A_4. \quad (\text{C-7})$$

Data source: Bulletin T-113A (Ref. 135).

8. Saturated liquid viscosity as a function of temperature.

$$\mu_l = f(T_{\text{sat}}); 1.849 \leq \mu_l \text{ (lb}_m\text{/ft}\cdot\text{hr)} \leq 0.643, 60 \leq T \text{ (}^\circ\text{F)} \leq 240.$$

$$\ln \mu_l = A_1 Z + A_2, \quad (\text{C-8})$$

$$\text{where } Z \equiv (T_{\text{sat}} + 459.67)^{-1}. \quad (\text{C-9})$$

Data source: DuPont Company.¹³⁶

9. Saturated vapor viscosity as a function of temperature.

$$\mu_g = f(T_{\text{sat}}); 0.0257 \leq \mu_g \text{ (lb}_m\text{/ft}\cdot\text{hr)} \leq 0.0304, 120 \leq T_{\text{sat}} \text{ (}^\circ\text{F)} \leq 240.$$

$$\mu_g = A_1 T_{\text{sat}}^3 + A_2 T_{\text{sat}}^2 + A_3 T_{\text{sat}} + A_4. \quad (\text{C-10})$$

Data source: DuPont Company.¹³⁶

10. Saturated liquid thermal conductivity as a function of temperature.

$$k_l = f(T_{\text{sat}}); 0.0445 \leq k_l \text{ (Btu/hr}\cdot\text{ft}\cdot\text{}^\circ\text{F)} \leq 0.0379, 100 \leq T_{\text{sat}} \text{ (}^\circ\text{F)} \leq 160.$$

$$k_l = A_1 T_{\text{sat}} + A_2. \quad (\text{C-11})$$

Data source: DuPont Company.¹³⁶

11. Saturated liquid specific heat as a function of temperature.

$$C_p = f(T_{\text{sat}}); 0.226 \leq C_p \text{ (Btu/lb}_m\text{ }^\circ\text{F)} \leq 0.240, 60 \leq T_{\text{sat}} \text{ (}^\circ\text{F)} \leq 160.$$

$$C_p = A_1 T_{\text{sat}} + A_2. \quad (\text{C-12})$$

Data source: DuPont Company.¹³⁶

12. Saturated liquid surface tension as a function of temperature.

$$\sigma = f(T_{\text{sat}}); 17.87 \leq \sigma \text{ (dynes/cm)} \leq 10.31, 68 \leq T_{\text{sat}} \text{ (}^\circ\text{F)} \leq 194.$$

$$\sigma = A_1 T_{\text{sat}} + A_2. \quad (\text{C-13})$$

Data source: Sinitsyn et al.¹³⁷

APPENDIX D

DATA SHEETS, REDUCED DATA TABULATIONS, AND
CROSS REFERENCE LISTING OF EXPERIMENTS

HEAT METER FLOW TEST

ONEL DMC 74-1233

Sheet 1

Approximate Stream Velocity _____ ft/sec		Run No. _____	
Fluid _____ Direction of Flow _____		Recorded by _____	
IN. NO.	Date: _____ Time: _____		
	Barometer (Direct)		
	Average Gravity Correction		
	(°F) Temperature Correction		
IN. FLUID	Corrected Barometer		
	Manometer (uncorrected) 1		
	2		
	Mercury-Water-Film 3		
	4		
PGI	Manometer Pressure 5		
	Static Gas Pressure		
	(height correction)		
WERTZ	Corrected Manometer Pressure		
	Frequency Flow Meter 1		
	2		
OPM	3		
	Estimated OPM Meter 1		
	2		
In.	Total Estimated Stream Flow		
	Estimated OPM Meter 3		
	Drum Level		
ON-OFF	Trim Cooler	On	Off
ON-OFF	After Cooler	On	Off
	Reactor No. 1, $\frac{g}{min}$		
	No. 2, $\frac{g}{min}$		
	No. 3, $\frac{g}{min}$		
	No. 4, $\frac{g}{min}$		
	HE-1 Voltage		
	HE-1 Shunt RWP X (0.09777)		
	Approximate HE-1 Power		
	HE-2 Voltage		
	HE-2 Shunt RWP X (0.09969)		
	Approximate HE-2 Power		
	Main EM Variac, %		

Req. Supp. Press. _____ in. Fluid
 at _____ °F
 Approximate Heat Flux
 HE-1 _____
 HE-2 _____

General Information and Test Conditions

Fig. D-1. Data sheet 1 for flow tests.

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POOL BOILING HEAT METER TESTS			Sheet 1
Type _____	Recorded by _____		
Position _____	Run No. _____		
Fluid _____			
Time _____			Date: _____
Barometer (direct) (in. Hg)			Approximate Heat Flux
Average gravity correction	-0.025		
(°F) Temperature correction	-0.		HM-1 _____
Corrected Barometer (in. Hg)			HM-2 _____
Heater No. 1 $\frac{W}{sqft}$			
Heater No. 2 $\frac{W}{sqft}$			
HM-1 Voltage (V)			
HM-1 Shunt EMF (V)			
HM-2 Voltage (V)			
HM-2 Shunt EMF (V)			
Pool depth (in.)			
Hot Plate Temperature (°F)			
General Information			

Fig. D-3. Data sheet 1 for pool boiling tests.

CHANNEL-ANAL
T/C SWP

Sheet 2

DATE: _____

Recorded by _____

Run No. _____

SWITCH POINT NO.		TIME	
	7		
	8		
	9		
	10		
	11		
	Δ 12		
	Δ 13		
	Δ 14		
	Δ 15		
	Δ 16		
	Δ 17		
	Δ 18		
	19		
SWITCHING POINT	20	No	TC
	21		
	22		
	23		
	24		
	25		
	Δ 26		
	Δ 27		
	Δ 28		
	Δ 29		
	Δ 30		
	Δ 31		
	Δ 32		
	33		
	34		
BULK	35		
BULK	36		
SW-2 BULK	37		
SW-1 BULK	38		
	39		
	40		

Number of Nucleation Sites
 1 _____ 2 _____
 O Cont. _____ O Cont. _____
 Δ Int. _____ at _____ Δ Int. _____ at _____

 Lower Upper

Pictures Taken _____

Operating conditions last changed
 at _____
 SW-1 SW-2

General Inform. and Test Conditions

Fig. D-4. Data sheet 2 for pool boiling tests.

TABLE D-1

REDUCED DATA FOR POOL BOILING OF FREON-113 ON A HORIZONTAL COPPER SURFACE
USING HEAT METER HM-1

Run Ident.	Date	Hour	q'' (HM-1)	Meas. Press. Psia	Temperature			h Coeff. Btu hr-ft ² -°F
			$\frac{\text{Btu}}{\text{hr-ft}^2}$		Wall °F	Saturation °F	Diff °F	
HPBPC1-001	12-04-73	1245	314	14.51	120.69	116.51	4.18	75
HPBPC1-002	12-05-73	1045	15271	14.49	133.45	116.10	17.35	880
HPBPC1-003	12-05-73	1605	4842	14.53	133.42	114.44	18.99	285
HPBPC1-004	12-05-73	1710	4254	14.54	133.40	116.50	16.90	253
HPBPC1-005	12-05-73	1810	2985	14.56	133.47	116.94	16.54	181
HPBPC1-006	12-05-73	1915	2092	14.57	133.20	117.08	16.12	130
HPBPC1-007	12-05-73	2025	1234	14.59	129.30	117.07	12.23	101
HPBPC1-008	12-05-73	2130	826	14.60	126.54	117.26	9.29	88
HPBPC1-009	12-06-73	1015	285	14.68	120.80	117.31	3.49	82
HPBPC1-010	12-06-73	1115	12388	14.68	134.73	117.32	17.41	711
HPBPC1-011	12-06-73	1220	9859	14.66	134.56	117.21	17.36	560
HPBPC1-012	12-06-73	1320	7340	14.66	134.55	117.23	17.32	424
HPBPC1-013	12-06-73	1415	4946	14.66	134.36	117.26	17.10	289
HPBPC1-014	12-06-73	1500	45408	14.66	137.90	116.99	20.91	2171
HPBPC1-015	12-06-73	1535	32162	14.67	137.43	117.14	20.29	1585
HPBPC1-016	12-06-73	1620	25657	14.67	137.08	117.18	19.91	1289
HPBPC1-017	12-06-73	1700	22947	14.67	136.92	117.29	19.64	1149
HPBPC1-018	12-06-73	1740	20207	14.67	136.78	117.30	19.48	1037
HPBPC1-019	12-06-73	1830	17396	14.68	136.70	117.33	19.37	890
HPBPC1-020	12-06-73	1910	13626	14.69	136.45	117.35	19.11	713
HPBPC1-021	12-06-73	2000	8336	14.69	136.16	117.47	18.69	446
HPBPC1-022	12-07-73	0830	277	14.75	121.11	117.70	3.41	81
HPBPC1-023	12-07-73	1015	45132	14.76	139.07	117.41	21.66	2004
HPBPC1-024	12-07-73	1050	32222	14.76	138.60	117.49	21.11	1526

TABLE D-1 (Continued)

Run Ident.	Date	Hour	q'' (HR-1) Rth HR-Ptz	Meas. Press. Pala	TEMPERATURE		h Coeff. Rth HR-Ptz-SP
					Wall Cp	Saturation Diff Op	
NPBPC1-025	12-07-73	1130	22981	14.75	138.08	117.51	20.57
NPBPC1-026	12-07-73	1210	14888	14.74	137.45	117.50	19.96
NPBPC1-027	12-07-73	1245	9704	14.73	137.03	117.50	19.54
NPBPC1-028	12-07-73	1330	7266	14.72	136.77	117.49	19.28
NPBPC1-029	12-07-73	1410	6795	14.72	136.50	117.53	18.97
NPBPC1-030	12-07-73	1500	2965	14.72	135.89	117.50	18.40
NPBPC1-031	12-07-73	1530	2009	14.71	134.25	117.61	16.64
NPBPC1-032	12-07-73	1620	1234	14.71	129.40	117.61	11.79
							1112
							746
							497
							377
							251
							161
							121
							105

TABLE D-2

REDUCED DATA FOR POOL BOILING OF PWRON-113 ON A HORIZONTAL COPPER SURFACE
USING HEAT METER HM-2

Run Ident.	Date	Hour	q'' (HM-2)		Press. Pria	TEMPERATURE		h COEFF.	
			Rth W-Pts	W-Pts		Wall Saturation	DIFF	Rth W-Pts	W-Pts
HPBPC2-001	12-04-73	1245	180	18.51	120.71	116.58	4.13	84	
HPBPC2-002	12-04-73	1345	429	18.51	123.28	116.19	7.09	60	
HPBPC2-003	12-04-73	1450	812	18.51	128.00	116.47	11.53	70	
HPBPC2-004	12-05-73	1045	15985	18.49	134.07	116.13	17.94	891	
HPBPC2-005	12-05-73	1605	5447	18.53	133.78	116.50	17.27	315	
HPBPC2-006	12-05-73	1710	4829	18.54	133.71	116.60	17.11	259	
HPBPC2-007	12-05-73	1810	3404	18.56	133.82	116.93	16.90	201	
HPBPC2-008	12-05-73	1915	2317	18.57	133.73	117.21	16.52	140	
HPBPC2-009	12-05-73	2025	1379	18.59	132.19	117.34	14.85	93	
HPBPC2-010	12-05-73	2130	806	18.60	129.05	117.47	11.62	69	
HPBPC2-011	12-06-73	1015	148	18.68	121.22	117.51	3.71	40	
HPBPC2-012	12-06-73	1115	13249	18.68	135.16	117.44	17.72	748	
HPBPC2-013	12-06-73	1220	10638	18.66	134.97	117.31	17.66	602	
HPBPC2-014	12-06-73	1320	8009	18.66	134.90	117.34	17.57	456	
HPBPC2-015	12-06-73	1415	5306	18.66	134.73	117.33	17.41	305	
HPBPC2-016	12-06-73	1500	39291	18.66	138.05	117.01	21.04	1868	
HPBPC2-017	12-06-73	1535	30489	18.67	137.66	117.23	20.43	1492	
HPBPC2-018	12-06-73	1620	25411	18.67	137.39	117.25	20.14	1262	
HPBPC2-019	12-06-73	1700	22941	18.67	137.28	117.28	20.01	1147	
HPBPC2-020	12-06-73	1740	20579	18.67	137.15	117.34	19.81	1039	
HPBPC2-021	12-06-73	1830	17876	18.68	137.13	117.41	19.73	906	
HPBPC2-022	12-06-73	1910	13980	18.69	136.75	117.49	19.26	726	
HPBPC2-023	12-06-73	2000	9930	18.69	136.49	117.59	18.90	525	
HPBPC2-024	12-07-73	0830	179	18.75	121.43	117.75	3.69	49	

TABLE n-2 (Continued)

Run Ident.	Date	Hour	q'' (HH-2)	Meas. Press. Psia	Temperature			h Coeff.
			$\frac{Btu}{Hr-Ft^2}$		Wall °F	Saturation °F	Diff °F	
								$\frac{Btu}{Hr-Ft^2-°F}$
HPBFC2-025	12-07-73	1015	39130	14.76	138.99	117.55	21.44	1025
HPBFC2-026	12-07-73	1050	31083	14.76	138.72	117.49	21.24	1060
HPBFC2-027	12-07-73	1130	22877	14.75	138.24	117.59	20.65	1108
HPBFC2-028	12-07-73	1210	15574	14.74	137.73	117.63	20.09	775
HPBFC2-029	12-07-73	1245	10749	14.73	137.15	117.59	19.56	550
HPBFC2-030	12-07-73	1330	7995	14.72	136.97	117.61	19.36	413
HPBFC2-031	12-07-73	1410	5383	14.72	136.60	117.64	18.96	284
HPBFC2-032	12-07-73	1500	3309	14.72	136.16	117.58	18.59	178
HPBFC2-033	12-07-73	1530	2162	14.71	135.93	117.56	18.37	118
HPBFC2-034	12-07-73	1620	1265	14.71	132.80	117.75	15.05	84

TABLE D-3

REDUCED DATA FOR POOL BOILING OF FREON-113 ON A VERTICAL COPPER SURFACE
USING HEAT METER NM-1

Run Ident.	Date	Hour	q'' (NM-1)	Meas. Press. Psia	Temperature			h Coeff.
			$\frac{\text{Btu}}{\text{Hr-Ft}^2}$		Wall °F	Saturation °F	Diff °F	$\frac{\text{Btu}}{\text{Hr-Ft}^2-\text{°F}}$
For Varying NM-1 Flux								
VPBFC1-001	12-12-73	0820	342	14.80	121.05	117.15	3.90	88
VPBFC1-002	12-13-73	0830	503	14.47	123.99	116.46	7.53	67
VPBFC1-003	12-13-73	1045	45293	14.50	138.21	116.02	22.19	2042
VPBFC1-004	12-13-73	1120	32156	14.47	137.41	115.97	21.44	1500
VPBFC1-005	12-13-73	1210	25630	14.47	136.95	116.00	20.94	1224
VPBFC1-006	12-13-73	1240	20227	14.46	136.49	115.92	20.57	983
VPBFC1-007	12-13-73	1320	14912	14.46	135.81	115.95	19.85	751
VPBFC1-008	12-13-73	1405	9620	14.46	134.81	116.04	18.77	513
VPBFC1-009	12-13-73	1450	7231	14.46	134.25	116.14	18.11	399
VPBFC1-010	12-13-73	1525	4886	14.45	133.32	116.25	17.07	286
VPBFC1-011	12-13-73	1610	3009	14.46	131.89	116.27	15.62	193
VPBFC1-012	12-13-73	1705	2096	14.46	131.05	116.31	14.74	142
VPBFC1-013	12-13-73	1755	1315	14.48	128.81	116.51	12.31	107
VPBFC1-014	12-14-73	0820	924	14.66	128.68	117.21	11.47	81
VPBFC1-015	12-15-73	1200	424	14.65	122.19	117.15	5.03	84
For Constant NM-1 Flux								
VPBFC1-016	12-14-73	0930	32103	14.66	138.19	116.79	21.40	1500
VPBFC1-017	12-14-73	1010	32059	14.65	138.32	116.86	21.46	1494
VPBFC1-018	12-14-73	1045	32014	14.66	138.41	116.86	21.55	1486
VPBFC1-019	12-14-73	1130	32023	14.66	138.51	116.86	21.65	1479

TABLE D-3 (Continued)

Run Ident.	Date	Hour	q'' (HM-1)	Meas. Press. rsia	Temperature			h Coeff. Btu Hr-Ft ² -°F
			$\frac{\text{Btu}}{\text{Hr-Ft}^2}$		Wall °F	Saturation °F	Diff °F	
VPBFC1-020	12-14-73	1210	32225	14.66	138.55	116.92	21.63	1490
VPBFC1-021	12-14-73	1250	14806	14.66	137.59	116.88	20.71	715
VPBFC1-022	12-14-73	1330	14840	14.65	137.56	116.80	20.75	715
VPBFC1-023	12-14-73	1400	14881	14.65	137.56	116.90	20.66	720
VPBFC1-024	12-14-73	1440	14821	14.66	137.59	116.92	20.67	717
VPBFC1-025	12-14-73	1515	14828	14.66	137.69	117.00	20.68	717
VPBFC1-026	12-14-73	1615	14884	14.66	137.73	117.00	20.73	718

TABLE D-4

REDUCED DATA FOR POOL BOILING OF FREON-113 ON A VERTICAL COPPER SURFACE
USING HEAT METER HM-2

Run Ident.	Date	Hour	q'' (HM-2)	meas. Press. Psia	Temperature			h Coeff. Btu Hr-Ft ² -°F	q'' (HM-1)
			$\frac{\text{Btu}}{\text{Hr-Ft}^2}$		Wall °F	Saturation °F	Diff °F		$\frac{\text{Btu}}{\text{Hr-Ft}^2}$
For Varying HM-2 Flux									
VPBFC2-001	12-12-73	0940	39171	14.65	139.49	117.22	22.27	1759	(0)
VPBFC2-002	12-12-73	1030	30119	14.64	138.97	117.14	21.82	1380	(0)
VPBFC2-003	12-12-73	1120	20703	14.64	138.17	117.10	21.08	982	(0)
VPBFC2-004	12-12-73	1200	15602	14.63	137.70	117.02	20.68	755	(0)
VPBFC2-005	12-12-73	1315	10578	14.58	136.66	117.21	19.44	544	(0)
VPBFC2-006	12-12-73	1430	7916	14.54	136.11	117.19	18.92	418	(0)
VPBFC2-007	12-12-73	1455	39377	14.54	139.12	117.00	22.12	1780	(0)
VPBFC2-008	12-12-73	1620	30246	14.52	139.03	117.03	21.99	1375	(0)
VPBFC2-009	12-12-73	1700	20594	14.52	138.62	117.08	21.55	956	(0)
VPBFC2-010	12-12-73	1740	15627	14.51	138.10	117.02	21.09	741	(0)
VPBFC2-011	12-12-73	1820	10519	14.51	137.19	117.06	20.13	523	(0)
VPBFC2-012	12-12-73	1855	7932	14.51	136.59	117.09	19.50	407	(0)
VPBFC2-013	12-12-73	1940	5349	14.50	135.69	117.08	18.61	287	(0)
VPBFC2-014	12-12-73	2020	3183	14.50	134.85	117.07	17.78	179	(0)
VPBFC2-015	12-12-73	2055	2359	14.49	134.31	116.99	17.31	136	(0)
VPBFC2-016	12-12-73	2135	1238	14.49	132.59	117.02	15.57	80	(0)
VPBFC2-017	12-13-73	0830	336	14.31	122.62	116.30	6.32	53	(503)
VPBFC2-018	12-13-73	0920	39328	14.32	138.84	116.14	22.70	1733	(0)
VPBFC2-019	12-13-73	0955	25260	14.32	138.57	116.22	22.35	1130	(0)
VPBFC2-020	12-14-73	0820	668	14.50	128.36	117.11	11.25	59	(924)
VPBFC2-021	12-15-73	1200	354	14.49	121.45	117.16	4.29	59	(424)
VPBFC2-022	12-15-73	1600	448	14.48	125.68	117.16	8.52	53	(0)

TABLE D-4 (Continued)

Run Ident.	Date	Hour	q'' (HM-2)	Meas. Press. Psia	Temperature			h Coeff.	q'' (HM-1)
			$\frac{\text{Btu}}{\text{Hr-Ft}^2}$		Wall °F	Saturation °F	Diff °F	$\frac{\text{Btu}}{\text{Hr-Ft}^2-\text{°F}}$	$\frac{\text{Btu}}{\text{Hr-Ft}^2}$
For Constant HM-2 Flux with Varying HM-1 Flux									
VPBFC2-023	12-13-73	1045	25367	14.34	138.67	116.02	22.65	1120	(45293)
VPBFC2-024	12-13-73	1120	25264	14.31	138.67	115.96	22.72	1112	(32156)
VPBFC2-025	12-13-73	1210	25386	14.31	138.69	115.99	22.71	1118	(25630)
VPBFC2-026	12-13-73	1240	25473	14.30	138.68	115.95	22.73	1121	(20227)
VPBFC2-027	12-13-73	1320	15639	14.30	138.18	115.95	22.22	704	(14912)
VPBFC2-028	12-13-73	1405	15493	14.30	138.22	116.06	22.15	699	(9620)
VPBFC2-029	12-13-73	1450	15580	14.30	138.22	116.12	22.09	705	(7231)
VPBFC2-030	12-13-73	1525	15559	14.30	138.20	116.25	21.95	709	(4886)
VPBFC2-031	12-13-73	1610	15614	14.30	138.20	116.23	21.97	711	(3009)
VPBFC2-032	12-13-73	1705	15555	14.31	138.27	116.31	21.96	708	(2096)
VPBFC2-033	12-13-73	1755	15551	14.33	138.35	116.41	21.94	709	(1315)
For Varying HM-2 Flux with Constant HM-1 Flux									
VPBFC2-034	12-14-73	0930	31982	14.50	139.65	116.80	22.85	1400	(32103)
VPBFC2-035	12-14-73	1010	22883	14.50	139.44	116.84	22.60	1013	(32059)
VPBFC2-036	12-14-73	1045	15518	14.50	139.16	116.84	22.31	695	(32014)
VPBFC2-037	12-14-73	1130	10538	14.50	138.26	116.86	21.40	493	(32023)
VPBFC2-038	12-14-73	1210	5443	14.50	133.98	116.90	17.08	319	(32225)
VPBFC2-039	12-14-73	1250	18243	14.50	139.30	116.84	22.47	812	(14806)
VPBFC2-040	12-14-73	1330	25418	14.49	139.48	116.79	22.69	1120	(14840)
VPBFC2-041	12-14-73	1400	10554	14.49	138.32	116.89	21.43	493	(14881)
VPBFC2-042	12-14-73	1440	7824	14.50	137.49	116.89	20.59	380	(14821)
VPBFC2-043	12-14-73	1515	5416	14.50	135.27	117.00	18.27	296	(14828)
VPBFC2-044	12-14-73	1615	3526	14.51	130.26	117.00	13.26	266	(14884)

TABLE D-5

REDUCED DATA FOR POOL BOILING OF WATER OF A HORIZONTAL COPPER SURFACE
USING HEAT METER HM-1

Run Ident.	Date	Hour	q'' (HM-1)	Trans. Press. Psia	Temperature			h Coeff.
			$\frac{Btu}{Hr-Ft^2}$		Wall $^{\circ}F$	Saturation $^{\circ}F$	Diff $^{\circ}F$	$\frac{Btu}{Hr-Ft^2-^{\circ}F}$
HPBWC1-001	12-21-73	1000	1615	14.48	213.68	211.01	2.67	606
HPBWC1-002	12-21-73	1100	66342	14.48	225.22	210.99	14.23	4662
HPBWC1-003	12-21-73	1140	56696	14.47	225.05	210.93	14.13	4013
HPBWC1-004	12-21-73	1225	46916	14.47	224.99	210.89	14.10	3327
HPBWC1-005	12-21-73	1305	36270	14.47	224.73	210.86	13.87	2614
HPBWC1-006	12-21-73	1350	26356	14.47	224.51	210.91	13.60	1938
HPBWC1-007	12-21-73	1430	17710	14.48	224.18	210.97	13.21	1340
HPBWC1-008	12-21-73	1505	12399	14.48	223.90	210.97	12.93	959
HPBWC1-009	12-21-73	1615	7394	14.49	223.25	211.00	12.25	604
HPBWC1-010	12-21-73	1715	5031	14.50	220.80	211.19	9.61	524
HPBWC1-011	12-21-73	1815	3204	14.51	217.13	211.10	6.03	531
HPBWC1-012	12-22-73	0830	2317	14.56	215.22	211.27	3.95	586
HPBWC1-013	12-22-73	1000	71663	14.60	226.52	211.29	15.23	4706
HPBWC1-014	12-22-73	1035	61579	14.60	226.53	211.29	15.24	4039
HPBWC1-015	12-22-73	1120	51410	14.59	226.61	211.29	15.31	3357
HPBWC1-016	12-22-73	1215	41495	14.57	226.60	211.28	15.33	2707
HPBWC1-017	12-22-73	1305	31548	14.56	226.38	211.25	15.13	2085
HPBWC1-018	12-22-73	1340	21212	14.55	226.23	211.24	14.99	1415
HPBWC1-019	12-22-73	1410	16040	14.54	225.88	211.16	14.72	1090
HPBWC1-020	12-22-73	1450	11235	14.53	225.41	211.24	14.17	793
HPBWC1-021	12-22-73	1540	4925	14.52	222.04	211.14	10.90	452
HPBWC1-022	12-22-73	1640	2827	14.52	216.86	211.17	5.69	497

TABLE D-6

REDUCED DATA FOR POOL BOILING OF WATER ON A HORIZONTAL COPPER SURFACE
USING HEAT METER HM-2

Run Ident.	Date	Hour	q'' (HM-2)	Meas. Press. Psia	Temperature			h Coeff. Btu Hr-Ft ² -°F
			$\frac{\text{Btu}}{\text{Hr-Ft}^2}$		Wall °F	Saturation °F	Diff °F	
HPBWC2-001	12-21-73	1000	1347	14.48	213.85	211.02	2.82	477
HPBWC2-002	12-21-73	1100	66330	14.48	225.92	211.00	14.92	4445
HPBWC2-003	12-21-73	1140	57507	14.47	225.65	210.94	14.71	3910
HPBWC2-004	12-21-73	1225	46769	14.47	225.50	210.98	14.52	3220
HPBWC2-005	12-21-73	1305	36310	14.47	225.21	210.94	14.27	2545
HPBWC2-006	12-21-73	1350	26641	14.47	224.89	210.98	13.91	1915
HPBWC2-007	12-21-73	1430	18272	14.48	224.51	210.99	13.52	1352
HPBWC2-008	12-21-73	1505	13117	14.48	224.20	211.00	13.20	994
HPBWC2-009	12-21-73	1615	7985	14.49	223.46	211.09	12.37	646
HPBWC2-010	12-21-73	1715	5303	14.50	222.43	211.25	11.18	474
HPBWC2-011	12-21-73	1815	3630	14.51	218.52	211.20	7.32	496
HPBWC2-012	12-22-73	0830	1992	14.56	215.54	211.29	4.30	464
HPBWC2-013	12-22-73	1000	72403	14.60	227.34	211.29	16.05	4510
HPBWC2-014	12-22-73	1035	62296	14.60	227.35	211.33	16.02	3889
HPBWC2-015	12-22-73	1120	51764	14.59	227.23	211.31	15.93	3250
HPBWC2-016	12-22-73	1215	41883	14.57	227.22	211.29	15.94	2628
HPBWC2-017	12-22-73	1305	31354	14.56	227.02	211.25	15.77	1988
HPBWC2-018	12-22-73	1340	21459	14.55	226.68	211.26	15.42	1392
HPBWC2-019	12-22-73	1410	16315	14.54	226.42	211.30	15.11	1080
HPBWC2-020	12-22-73	1450	11203	14.53	225.97	211.23	14.73	760
HPBWC2-021	12-22-73	1540	5568	14.52	221.86	211.26	10.60	525
HPBWC2-022	12-22-73	1640	2879	14.52	217.11	211.21	5.90	488

TABLE D-7

REDUCED DATA FOR POOL BOILING OF WATER ON A VERTICAL COPPER SURFACE
USING HEAT METER HM-1

Run Ident.	Date	Hour	q'' (HM-1)	Meas. Press. psia	Temperature			h Coeff. Btu Hr-Ft ² -°F
			$\frac{\text{Btu}}{\text{Hr-Ft}^2}$		Wall °F	Saturation °F	Diff °F	
VPBWC1-001	12-18-73	0820	1513	14.63	213.97	211.44	2.53	598
VPBWC1-002	12-19-73	0930	2917	14.67	216.93	211.46	5.48	533
VPBWC1-003	12-19-73	1130	71631	14.65	227.03	211.44	15.59	4594
VPBWC1-004	12-19-73	1215	61385	14.64	225.30	211.37	14.93	4111
VPBWC1-005	12-19-73	1240	51517	14.63	225.25	211.32	13.93	3698
VPBWC1-006	12-19-73	1320	41394	14.62	224.62	211.37	13.24	3126
VPBWC1-007	12-19-73	1355	31510	14.60	223.63	211.29	12.34	2554
VPBWC1-008	12-19-73	1430	21325	14.60	222.32	211.26	11.05	1930
VPBWC1-009	12-19-73	1540	36367	14.59	224.03	211.32	12.71	2862
VPBWC1-010	12-19-73	1615	26602	14.58	222.88	211.29	11.59	2295
VPBWC1-011	12-19-73	1655	16090	14.58	221.37	211.21	10.16	1583
VPBWC1-012	12-19-73	1730	11063	14.57	220.65	211.22	9.43	1173
VPBWC1-013	12-19-73	1800	5277	14.57	219.38	211.18	8.20	643
VPBWC1-014	12-20-73	0840	3268	14.46	218.21	210.82	7.40	442

TABLE D-8

REDUCED DATA FOR POOL BOILING OF WATER ON A VERTICAL COPPER SURFACE
USING HEAT METER HM-2

Run Ident.	Date	Hour	q'' (HM-2)	Meas. Press. Psia	Temperature			h Coeff.	q'' (HM-1)
			Btu Hr-Ft ²		Wall °F	Saturation °F	Diff °F	Btu Hr-Ft ² -°F	Btu Hr-Ft ²
For Varying HM-2 Flux									
VPBWC2-001	12-18-73	0820	1095	14.53	212.91	211.32	1.59	689	(1513)
VPBWC2-002	12-18-73	1000	66446	14.58	225.28	211.43	13.86	4796	(0)
VPBWC2-003	12-18-73	1100	57699	14.58	225.06	211.40	13.67	4222	(0)
VPBWC2-004	12-18-73	1120	47002	14.57	224.56	211.42	13.14	3577	(0)
VPBWC2-005	12-18-73	1145	36379	14.57	223.99	211.38	12.61	2885	(0)
VPBWC2-006	12-18-73	1215	26187	14.57	223.33	211.39	11.94	2194	(0)
VPBWC2-007	12-18-73	1245	21273	14.56	222.96	211.37	11.58	1836	(0)
VPBWC2-008	12-18-73	1315	16695	14.55	222.59	211.39	11.20	1491	(0)
VPBWC2-009	12-18-73	1350	13226	14.55	222.26	211.37	10.88	1215	(0)
VPBWC2-010	12-18-73	1450	11345	14.55	222.13	211.42	10.71	1060	(0)
VPBWC2-011	12-18-73	1520	7997	14.55	221.69	211.44	10.25	780	(0)
VPBWC2-012	12-18-73	1540	5411	14.55	219.51	211.41	8.09	669	(0)
VPBWC2-013	12-18-73	1610	4451	14.55	217.90	211.44	6.46	689	(0)
VPBWC2-014	12-19-73	0830	1839	14.57	214.35	211.40	2.96	622	(2917)
VPBWC2-015	12-19-73	0950	66330	14.57	225.88	211.42	14.46	4587	(0)
VPBWC2-016	12-19-73	1045	52088	14.57	225.06	211.44	13.62	3823	(0)
VPBWC2-017	12-19-73	1500	26336	14.49	223.28	211.20	12.08	2180	(0)
VPBWC2-018	12-20-73	0840	3555	14.36	215.73	210.78	4.95	719	(3268)

TABLE D-8 (Continued)

Run Ident.	Date	Hour	q'' (HM-2)	Meas. Press. Psia	Temperature			h Coeff.	q'' (HM-1)
			Btu		Wall	Saturation	Diff	Btu	Btu
			Hr-Ft ²		°F	°F	°F	Hr-Ft ² -°F	Hr-Ft ²
For Constant HM-2 Flux with Varying HM-1 Flux									
VPBWC2-019	12-19-73	1130	52246	14.55	225.65	211.41	14.23	3671	(71631)
VPBWC2-020	12-19-73	1215	52257	14.54	225.53	211.36	14.16	3689	(61385)
VPBWC2-021	12-19-73	1320	52276	14.52	225.50	211.30	14.21	3680	(41394)
VPBWC2-022	12-19-73	1355	52188	14.51	225.47	211.28	14.20	3676	(31510)
VPBWC2-023	12-19-73	1430	52345	14.50	225.23	211.28	13.95	3752	(21325)
VPBWC2-024	12-19-73	1540	26439	14.49	223.47	211.31	12.17	2173	(36367)
VPBWC2-025	12-19-73	1615	26266	14.48	223.46	211.22	12.24	2146	(26602)
VPBWC2-026	12-19-73	1655	26294	14.48	223.43	211.20	12.23	2150	(16090)
VPBWC2-027	12-19-73	1730	26321	14.47	223.40	211.21	12.19	2160	(11063)
VPBWC2-028	12-19-73	1800	26246	14.47	223.37	211.15	12.22	2147	(5277)

TABLE D-9

ROHSENOW AND ZUBER CORRELATION PARAMETERS FOR FULLY DEVELOPED, NUCLEATE, POOL BOILING OF
FREON-113 ON A HORIZONTAL COPPER SURFACE USING HEAT METER HM-1

Run Ident.	Rohsenow Correlation Parameters				Zuber Correlation Parameters			
	$C_p \cdot T_{diff}$	Pr	$C_p \cdot T_{diff}$	Bubble	Bubble Rad.	Bubble	Bubble	Bubble
	Hfg		Hfg · Pr ^{1/3}	Re	Ft	Nu	Nu/Pr ^{1/3}	Re
HPBFC1-02	0.06421	7.105	2.291·10 ⁻³	0.6247	0.7005·10 ⁻⁵	0.1510	0.07853	80.4
HPBFC1-03	0.06289	7.094	2.249·10 ⁻³	0.2042	0.6911·10 ⁻⁵	0.0483	0.02512	76.3
HPBFC1-04	0.06231	7.090	2.231·10 ⁻³	0.1796	0.6871·10 ⁻⁵	0.0426	0.02217	74.6
HPBFC1-05	0.06128	7.080	2.199·10 ⁻³	0.1263	0.6779·10 ⁻⁵	0.0300	0.01563	71.3
HPBFC1-06	0.05976	7.075	2.147·10 ⁻³	0.0886	0.6722·10 ⁻⁵	0.0214	0.01115	67.4
HPBFC1-10	0.06457	7.068	2.324·10 ⁻³	0.5253	0.6756·10 ⁻⁵	0.1179	0.06146	78.1
HPBFC1-11	0.06435	7.072	2.314·10 ⁻³	0.4178	0.6776·10 ⁻⁵	0.0944	0.04920	79.9
HPBFC1-12	0.06421	7.071	2.309·10 ⁻³	0.3111	0.6768·10 ⁻⁵	0.0704	0.03668	77.5
HPBFC1-13	0.06340	7.070	2.281·10 ⁻³	0.2097	0.6749·10 ⁻⁵	0.0479	0.02496	75.5
HPBFC1-14	0.07750	7.078	2.783·10 ⁻³	1.9220	0.7010·10 ⁻⁵	0.3733	0.19450	113.8
HPBFC1-15	0.07521	7.074	2.703·10 ⁻³	1.3620	0.6949·10 ⁻⁵	0.2703	0.14080	106.6
HPBFC1-16	0.07381	7.072	2.654·10 ⁻³	1.0870	0.6923·10 ⁻⁵	0.2189	0.11400	102.6
HPBFC1-17	0.07281	7.069	2.620·10 ⁻³	0.9729	0.6886·10 ⁻⁵	0.1975	0.10290	99.5
HPBFC1-18	0.07223	7.069	2.599·10 ⁻³	0.8567	0.6877·10 ⁻⁵	0.1750	0.09121	97.9
HPBFC1-19	0.07182	7.068	2.585·10 ⁻³	0.7377	0.6864·10 ⁻⁵	0.1513	0.07884	96.6
HPBFC1-20	0.07086	7.067	2.551·10 ⁻³	0.5779	0.6846·10 ⁻⁵	0.1198	0.06244	94.0
HPBFC1-21	0.05930	7.064	2.497·10 ⁻³	0.3538	0.6798·10 ⁻⁵	0.0744	0.03880	89.6
HPBFC1-23	0.08033	7.066	2.893·10 ⁻³	1.9150	0.6956·10 ⁻⁵	0.3557	0.18540	120.6
HPBFC1-24	0.07832	7.063	2.822·10 ⁻³	1.3680	0.6916·10 ⁻⁵	0.2591	0.13500	114.3
HPBFC1-25	0.07632	7.063	2.750·10 ⁻³	0.9714	0.6886·10 ⁻⁵	0.1880	0.09800	108.5
HPBFC1-26	0.07404	7.063	2.668·10 ⁻³	0.6320	0.6860·10 ⁻⁵	0.1256	0.06547	102.1
HPBFC1-27	0.07247	7.063	2.611·10 ⁻³	0.4119	0.6839·10 ⁻⁵	0.0834	0.04346	97.9
HPBFC1-28	0.07154	7.063	2.578·10 ⁻³	0.3084	0.6827·10 ⁻⁵	0.0631	0.03291	95.4

TABLE D-9 (Continued)

Run Ident.	Rohsenow Correlation Parameters				Zuber Correlation Parameters			
	$\frac{Cp \cdot T_{diff}}{Hfg}$	Pr	$\frac{Cp \cdot T_{diff}}{Hfg \cdot Pr^{1/4}}$	Bubble Re	Bubble Rad. Ft	Bubble Nu	Bubble Nu/Pr ^{1/4}	Bubble Re
HPBFC1-29	0.07038	7.062	$2.537 \cdot 10^{-3}$	0.2036	$0.6802 \cdot 10^{-8}$	0.0422	0.02200	92.2
HPBFC1-30	0.06824	7.063	$2.459 \cdot 10^{-3}$	0.1259	$0.6778 \cdot 10^{-8}$	0.0268	0.01398	86.8
HPBFC1-31	0.06176	7.058	$2.227 \cdot 10^{-3}$	0.0835	$0.6650 \cdot 10^{-8}$	0.0197	0.01027	70.8

TABIE 7-10

ROHSENOW AND ZUBER CORRELATION PARAMETERS FOR FULLY DEVELOPED, NUCLEATE, POOL BOILING OF
FREON-113 ON A HORIZONTAL COPPER SURFACE USING HEAT METER HM-2

Run Ident.	Rohsenow Correlation Parameters				Zuber Correlation Parameters			
	$\frac{Cp \cdot T_{diff}}{Hfg}$	Pr	$\frac{Cp \cdot T_{diff}}{Hfg \cdot Pr^{1/4}}$	Bubble Re	Bubble Rad. Ft	Bubble Nu	Bubble $Nu/Pr^{1/3}$	Bubble Re
HPBFC2-04	0.06637	7.104	$2.368 \cdot 10^{-3}$	0.6729	$0.7035 \cdot 10^{-5}$	0.1535	0.07988	85.8
HPBFC2-05	0.06397	7.093	$2.289 \cdot 10^{-3}$	0.2298	$0.6916 \cdot 10^{-5}$	0.0534	0.02782	78.8
HPBFC2-06	0.06337	7.090	$2.269 \cdot 10^{-3}$	0.1870	$0.6886 \cdot 10^{-5}$	0.0437	0.02274	77.1
HPBFC2-07	0.06263	7.080	$2.248 \cdot 10^{-3}$	0.1440	$0.6804 \cdot 10^{-5}$	0.0336	0.01750	74.5
HPBFC2-08	0.06126	7.072	$2.203 \cdot 10^{-3}$	0.0982	$0.6723 \cdot 10^{-5}$	0.0231	0.01205	70.6
HPBFC2-12	0.06572	7.065	$2.367 \cdot 10^{-3}$	0.5622	$0.6750 \cdot 10^{-5}$	0.1239	0.06456	80.6
HPBFC2-13	0.06549	7.068	$2.357 \cdot 10^{-3}$	0.4511	$0.6772 \cdot 10^{-5}$	0.1001	0.05217	80.4
HPBFC2-14	0.06516	7.068	$2.345 \cdot 10^{-3}$	0.3396	$0.6763 \cdot 10^{-5}$	0.0756	0.03942	79.5
HPBFC2-15	0.06455	7.068	$2.323 \cdot 10^{-3}$	0.2250	$0.6755 \cdot 10^{-5}$	0.0505	0.02633	78.1
HPBFC2-16	0.07797	7.077	$2.800 \cdot 10^{-3}$	1.6630	$0.7011 \cdot 10^{-5}$	0.3212	0.16730	115.1
HPBFC2-17	0.07577	7.071	$2.725 \cdot 10^{-3}$	1.2920	$0.6938 \cdot 10^{-5}$	0.2540	0.13230	107.9
HPBFC2-18	0.07468	7.070	$2.686 \cdot 10^{-3}$	1.0770	$0.6919 \cdot 10^{-5}$	0.2142	0.11160	104.8
HPBFC2-19	0.07418	7.069	$2.669 \cdot 10^{-3}$	0.9726	$0.6907 \cdot 10^{-5}$	0.1943	0.10130	103.3
HPBFC2-20	0.07348	7.068	$2.645 \cdot 10^{-3}$	0.8727	$0.6885 \cdot 10^{-5}$	0.1755	0.09184	101.1
HPBFC2-21	0.07317	7.066	$2.635 \cdot 10^{-3}$	0.7584	$0.6866 \cdot 10^{-5}$	0.1527	0.07958	100.0
HPBFC2-22	0.07144	7.063	$2.574 \cdot 10^{-3}$	0.5934	$0.6825 \cdot 10^{-5}$	0.1216	0.06339	95.1
HPBFC2-23	0.07013	7.060	$2.529 \cdot 10^{-3}$	0.4218	$0.6787 \cdot 10^{-5}$	0.0875	0.04563	91.4
HPBFC2-25	0.07956	7.061	$2.868 \cdot 10^{-3}$	1.6620	$0.6917 \cdot 10^{-5}$	0.3099	0.16150	117.7
HPBFC2-26	0.07877	7.063	$2.838 \cdot 10^{-3}$	1.3190	$0.6921 \cdot 10^{-5}$	0.2487	0.12960	115.7
HPBFC2-27	0.07662	7.060	$2.763 \cdot 10^{-3}$	0.9717	$0.6872 \cdot 10^{-5}$	0.1869	0.09744	109.0
HPBFC2-28	0.07456	7.059	$2.689 \cdot 10^{-3}$	0.6617	$0.6838 \cdot 10^{-5}$	0.1301	0.06784	103.1
HPBFC2-29	0.07256	7.060	$2.616 \cdot 10^{-3}$	0.4566	$0.6819 \cdot 10^{-5}$	0.0920	0.04797	97.8
HPBFC2-30	0.07185	7.060	$2.591 \cdot 10^{-3}$	0.3396	$0.6807 \cdot 10^{-5}$	0.0690	0.03597	95.8

TABLE D-10 (Continued)

Run Ident.	Rohsenow Correlation Parameters				Zuber Correlation Parameters			
	$\frac{Cp \cdot T_{diff}}{h_{fg}}$	Pr	$\frac{Cp \cdot T_{diff}}{h_{fg} \cdot Pr^{1/4}}$	Bubble Re	Bubble Rad. Ft	Bubble Nu	Bubble Nu/Pr ^{1/4}	Bubble Re
HPBFC2-31	0.07036	7.059	$2.538 \cdot 10^{-3}$	0.2287	$0.6780 \cdot 10^{-3}$	0.0472	0.02463	91.8
HPBFC2-32	0.06896	7.061	$2.486 \cdot 10^{-3}$	0.1405	$0.6772 \cdot 10^{-3}$	0.0296	0.01543	88.4
HPBFC2-33	0.06815	7.061	$2.457 \cdot 10^{-3}$	0.0918	$0.6763 \cdot 10^{-3}$	0.0195	0.01019	86.4

TABLE D-11

ROHSENOW AND ZUBER CORRELATION PARAMETERS FOR FULLY DEVELOPED, NUCLEATE, POOL BOILING OF
FREON-113 ON A VERTICAL COPPER SURFACE USING HEAT METER HM-1

Run Ident.	Rohsenow Correlation Parameters				Zuber Correlation Parameters			
	$Cp \cdot T_{diff}$ Hfg	Pr	$Cp \cdot T_{diff}$ Hfg $\cdot Pr^{1/4}$	Bubble Re	Bubble Re Pr	Bubble Nu	Bubble Nu/Pr ^{1/3}	Bubble Re
VPBFC1-03	0.08208	7.107	$2.927 \cdot 10^{-3}$	1.9050	$0.7274 \cdot 10^{-8}$	0.3636	0.18910	138.1
VPBFC1-04	0.07931	7.108	$2.827 \cdot 10^{-3}$	1.3520	$0.7251 \cdot 10^{-8}$	0.2663	0.13850	123.2
VPBFC1-05	0.07749	7.107	$2.763 \cdot 10^{-3}$	1.0780	$0.7222 \cdot 10^{-8}$	0.2164	0.11260	117.5
VPBFC1-06	0.07608	7.110	$2.711 \cdot 10^{-3}$	0.8504	$0.7222 \cdot 10^{-8}$	0.1739	0.09044	113.6
VPBFC1-07	0.07344	7.109	$2.617 \cdot 10^{-3}$	0.6270	$0.7179 \cdot 10^{-8}$	0.1320	0.06867	105.7
VPBFC1-08	0.06944	7.106	$2.476 \cdot 10^{-3}$	0.4047	$0.7101 \cdot 10^{-8}$	0.0891	0.04636	94.2
VPBFC1-09	0.06700	7.103	$2.391 \cdot 10^{-3}$	0.3044	$0.7042 \cdot 10^{-8}$	0.0689	0.03583	87.4
VPBFC1-10	0.06318	7.100	$2.256 \cdot 10^{-3}$	0.2058	$0.6955 \cdot 10^{-8}$	0.0488	0.02538	77.5
VPBFC1-11	0.05782	7.099	$2.065 \cdot 10^{-3}$	0.1268	$0.6851 \cdot 10^{-8}$	0.0323	0.01682	64.8
VPBFC1-12	0.05457	7.098	$1.950 \cdot 10^{-3}$	0.0883	$0.6777 \cdot 10^{-8}$	0.0236	0.01228	57.7
VPBFC1-16	0.07930	7.084	$2.843 \cdot 10^{-3}$	1.3570	$0.7075 \cdot 10^{-8}$	0.2602	0.13550	119.9
VPBFC1-17	0.07951	7.082	$2.852 \cdot 10^{-3}$	1.3560	$0.7062 \cdot 10^{-8}$	0.2587	0.13470	120.3
VPBFC1-18	0.07983	7.082	$2.864 \cdot 10^{-3}$	1.3540	$0.7065 \cdot 10^{-8}$	0.2574	0.13410	121.2
VPBFC1-19	0.08023	7.082	$2.878 \cdot 10^{-3}$	1.3540	$0.7071 \cdot 10^{-8}$	0.2564	0.13350	122.5
VPBFC1-20	0.08014	7.080	$2.876 \cdot 10^{-3}$	1.3630	$0.7056 \cdot 10^{-8}$	0.2578	0.13430	121.9
VPBFC1-21	0.07675	7.081	$2.753 \cdot 10^{-3}$	0.6261	$0.7024 \cdot 10^{-8}$	0.1231	0.06412	112.0
VPBFC1-22	0.07689	7.084	$2.757 \cdot 10^{-3}$	0.6273	$0.7042 \cdot 10^{-8}$	0.1235	0.06429	112.7
VPBFC1-23	0.07655	7.081	$2.747 \cdot 10^{-3}$	0.6294	$0.7017 \cdot 10^{-8}$	0.1240	0.06456	111.3
VPBFC1-24	0.07660	7.080	$2.749 \cdot 10^{-3}$	0.6269	$0.7014 \cdot 10^{-8}$	0.1233	0.06422	111.4
VPBFC1-25	0.07666	7.078	$2.753 \cdot 10^{-3}$	0.6275	$0.6997 \cdot 10^{-8}$	0.1230	0.06408	111.3
VPBFC1-26	0.07683	7.078	$2.759 \cdot 10^{-3}$	0.6299	$0.6999 \cdot 10^{-8}$	0.1233	0.06420	111.8

TABLE D-12

ROHSENOW AND ZUBER CORRELATION PARAMETERS FOR FULLY DEVELOPED, NUCLEATE, POOL BOILING OF
FREON-113 ON A VERTICAL COPPER SURFACE USING HEAT METER HM-2

Run Ident.	Rohsenow Correlation Parameters				Zuber Correlation Parameters			
	$C_p \cdot T_{diff}$	Pr	$C_p \cdot T_{diff}$	Bubble Re	Bubble Rad.	Bubble	Bubble	Bubble
	Hfg		Hfg · Pr ^{1/4}		ft	Nu	Nu/Pr ^{1/3}	Re
VPBFC2-01	0.08258	7.071	2.970·10 ⁻³	1.6600	0.7021·10 ⁻⁵	0.3030	0.15790	128.2
VPBFC2-02	0.07869	7.073	2.828·10 ⁻³	1.2760	0.6992·10 ⁻⁵	0.2434	0.12680	116.7
VPBFC2-03	0.07813	7.075	2.807·10 ⁻³	0.8767	0.6995·10 ⁻⁵	0.1685	0.08780	115.2
VPBFC2-04	0.07664	7.077	2.753·10 ⁻³	0.6604	0.6992·10 ⁻⁵	0.1294	0.06740	111.2
VPBFC2-05	0.07210	7.071	2.593·10 ⁻³	0.4483	0.6892·10 ⁻⁵	0.0920	0.04793	97.8
VPBFC2-06	0.07014	7.072	2.522·10 ⁻³	0.3354	0.6868·10 ⁻⁵	0.0705	0.03674	92.6
VPBFC2-07	0.08199	7.078	2.944·10 ⁻³	1.6600	0.7062·10 ⁻⁵	0.3083	0.16060	127.3
VPBFC2-08	0.08152	7.077	2.928·10 ⁻³	1.2800	0.7049·10 ⁻⁵	0.2378	0.12380	125.7
VPBFC2-09	0.07987	7.075	2.869·10 ⁻³	0.8720	0.7021·10 ⁻⁵	0.1646	0.08574	120.5
VPBFC2-10	0.07817	7.077	2.807·10 ⁻³	0.6614	0.7013·10 ⁻⁵	0.1275	0.06639	115.7
VPBFC2-11	0.07463	7.076	2.681·10 ⁻³	0.4453	0.6959·10 ⁻⁵	0.0892	0.04645	105.3
VPBFC2-12	0.07228	7.075	2.597·10 ⁻³	0.3359	0.6920·10 ⁻⁵	0.0690	0.03597	98.6
VPBFC2-13	0.06899	7.075	2.479·10 ⁻³	0.2265	0.6874·10 ⁻⁵	0.0485	0.02525	89.9
VPBFC2-14	0.06591	7.076	2.368·10 ⁻³	0.1348	0.6830·10 ⁻⁵	0.0300	0.01562	82.1
VPBFC2-15	0.06416	7.078	2.304·10 ⁻³	0.0998	0.6817·10 ⁻⁵	0.0228	0.01187	78.0
VPBFC2-18	0.08400	7.103	2.998·10 ⁻³	1.6560	0.7269·10 ⁻⁵	0.3085	0.16050	137.4
VPBFC2-19	0.08273	7.101	2.954·10 ⁻³	1.0640	0.7239·10 ⁻⁵	0.2004	0.10420	133.0
VPBFC2-23	0.08381	7.107	2.988·10 ⁻³	1.0670	0.7295·10 ⁻⁵	0.2000	0.10400	137.4
VPBFC2-24	0.08403	7.109	2.995·10 ⁻³	1.0620	0.7310·10 ⁻⁵	0.1991	0.10350	138.4
VPBFC2-25	0.08400	7.108	2.994·10 ⁻³	1.0680	0.7303·10 ⁻⁵	0.1999	0.10400	138.1
VPBFC2-26	0.08408	7.109	2.997·10 ⁻³	1.0710	0.7312·10 ⁻⁵	0.2006	0.10430	138.6
VPBFC2-27	0.08221	7.109	2.930·10 ⁻³	0.6576	0.7291·10 ⁻⁵	0.1256	0.06533	132.5
VPBFC2-28	0.08197	7.106	2.924·10 ⁻³	0.6519	0.7263·10 ⁻⁵	0.1244	0.06470	131.2

TABLE D-12 (Continued)

Run Ident.	Rohsenow Correlation Parameters				Zuber Correlation Parameters			
	$Cp \cdot T_{diff}$	Pr	$Cp \cdot T_{diff}$	Bubble Re	Bubble Rad.	Bubble Nu	Bubble $Nu/Pr^{1/3}$	Bubble Re
	Hfg		$Hfg \cdot Pr^{1/3}$		Pt			
VPBFC2-29	0.08176	7.104	$2.917 \cdot 10^{-3}$	0.6558	$0.7248 \cdot 10^{-3}$	0.1252	0.06511	130.3
VPBFC2-30	0.08124	7.100	$2.901 \cdot 10^{-3}$	0.6555	$0.7213 \cdot 10^{-3}$	0.1253	0.06517	128.1
VPBFC2-31	0.08130	7.101	$2.903 \cdot 10^{-3}$	0.6577	$0.7219 \cdot 10^{-3}$	0.1257	0.06540	128.4
VPBFC2-32	0.08127	7.098	$2.904 \cdot 10^{-3}$	0.6555	$0.7200 \cdot 10^{-3}$	0.1250	0.06503	127.9
VPBFC2-33	0.08123	7.095	$2.904 \cdot 10^{-3}$	0.6558	$0.7179 \cdot 10^{-3}$	0.1247	0.06489	127.4
VPBFC2-34	0.08465	7.084	$3.035 \cdot 10^{-3}$	1.3520	$0.7132 \cdot 10^{-3}$	0.2448	0.12750	136.6
VPBFC2-35	0.08373	7.082	$3.003 \cdot 10^{-3}$	0.9675	$0.7114 \cdot 10^{-3}$	0.1766	0.09199	133.4
VPBFC2-36	0.08268	7.082	$2.965 \cdot 10^{-3}$	0.6561	$0.7103 \cdot 10^{-3}$	0.1211	0.06307	130.1
VPBFC2-37	0.07928	7.082	$2.844 \cdot 10^{-3}$	0.4456	$0.7059 \cdot 10^{-3}$	0.0852	0.04440	119.6
VPBFC2-38	0.06329	7.081	$2.271 \cdot 10^{-3}$	0.2302	$0.6821 \cdot 10^{-3}$	0.0533	0.02776	76.1
VPBFC2-39	0.08324	7.083	$2.986 \cdot 10^{-3}$	0.7713	$0.7110 \cdot 10^{-3}$	0.1416	0.07372	131.9
VPBFC2-40	0.08406	7.084	$3.014 \cdot 10^{-3}$	1.0740	$0.7123 \cdot 10^{-3}$	0.1958	0.10200	134.7
VPBFC2-41	0.07940	7.081	$2.849 \cdot 10^{-3}$	0.4464	$0.7054 \cdot 10^{-3}$	0.0852	0.04437	119.8
VPBFC2-42	0.07630	7.081	$2.738 \cdot 10^{-3}$	0.3309	$0.7015 \cdot 10^{-3}$	0.0654	0.03404	110.6
VPBFC2-43	0.06773	7.078	$2.432 \cdot 10^{-3}$	0.2292	$0.6873 \cdot 10^{-3}$	0.0500	0.02602	86.9
VPBFC2-44	0.04913	7.078	$1.764 \cdot 10^{-3}$	0.1492	$0.6516 \cdot 10^{-3}$	0.0425	0.02214	45.7

TABLE D-13

ROHSENOW AND ZUBER CORRELATION PARAMETERS FOR FULLY DEVELOPED, NUCLEATE, POOL BOILING OF WATER ON A HORIZONTAL COPPER SURFACE USING HEAT METER HM-1

Run Ident.	Rohsenow Correlation Parameters				Zuber Correlation Parameters			
	$\frac{C_p \Delta T_{diff}}{h_{fg}}$	Pr	$\frac{C_p \Delta T_{diff}}{h_{fg} Pr}$	Bubble Re	Bubble Rad. Ft	Bubble Nu	Bubble $Nu/Pr^{1/3}$	Bubble Re
HPBWC1-02	0.01476	1.740	$8.484 \cdot 10^{-3}$	0.9647	$5.4080 \cdot 10^{-8}$	0.6413	0.53320	1051.0
HPBWC1-03	0.01465	1.741	$8.419 \cdot 10^{-3}$	0.8241	$5.4120 \cdot 10^{-8}$	0.5524	0.45920	1037.0
HPBWC1-04	0.01463	1.741	$8.402 \cdot 10^{-3}$	0.6818	$5.4150 \cdot 10^{-8}$	0.4583	0.38100	1035.0
HPBWC1-05	0.01439	1.741	$8.264 \cdot 10^{-3}$	0.5269	$5.4070 \cdot 10^{-8}$	0.3596	0.29890	1002.0
HPBWC1-06	0.01411	1.741	$8.106 \cdot 10^{-3}$	0.3830	$5.3810 \cdot 10^{-8}$	0.2652	0.22040	962.3
HPBWC1-07	0.01371	1.740	$7.878 \cdot 10^{-3}$	0.2575	$5.3460 \cdot 10^{-8}$	0.1822	0.15150	906.6
HPBWC1-08	0.01341	1.740	$7.708 \cdot 10^{-3}$	0.1803	$5.3250 \cdot 10^{-8}$	0.1299	0.10800	867.7
HPBWC1-09	0.01270	1.740	$7.302 \cdot 10^{-3}$	0.1075	$5.2710 \cdot 10^{-8}$	0.0809	0.06730	777.9
HPBWC1-13	0.01580	1.737	$9.098 \cdot 10^{-3}$	1.0440	$5.4180 \cdot 10^{-8}$	0.6484	0.53940	1192.0
HPBWC1-14	0.01582	1.737	$9.108 \cdot 10^{-3}$	0.8972	$5.4190 \cdot 10^{-8}$	0.5567	0.46310	1195.0
HPBWC1-15	0.01589	1.737	$9.150 \cdot 10^{-3}$	0.7491	$5.4210 \cdot 10^{-8}$	0.4628	0.38510	1205.0
HPBWC1-16	0.01590	1.737	$9.156 \cdot 10^{-3}$	0.6046	$5.4250 \cdot 10^{-8}$	0.3735	0.31070	1208.0
HPBWC1-17	0.01570	1.737	$9.038 \cdot 10^{-3}$	0.4595	$5.4190 \cdot 10^{-8}$	0.2873	0.23900	1179.0
HPBWC1-18	0.01556	1.737	$8.951 \cdot 10^{-3}$	0.3090	$5.4130 \cdot 10^{-8}$	0.1948	0.16200	1158.0
HPBWC1-19	0.01527	1.738	$8.784 \cdot 10^{-3}$	0.2335	$5.4090 \cdot 10^{-8}$	0.1499	0.12470	1118.0
HPBWC1-20	0.01470	1.737	$8.463 \cdot 10^{-3}$	0.1637	$5.3630 \cdot 10^{-8}$	0.1081	0.08996	1034.0

TABLE D-14

ROHSPNOW AND ZUBER CORRELATION PARAMETERS FOR FULLY DEVELOPED, NUCLEATE, POOL BOILING OF WATER ON A HORIZONTAL COPPER SURFACE USING HEAT METER HM-2

Run Ident.	Pohsenow Correlation Parameters				Zuber Correlation Parameters			
	$Cp \cdot T_{diff}$ Hfg	Pr	$Cp \cdot T_{diff}$ Hfg · Pr	Bubble Re	Bubble Rad. P _t	Bubble Nu	Bubble Nu/Pr ^{1/3}	Bubble Re
HPBWC2-02	0.01548	1.740	8.898·10 ⁻³	0.8272	4.6710·10 ⁻⁸	0.5281	0.43910	1155.0
HPBWC2-03	0.01526	1.740	8.766·10 ⁻³	0.7169	4.6690·10 ⁻⁸	0.4644	0.38610	1124.0
HPBWC2-04	0.01507	1.740	8.659·10 ⁻³	0.5832	4.6550·10 ⁻⁸	0.3812	0.31700	1095.0
HPBWC2-05	0.01480	1.740	8.504·10 ⁻³	0.4527	4.6460·10 ⁻⁸	0.3008	0.25000	1057.0
HPBWC2-06	0.01443	1.740	8.295·10 ⁻³	0.3322	4.6220·10 ⁻⁸	0.2251	0.18720	1005.0
HPBWC2-07	0.01402	1.740	8.060·10 ⁻³	0.2279	4.5990·10 ⁻⁸	0.1581	0.13150	948.3
HPBWC2-08	0.01369	1.740	7.871·10 ⁻³	0.1636	4.5790·10 ⁻⁸	0.1157	0.09622	903.8
HPBWC2-09	0.01283	1.739	7.378·10 ⁻³	0.0996	4.5150·10 ⁻⁸	0.0741	0.06166	790.9
HPBWC2-13	0.01666	1.737	9.591·10 ⁻³	0.9045	4.6820·10 ⁻⁸	0.5371	0.44680	1325.0
HPBWC2-14	0.01662	1.736	9.572·10 ⁻³	0.7784	4.6750·10 ⁻⁸	0.4624	0.38480	1317.0
HPBWC2-15	0.01653	1.736	9.517·10 ⁻³	0.6467	4.6740·10 ⁻⁸	0.3863	0.32140	1303.0
HPBWC2-16	0.01653	1.737	9.521·10 ⁻³	0.5232	4.6780·10 ⁻⁸	0.3127	0.26010	1306.0
HPBWC2-17	0.01636	1.737	9.419·10 ⁻³	0.3916	4.6760·10 ⁻⁸	0.2364	0.19670	1280.0
HPBWC2-18	0.01600	1.737	9.211·10 ⁻³	0.2680	4.6570·10 ⁻⁸	0.1648	0.13710	1223.0
HPBWC2-19	0.01568	1.736	9.030·10 ⁻³	0.2038	4.6370·10 ⁻⁸	0.1273	0.10590	1174.0
HPBWC2-20	0.01528	1.737	8.798·10 ⁻³	0.1399	4.6280·10 ⁻⁸	0.0895	0.07446	1118.0

TABLE D-15

ROHSENOW AND ZUBER CORRELATION PARAMETERS FOR FULLY DEVELOPED, NUCLEATE, POOL BOILING OF WATER ON A VERTICAL COPPER SURFACE USING HEAT METER HM-1

Run Ident.	Rohsenow Correlation Parameters				Zuber Correlation Parameters			
	$\frac{C_p \Delta T_{diff}}{h_{fg}}$	Pr	$\frac{C_p \Delta T_{diff}}{h_{fg} Pr}$	Bubble Re	Bubble Rad. Pe	Bubble Nu	Bubble Nu/Pr ^{1/3}	Bubble Re
VPBWC1-03	0.01618	1.735	$9.326 \cdot 10^{-3}$	1.0450	$5.4130 \cdot 10^{-8}$	0.6324	0.52630	1244.0
VPBWC1-04	0.01549	1.736	$8.926 \cdot 10^{-3}$	0.8949	$5.3980 \cdot 10^{-8}$	0.5633	0.46870	1144.0
VPBWC1-05	0.01445	1.736	$8.324 \cdot 10^{-3}$	0.7508	$5.3360 \cdot 10^{-8}$	0.5019	0.41760	996.8
VPBWC1-06	0.01374	1.736	$7.917 \cdot 10^{-3}$	0.6035	$5.2930 \cdot 10^{-8}$	0.4199	0.34940	899.4
VPBWC1-07	0.01280	1.737	$7.370 \cdot 10^{-3}$	0.4591	$5.2310 \cdot 10^{-8}$	0.3399	0.28270	782.1
VPBWC1-08	0.01147	1.737	$6.601 \cdot 10^{-3}$	0.3107	$5.1150 \cdot 10^{-8}$	0.2520	0.20960	628.2
VPBWC1-09	0.01319	1.736	$7.594 \cdot 10^{-3}$	0.5300	$5.2540 \cdot 10^{-8}$	0.3824	0.31820	829.4
VPBWC1-10	0.01203	1.737	$6.925 \cdot 10^{-3}$	0.3876	$5.1750 \cdot 10^{-8}$	0.3021	0.25130	690.7
VPBWC1-11	0.01054	1.738	$6.068 \cdot 10^{-3}$	0.2343	$5.0650 \cdot 10^{-8}$	0.2040	0.16970	532.3
VPBWC1-12	0.00979	1.737	$5.632 \cdot 10^{-3}$	0.1611	$4.9910 \cdot 10^{-8}$	0.1489	0.12390	458.3

TABLE D-16

ROHSENOW AND ZUBER CORRELATION PARAMETERS FOR FULLY DEVELOPED, NUCLEATE, POOL BOILING OF WATER ON A VERTICAL COPPER SURFACE USING HEAT METER NH-2

Run Ident.	Rohsenow Correlation Parameters				Zuber Correlation Parameters			
	$Cp \cdot T_{diff}$	Pr	$Cp \cdot T_{diff}$	Bubble Re	Bubble Rad. Wt	Bubble Wu	Bubble $Wu/Pr^{1/3}$	Bubble Re
	Hfg		$Hfg \cdot Pr$					
VPBWC2-02	0.01438	1.735	$8.286 \cdot 10^{-3}$	0.8307	$4.5550 \cdot 10^{-8}$	0.5556	0.46230	982.7
VPBWC2-03	0.01418	1.735	$8.171 \cdot 10^{-3}$	0.7212	$4.5490 \cdot 10^{-8}$	0.4885	0.40650	957.0
VPBWC2-04	0.01364	1.735	$7.858 \cdot 10^{-3}$	0.5876	$4.5160 \cdot 10^{-8}$	0.4108	0.34190	884.0
VPBWC2-05	0.01309	1.736	$7.540 \cdot 10^{-3}$	0.4547	$4.4900 \cdot 10^{-8}$	0.3294	0.27410	815.4
VPBWC2-06	0.01239	1.736	$7.137 \cdot 10^{-3}$	0.3273	$4.4460 \cdot 10^{-8}$	0.2480	0.20640	730.4
VPBWC2-07	0.01202	1.736	$6.925 \cdot 10^{-3}$	0.2659	$4.4250 \cdot 10^{-8}$	0.2067	0.17200	688.2
VPBWC2-08	0.01162	1.736	$6.694 \cdot 10^{-3}$	0.2087	$4.3950 \cdot 10^{-8}$	0.1667	0.13870	642.3
VPBWC2-09	0.01129	1.736	$6.506 \cdot 10^{-3}$	0.1653	$4.3760 \cdot 10^{-8}$	0.1352	0.11250	607.4
VPBWC2-10	0.01111	1.735	$6.402 \cdot 10^{-3}$	0.1418	$4.3550 \cdot 10^{-8}$	0.1174	0.09768	586.8
VPBWC2-11	0.01064	1.735	$6.132 \cdot 10^{-3}$	0.1000	$4.3190 \cdot 10^{-8}$	0.0857	0.07129	538.1
VPBWC2-15	0.01501	1.735	$8.648 \cdot 10^{-3}$	0.8293	$4.5880 \cdot 10^{-8}$	0.5352	0.44530	1071.0
VPBWC2-16	0.01414	1.735	$8.147 \cdot 10^{-3}$	0.6513	$4.5410 \cdot 10^{-8}$	0.4416	0.36750	949.8
VPBWC2-17	0.01254	1.738	$7.214 \cdot 10^{-3}$	0.3288	$4.4820 \cdot 10^{-8}$	0.2485	0.20670	752.4
VPBWC2-19	0.01477	1.735	$8.511 \cdot 10^{-3}$	0.6531	$4.5770 \cdot 10^{-8}$	0.4273	0.35560	1037.0
VPBWC2-20	0.01470	1.736	$8.467 \cdot 10^{-3}$	0.6531	$4.5810 \cdot 10^{-8}$	0.4298	0.35760	1029.0
VPBWC2-21	0.01474	1.737	$8.488 \cdot 10^{-3}$	0.6531	$4.5920 \cdot 10^{-8}$	0.4298	0.35760	1037.0
VPBWC2-22	0.01473	1.737	$8.480 \cdot 10^{-3}$	0.6519	$4.5950 \cdot 10^{-8}$	0.4296	0.35740	1036.0
VPBWC2-23	0.01447	1.737	$8.334 \cdot 10^{-3}$	0.6539	$4.5820 \cdot 10^{-8}$	0.4372	0.36370	1001.0
VPBWC2-24	0.01262	1.736	$7.270 \cdot 10^{-3}$	0.3303	$4.4730 \cdot 10^{-8}$	0.2472	0.20560	760.6
VPBWC2-25	0.01270	1.737	$7.309 \cdot 10^{-3}$	0.3280	$4.4890 \cdot 10^{-8}$	0.2450	0.20380	771.7
VPBWC2-26	0.01269	1.738	$7.301 \cdot 10^{-3}$	0.3283	$4.4190 \cdot 10^{-8}$	0.2456	0.20430	770.6
VPBWC2-27	0.01264	1.737	$7.275 \cdot 10^{-3}$	0.3287	$4.4470 \cdot 10^{-8}$	0.2465	0.20510	764.8
VPBWC2-28	0.01268	1.738	$7.296 \cdot 10^{-3}$	0.3276	$4.4980 \cdot 10^{-8}$	0.2456	0.20430	771.5

TABLE D-17
REDUCED DATA FOR FORCED UPFLOW CONVECTION OF WATER AND PREOP-113 - HEAT METER HM-1

Run Ident.	q'' (HM-1)	Temperature		h Coeff.		G Flux		Re	Pr	Nu	Visc. Ratio
	$\frac{\text{Btu}}{\text{Hr-Ft}^2}$	Wall °F	Bulk °F	$\frac{\text{Btu}}{\text{Hr-Ft}^2-\text{°F}}$	$\frac{\text{Ft}}{\text{Sec}}$	$\frac{10^6 \text{ lbm}}{\text{Hr-Ft}^2}$					
For Water on Nickel											
SPCWN1-001	24628	229.97	214.75	1618	1.962	0.4220	45365	1.700	294	1.088	
SPCWN1-002	38575	238.79	214.86	1612	1.953	0.4201	45189	1.699	292	1.139	
SPCWN1-003	49415	245.73	214.76	1596	1.944	0.4182	44964	1.700	290	1.181	
SPCWN1-004	21097	227.50	215.19	1714	2.005	0.4313	46481	1.696	311	1.071	
SPCWN1-005	20060	227.06	215.15	1685	1.991	0.4282	46135	1.696	306	1.069	
SPCWN1-006	34554	252.09	214.98	931	0.987	0.2122	22845	1.698	169	1.217	
SPCWN1-007	29859	247.20	215.00	927	0.992	0.2135	22979	1.698	168	1.188	
SPCWN1-008	19958	236.40	215.03	934	0.992	0.2135	22984	1.697	169	1.124	
SPCWN1-009	24687	253.45	214.94	641	0.494	0.1063	11443	1.698	116	1.225	
SPCWN1-010	19851	246.02	214.92	638	0.494	0.1063	11442	1.698	116	1.181	
SPCWN1-011	14665	237.94	214.92	637	0.491	0.1057	11376	1.698	116	1.134	
SPCWN1-012	40189	236.52	215.12	1878	2.987	0.6425	69219	1.696	341	1.124	
SPCWN1-013	30278	231.22	215.06	1873	3.002	0.6456	69529	1.697	340	1.094	
SPCWN1-014	20323	225.64	214.98	1907	3.002	0.6456	69498	1.698	346	1.062	
SPCWN1-015	14950	222.80	214.98	1913	3.016	0.6487	69630	1.698	347	1.045	
SPCWN1-016	25096	241.96	214.95	929	0.992	0.2135	22972	1.698	169	1.157	
SPCWN1-017	14850	230.87	214.92	931	0.987	0.2122	22836	1.698	169	1.092	
For Water on Copper											
SPCWC1-001	25102	228.75	215.00	1826	2.987	0.6425	69171	1.698	331	1.079	
SPCWC1-002	20178	225.97	215.04	1847	2.987	0.6425	69189	1.697	335	1.063	

TABLE D-17 (Continued)

Run Ident.	q'' (HM-1)	Temperature		h	Coeff.	Vel.	G	Flux				Visc.
	$\frac{\text{Btu}}{\text{Hr-Ft}^2}$	Wall	Bulk									
		$^{\circ}\text{F}$	$^{\circ}\text{F}$		$\frac{\text{Btu}}{\text{Hr-Ft}^2-^{\circ}\text{F}}$	$\frac{\text{Ft}}{\text{Sec}}$	$\frac{10^6 \text{Lbm}}{\text{Hr-Ft}^2}$		Re	Pr	Nu	Ratio
For Freon on Copper												
SPCFC1-001	3817	136.82	122.15		260	1.958	0.6617	40302	6.928	467	1.094	
SPCFC1-002	1273	127.36	122.11		243	1.957	0.6617	40292	6.929	436	1.033	

TABLE D-18
REDUCED DATA FOR FORCED UPFLOW CONVECTION OF WATER AND FREON-113 - HEAT METER HM-2

Run Ident.	q'' (HM-2)	Temperature		h Coeff.	Vel.	G Flux	Re	Pr	Nu	Visc. Ratio	q'' (HM-1)
	Btu	Wall	Bulk	Btu	Ft	10 ³ Lbm					Btu
	Hr-Ft ²	°F	°F	Hr-Ft ² -°F	Sec	Hr-Ft ²					Hr-Ft ²
For Water on Nickel - Convection Normal with Heat Meter HM-1 OFF											
SPCWN2-001	24124	230.12	213.31	1434	2.007	0.4322	45982	1.719	260	1.100	0
SPCWN2-002	41999	243.77	213.75	1399	1.990	0.4284	45695	1.714	254	1.178	0
SPCWN2-003	32370	235.94	213.75	1459	1.999	0.4302	45895	1.714	265	1.132	0
SPCWN2-004	26566	231.75	213.75	1476	1.999	0.4302	45895	1.714	268	1.107	0
SPCWN2-005	40289	239.73	213.64	1544	1.987	0.4278	45601	1.715	280	1.155	0
SPCWN2-006	44708	242.62	213.61	1541	1.987	0.4278	45594	1.715	280	1.172	0
SPCWN2-007	51297	246.31	213.51	1564	1.976	0.4253	45304	1.717	284	1.195	0
SPCWN2-008	43921	243.25	214.96	1553	2.019	0.4344	46669	1.701	282	1.167	0
SPCWN2-009	46815	245.01	214.99	1560	2.019	0.4344	46677	1.701	283	1.177	0
SPCWN2-010	46560	244.75	215.07	1568	1.990	0.4282	46026	1.700	285	1.175	0
SPCWN2-011	42477	242.11	215.07	1571	1.996	0.4294	46163	1.700	285	1.159	0
SPCWN2-012	30121	234.34	214.94	1553	2.002	0.4307	46262	1.702	282	1.115	0
SPCWN2-013	18181	226.92	215.16	1547	2.005	0.4313	46387	1.699	281	1.070	0
SPCWN2-014	38279	239.63	215.11	1561	2.011	0.4325	46506	1.700	283	1.145	0
SPCWN2-015	42697	243.94	215.18	1485	1.988	0.4275	45988	1.699	269	1.169	0
SPCWN2-016	38248	240.61	214.96	1491	1.999	0.4301	46200	1.701	271	1.151	0
SPCWN2-017	25269	231.86	214.97	1496	2.008	0.4319	46403	1.701	271	1.100	0
SPCWN2-018	43873	245.54	214.83	1429	1.999	0.4301	46167	1.703	259	1.181	0
SPCWN2-019	43545	245.49	215.04	1430	2.002	0.4307	46288	1.700	260	1.179	0
SPCWN2-020	38104	241.65	215.05	1433	1.996	0.4294	46156	1.700	260	1.157	0
SPCWN2-021	40353	244.35	215.05	1377	1.999	0.4300	46221	1.700	250	1.173	0
SPCWN2-022	30196	237.20	215.08	1365	1.999	0.4300	46228	1.700	248	1.130	0
SPCWN2-023	20074	229.94	215.04	1348	1.999	0.4300	46220	1.700	245	1.088	0
SPCWN2-024	9752	222.83	215.09	1259	1.996	0.4294	46164	1.700	228	1.047	0

TABLE D-18 (Continued)

Run Ident.	q'' (HH-2)	Temperature		h Coeff.	Vel.	G Flux		Re	Pr	Nu	Visc. Ratio	q'' (HH-1)
	$\frac{\text{Btu}}{\text{Hr-Ft}^2}$	Wall $^{\circ}\text{F}$	Bulk $^{\circ}\text{F}$			$\frac{\text{Btu}}{\text{Hr-Ft}^2-\text{F}^{\circ}}$	$\frac{\text{Ft}}{\text{Sec}}$					$\frac{\text{Btu}}{\text{Hr-Ft}^2}$
SPCWN2-025	35209	240.52	214.92	1376	1.999	0.4301	46190	1.702	250	1.151		0
SPCWN2-026	15237	226.12	214.83	1350	1.999	0.4301	46166	1.703	245	1.067		0
SPCWN2-027	64015	261.13	214.98	1387	2.002	0.4307	46271	1.701	252	1.271		0
SPCWN2-028	56020	255.62	215.03	1380	1.999	0.4300	46217	1.701	250	1.239		0
SPCWN2-029	49836	251.45	215.02	1368	2.005	0.4313	46349	1.701	248	1.215		0
SPCWN2-030	9859	226.48	215.08	865	0.998	0.2147	23078	1.700	157	1.068		0
SPCWN2-031	19591	237.58	215.06	870	0.998	0.2147	23075	1.700	158	1.133		0
SPCWN2-032	29812	248.65	215.04	887	0.995	0.2141	23008	1.700	161	1.198		0
SPCWN2-033	33995	253.82	215.39	884	0.995	0.2140	23051	1.697	160	1.226		0
SPCWN2-034	9891	231.94	215.04	585	0.497	0.1069	11493	1.700	106	1.100		0
SPCWN2-035	19521	247.57	215.02	600	0.494	0.1063	11426	1.701	109	1.192		0
SPCWN2-036	24155	254.98	215.03	605	0.494	0.1063	11428	1.701	110	1.236		0
SPCWN2-037	29479	262.85	214.98	616	0.494	0.1063	11424	1.701	112	1.283		0
SPCWN2-038	14799	223.60	215.07	1736	2.987	0.6425	69073	1.700	315	1.051		0
SPCWN2-039	20126	226.67	215.12	1743	2.987	0.6425	69090	1.700	316	1.069		0
SPCWN2-040	30117	232.27	215.07	1751	3.001	0.6456	69401	1.700	318	1.102		0
SPCWN2-041	40152	237.59	215.06	1783	2.987	0.6425	69067	1.700	323	1.133		0
SPCWN2-042	39303	258.66	214.96	899	0.993	0.2147	23063	1.701	163	1.258		0
SPCWN2-043	29789	248.10	215.03	901	0.990	0.2147	23072	1.701	163	1.195		0

For Water on Nickel - Convection Enhanced by Heat Meter HH-1 ON

SPCWN2-044	34097	254.64	214.98	860	0.987	0.2122	22802	1.701	156	1.234		34554
SPCWN2-045	24552	244.11	215.00	843	0.992	0.2135	22936	1.701	153	1.172		29859
SPCWN2-046	14916	232.89	215.03	835	0.992	0.2135	22941	1.701	152	1.106		19958
SPCWN2-047	29108	265.44	214.94	576	0.494	0.1063	11421	1.702	105	1.299		24687
SPCWN2-048	19285	249.31	214.92	561	0.494	0.1063	11420	1.702	102	1.203		19851

TABLE E-18 (Continued)

Run Ident.	q'' (HW-2)	Temperature		h Coeff.		Vel. G Flux		Re	Pr	Nu	Visc. Ratio	q'' (HW-1)
	$\frac{\text{Btu}}{\text{hr-ft}^2}$	Wall °F	Bulk °F	$\frac{\text{Btu}}{\text{hr-ft}^2-\text{°F}}$	$\frac{\text{ft}}{\text{sec}}$	$\frac{10^6 \text{ Lbm}}{\text{hr-ft}^2}$						$\frac{\text{Btu}}{\text{hr-ft}^2}$
SPCWN2-049	14359	240.74	214.92	556	0.491	0.1057	11354	1.702	101	1.152		14665
SPCWN2-050	40045	238.13	215.12	1740	2.987	0.6425	69090	1.700	316	1.136		40189
SPCWN2-051	30160	232.43	215.06	1736	3.001	0.6456	69390	1.700	315	1.103		30278
SPCWN2-052	20167	226.62	214.98	1733	3.001	0.6456	69368	1.701	315	1.069		20323
SPCWN2-053	15206	223.78	214.98	1728	3.016	0.6487	69699	1.701	314	1.053		14950
For Water on Copper - Convection Normal with Heat Meter HW-1 OFF												
SPCWC2-001	24989	233.81	215.02	1330	2.008	0.4319	46414	1.701	241	1.111		0
SPCWC2-002	20022	230.16	215.00	1321	2.008	0.4319	46409	1.701	240	1.090		0
SPCWC2-003	15015	226.33	214.98	1322	2.008	0.4319	46403	1.701	240	1.068		0
SPCWC2-004	14829	226.50	215.02	1291	2.005	0.4313	46346	1.701	234	1.068		0
SPCWC2-005	24783	233.77	215.00	1320	2.005	0.4313	46342	1.701	240	1.111		0
SPCWC2-006	29808	237.52	214.99	1323	1.999	0.4300	46207	1.701	240	1.133		0
SPCWC2-007	39822	244.88	214.96	1331	2.005	0.4313	46331	1.701	242	1.176		0
SPCWC2-008	44990	248.64	214.97	1336	2.008	0.4319	46402	1.701	243	1.199		0
SPCWC2-009	25047	229.46	215.00	1733	3.001	0.6456	69376	1.701	314	1.086		0
SPCWC2-010	19995	226.72	215.01	1707	3.001	0.6456	69377	1.701	310	1.070		0
SPCWC2-011	54978	232.75	215.98	2085	3.979	0.8556	92457	1.691	378	1.099		0
SPCWC2-012	14745	232.55	215.00	840	1.001	0.2153	23134	1.701	152	1.104		0
SPCWC2-013	10042	226.97	214.94	835	1.001	0.2153	23126	1.702	151	1.072		0
For Water on Copper - Convection Enhanced by Heat Meter HW-1 ON												
SPCWC2-014	24984	229.62	215.00	1709	2.987	0.6425	69042	1.701	310	1.087		25102
SPCWC2-015	19894	227.07	215.04	1655	2.987	0.6425	69059	1.700	300	1.071		20178
SPCWC2-016	30408	230.54	214.99	1916	2.005	0.4313	46341	1.701	348	1.092		75920

TABLE N-18 (Continued)

Run Ident.	q'' (HM-2)	Temperature		h Coeff.	Vel.	G Flux	Re	Pr	Nu	Visc. Ratio	q'' (HM-1)
	$\frac{Btu}{Hr-Ft^2}$	Wall $^{\circ}F$	Bulk $^{\circ}F$		$\frac{Ft}{Sec}$	$\frac{10^6 Lbm}{Hr-Ft^2}$					$\frac{Btu}{Hr-Ft^2}$
SPCWC2-017	30351	231.35	214.93	1813	2.002	0.4307	46260	1.702	329	1.097	50548
SPCWC2-018	22379	226.71	214.83	1813	2.005	0.4313	46300	1.703	323	1.071	75976
SPCWC2-019	20094	225.59	215.00	1840	1.999	0.4300	46208	1.701	314	1.053	76013
SPCWC2-020	15033	223.20	214.83	1729	1.999	0.4300	46164	1.703	314	1.050	76167
SPCWC2-021	14989	223.45	215.11	1730	2.005	0.4313	46371	1.700	314	1.050	75533
SPCWC2-022	14911	223.58	215.11	1696	2.008	0.4319	46437	1.700	302	1.051	49385
SPCWC2-023	19804	226.04	215.02	1746	2.005	0.4313	46348	1.701	317	1.066	49377
SPCWC2-024	24867	228.76	215.05	1772	2.005	0.4313	46355	1.700	322	1.081	49308

For Freon on Copper - Convection Normal with Heat Meter HM-1 OFF

SPCFC2-001	981	126.64	121.90	207	1.956	0.6619	40108	6.951	371	1.034	0
SPCFC2-002	628	125.07	121.87	196	1.954	0.6609	40042	6.952	352	1.024	0
SPCFC2-003	4800	142.20	122.08	239	1.951	0.6598	40028	6.946	428	1.134	0
SPCFC2-004	2437	132.92	121.98	223	1.953	0.6607	40059	6.949	400	1.074	0
SPCFC2-005	1017	128.47	121.90	155	0.969	0.3279	19867	6.951	278	1.045	0
SPCFC2-006	1098	127.75	123.84	281	2.946	0.9948	61038	6.895	506	1.028	0
SPCFC2-007	3084	132.59	124.38	375	3.928	1.3258	81625	6.880	677	1.055	0
SPCFC2-008	2053	130.14	124.31	352	3.900	1.3164	81012	6.882	634	1.040	0

For Freon on Copper - Convection Enhanced by Heat Meter HM-1 ON

SPCFC2-009	4932	142.48	122.15	243	1.957	0.6617	40163	6.944	435	1.136	3817
SPCFC2-010	2597	132.80	122.11	243	1.956	0.6617	40153	6.945	436	1.072	1273
SPCFC2-011	2552	129.25	121.94	324	1.953	0.6608	40054	6.950	582	1.050	24842
SPCFC2-012	2522	129.57	121.90	307	1.956	0.6618	40102	6.951	550	1.053	13815
SPCFC2-013	1221	125.37	121.97	309	1.953	0.6607	40056	6.949	555	1.025	24356

TABLE D-18 (Continued)

Run Ident.	q'' (HM-2)	Temperature		h Coeff.	Vel.	G Flux		Re	Pr	Nu	Visc. Ratio	q'' (HM-1)
	$\frac{\text{Btu}}{\text{Hr-Ft}^2}$	Wall °F	Bulk °F			$\frac{\text{Btu}}{\text{Hr-Ft}^2-\text{°F}}$	$\frac{\text{Pt}}{\text{Sec}}$					$\frac{\text{Btu}}{\text{Hr-Ft}^2}$
SPCPC2-014	1125	125.34	122.05	292	1.957	0.6618	40140	6.947	525	1.024		24316
SPCPC2-015	1121	125.56	122.00	272	1.959	0.6628	40187	6.948	489	1.026		14540
SPCPC2-016	2107	128.77	121.92	285	0.969	0.3278	19869	6.950	511	1.047		14471
SPCPC2-017	1083	125.88	121.92	240	0.969	0.3279	19870	6.950	430	1.029		14434

TABLE D-19

REDUCED DATA FOR FLOW BOILING
ON HEAT METER HM-1

Run Ident.		q'' (HM-1)	Temperatures			h Coef	Vel	G Flux	Re	Pr	Nu
		$\frac{\text{Btu}}{\text{Hr-Ft}^2}$	Wall $^{\circ}\text{F}$	Bulk $^{\circ}\text{F}$	Diff $^{\circ}\text{F}$	$\frac{\text{Btu}}{\text{Hr-Ft}^2-^{\circ}\text{F}}$	$\frac{\text{Ft}}{\text{Sec}}$	$\frac{10^6 \text{ Lbm}}{\text{Hr-Ft}^2}$			
Freon-113 with Essentially Zero Thermodynamic Quality											
TPBFC1-	1	14517	138.85	122.06	16.78	865	0.972	0.329	20012	6.93	1554
TPBFC1-	2	14543	138.77	122.06	16.70	871	0.970	0.328	19954	6.93	1564
TPBFC1-	3	14383	138.73	121.92	16.81	856	0.970	0.328	19938	6.93	1537
TPBFC1-	4	14515	138.64	121.89	16.76	866	0.970	0.328	19935	6.94	1556
TPBFC1-	5	14471	138.61	121.92	16.69	867	0.970	0.328	19939	6.93	1557
TPBFC1-	6	14434	138.64	121.92	16.72	863	0.970	0.328	19939	6.93	1551
TPBFC1-	7	25271	140.06	122.04	18.02	1402	1.960	0.663	40335	6.93	2519
TPBFC1-	8	14591	138.26	122.02	16.24	899	1.957	0.662	40273	6.93	1614
TPBFC1-	9	22720	139.70	122.00	17.70	1284	1.960	0.663	40327	6.93	2306
TPBFC1-	10	14673	138.21	122.02	16.19	906	1.960	0.663	40330	6.93	1628
TPBFC1-	11	25356	140.00	121.95	18.05	1405	1.966	0.665	40430	6.93	2523
TPBFC1-	12	14623	138.24	122.01	16.22	901	1.966	0.665	40445	6.93	1619
TPBFC1-	13	24857	140.39	122.01	18.38	1352	1.963	0.664	40387	6.93	2429
TPBFC1-	14	14516	138.45	121.98	16.47	881	1.960	0.663	40321	6.93	1583
TPBFC1-	15	14384	138.47	121.98	16.50	872	1.960	0.663	40322	6.93	1566
TPBFC1-	16	25016	140.15	121.87	18.28	1369	1.957	0.662	40238	6.94	2458
TPBFC1-	17	24842	140.18	121.94	18.24	1362	1.954	0.661	40194	6.93	2446
TPBFC1-	18	13815	138.37	121.90	16.47	839	1.957	0.662	40242	6.93	1506
TPBFC1-	19	24356	140.11	121.97	18.14	1342	1.954	0.661	40196	6.93	2411
TPBFC1-	20	24316	140.42	122.05	18.38	1323	1.958	0.662	40280	6.93	2377
TPBFC1-	21	14540	138.68	122.00	16.68	872	1.960	0.663	40328	6.93	1566
TPBFC1-	22	25250	142.38	124.05	18.34	1377	2.947	0.995	61303	6.88	2482

TABLE D-19 (Continued)

Run Ident.		q'' (HH-1)	Temperatures			h Coef	Vel	G Flux	Re	Pr	Nu
		$\frac{\text{Btu}}{\text{Hr-Ft}^2}$	Wall $^{\circ}\text{F}$	Bulk $^{\circ}\text{F}$	Diff $^{\circ}\text{F}$	$\frac{\text{Btu}}{\text{Hr-Ft}^2-^{\circ}\text{F}}$	$\frac{\text{Ft}}{\text{Sec}}$	$\frac{10^6 \text{ Lbm}}{\text{Hr-Ft}^2}$			
TPBFC1-	23	25275	142.41	124.09	18.32	1380	2.947	0.995	61316	6.87	2487
TPBFC1-	24	31990	143.01	123.83	19.18	1668	2.947	0.995	61233	6.88	3005
Water with Essentially Zero Thermodynamic Quality											
TPBWC1-	1	50209	231.17	215.00	16.17	3105	0.491	0.106	11380	1.70	563
TPBWC1-	2	34984	230.98	214.97	16.01	2185	0.489	0.105	11312	1.70	396
TPBWC1-	3	65228	232.23	214.97	17.26	3778	0.489	0.105	11312	1.70	686
TPBWC1-	4	75764	234.45	215.15	19.30	3926	2.005	0.431	46466	1.70	712
TPBWC1-	5	50362	233.35	215.15	18.20	2767	2.005	0.431	46466	1.70	502
TPBWC1-	6	75900	234.86	214.94	19.92	3810	2.008	0.432	46480	1.70	691
TPBWC1-	7	49879	234.10	214.99	19.11	2610	2.008	0.432	46494	1.70	474
TPBWC1-	8	75732	235.37	214.87	20.50	3694	2.008	0.432	46464	1.70	670
TPBWC1-	9	49785	234.41	215.00	19.41	2564	2.002	0.431	46363	1.70	465
TPBWC1-	10	75859	235.21	214.98	20.23	3749	2.004	0.432	46492	1.70	680
TPBWC1-	11	50298	234.70	214.94	19.76	2546	2.017	0.434	46684	1.70	462
TPBWC1-	12	75920	235.11	214.99	20.11	3774	2.005	0.431	46428	1.70	685
TPBWC1-	13	50548	234.57	214.93	19.63	2575	2.002	0.431	46347	1.70	467
TPBWC1-	14	75976	235.24	214.83	20.42	3721	2.005	0.431	46387	1.70	675
TPBWC1-	15	76013	235.28	215.00	20.28	3748	1.999	0.430	46295	1.70	680
TPBWC1-	16	76167	235.25	214.83	20.42	3730	1.999	0.430	46251	1.70	677
TPBWC1-	17	75533	239.12	215.11	24.00	3147	2.005	0.431	46457	1.70	571
TPBWC1-	18	49385	237.64	215.11	22.53	2192	2.008	0.432	46524	1.70	394
TPBWC1-	19	49377	237.47	215.02	22.45	2200	2.005	0.431	46435	1.70	399
TPBWC1-	20	49308	237.61	215.05	22.56	2186	2.005	0.431	46441	1.70	397
TPBWC1-	21	75861	228.32	215.00	13.32	5695	1.999	0.430	46296	1.70	1033
TPBWC1-	22	50388	228.20	214.96	13.24	3805	1.999	0.430	46285	1.70	690

TABLE D-20

REDUCED DATA FOR FLOW BOILING
ON HEAT METR H4-2

Run Ident.		q'' (HM-2)	Temperatures			h Coef	Vel	G Flux	Re	Pr	Nu	q'' (HM-1)
		Btu	Wall	Bulk	Diff	Rtu	Ft	10^4 Lbm				Btu
		Hr-Ft ²	°F	°F	°F	Hr-Ft ² -°F	Sec	Hr-Ft ²				Hr-Ft ²
Freon-113 with Essentially Zero Thermodynamic Quality												
TPBFC2-	1	15282	139.13	121.47	17.66	865	0.972	0.329	19937	6.95	1553	(0)
TPBFC2-	2	10209	137.84	121.49	16.40	622	0.969	0.328	19882	6.95	1117	(0)
TPBFC2-	3	5087	135.25	121.45	13.80	369	0.969	0.328	19878	6.95	662	(0)
TPBFC2-	4	3000	133.60	121.30	12.30	244	0.969	0.328	19862	6.95	438	(0)
TPBFC2-	5	2028	132.35	121.40	10.95	185	0.969	0.328	19873	6.95	332	(0)
TPBFC2-	6	25961	140.60	121.45	19.16	1355	1.959	0.663	40187	6.95	2432	(0)
TPBFC2-	7	20905	139.88	121.48	18.40	1136	1.959	0.663	40195	6.95	2039	(0)
TPBFC2-	8	13544	138.40	121.46	16.94	799	1.957	0.662	40132	6.95	1435	(0)
TPBFC2-	9	8461	136.63	121.34	15.29	553	1.962	0.664	40221	6.95	993	(0)
TPBFC2-	10	5074	134.56	121.37	13.19	385	1.959	0.663	40170	6.95	690	(0)
TPBFC2-	11	4097	133.58	121.37	12.21	336	1.959	0.663	40172	6.95	602	(0)
TPBFC2-	12	2972	132.65	121.38	11.27	264	1.959	0.663	40172	6.95	473	(0)
TPBFC2-	13	1959	130.41	121.38	9.04	217	1.959	0.663	40172	6.95	389	(0)
TPBFC2-	14	24964	140.44	121.53	18.91	1320	1.962	0.664	40263	6.95	2370	(0)
TPBFC2-	15	25121	140.50	121.42	19.07	1317	1.956	0.662	40124	6.95	2364	(0)
TPBFC2-	16	17877	139.39	121.34	18.05	990	1.956	0.662	40104	6.95	1777	(0)
TPBFC2-	17	11019	137.82	121.45	16.38	673	1.956	0.662	40128	6.95	1208	(0)
TPBFC2-	18	7564	136.39	121.49	14.90	508	1.962	0.664	40253	6.95	911	(0)
TPBFC2-	19	29837	143.32	123.55	19.76	1510	2.932	0.990	60821	6.89	2719	(0)
TPBFC2-	20	20297	141.99	123.61	18.39	1104	2.946	0.995	61129	6.89	1988	(0)
TPBFC2-	21	12859	140.49	123.58	16.92	760	2.932	0.990	60829	6.89	1369	(0)
TPBFC2-	22	7790	138.33	123.58	14.75	528	2.946	0.995	61122	6.89	951	(0)

TABLE D-20 (Continued)

Run Ident.	q'' (HH-2)	Temperatures			h Coef	Vel	G Flux	Re	Pr	Nu	q'' (HH-1)
	Btu Hr-Ft ²	Wall °F	Bulk °F	Diff °F	Btu Hr-Ft ² -°F	Ft Sec	10 ⁶ Lbm Hr-Ft ²				Btu Hr-Ft ²
TPBFC2- 23	5066	136.47	123.53	12.94	381	2.946	0.995	61105	6.89	705	(0)
TPBFC2- 24	3074	133.75	123.51	10.24	300	2.946	0.995	61100	6.89	540	(0)
TPBFC2- 25	34764	144.43	124.00	20.43	1702	3.900	1.316	81095	6.88	3067	(0)
TPBFC2- 26	20355	142.46	123.93	18.54	1098	3.914	1.321	81356	6.88	1979	(0)
TPBFC2- 27	12831	140.90	123.93	16.97	756	3.900	1.316	81068	6.88	1362	(0)
TPBFC2- 28	6347	137.35	123.65	13.70	463	3.914	1.321	81224	6.89	834	(0)
TPBFC2- 29	4048	134.85	123.83	11.02	367	3.928	1.326	81605	6.88	662	(0)
TPBFC2- 30	15291	139.26	121.51	17.75	862	0.972	0.329	19942	6.95	1546	(14517)
TPBFC2- 31	10246	137.80	121.51	16.28	629	0.969	0.328	19885	6.95	1129	(14543)
TPBFC2- 32	5065	134.44	121.37	13.07	387	0.969	0.328	19869	6.95	695	(14383)
TPBFC2- 33	3134	131.53	121.33	10.20	307	0.969	0.328	19866	6.95	551	(14515)
TPBFC2- 34	25111	140.55	121.49	19.07	1317	1.959	0.663	40195	6.95	2364	(25271)
TPBFC2- 35	25250	140.56	121.47	19.09	1323	1.956	0.662	40133	6.95	2374	(14591)
TPBFC2- 36	17801	139.60	121.45	18.15	981	1.959	0.663	40187	6.95	1760	(22720)
TPBFC2- 37	17889	139.46	121.47	17.99	994	1.959	0.663	40190	6.95	1785	(14673)
TPBFC2- 38	12750	138.39	121.39	16.99	750	1.965	0.665	40289	6.95	1346	(25356)
TPBFC2- 39	12756	138.49	121.46	17.03	749	1.965	0.665	40305	6.95	1344	(14623)
TPBFC2- 40	7690	135.98	121.46	14.52	530	1.962	0.664	40247	6.95	951	(24857)
TPBFC2- 41	7692	136.18	121.43	14.75	522	1.959	0.663	40181	6.95	936	(14516)
TPBFC2- 42	5076	133.97	121.42	12.55	404	1.959	0.663	40182	6.95	726	(14384)
TPBFC2- 43	5092	133.71	121.31	12.40	411	1.956	0.662	40098	6.95	737	(25016)
TPBFC2- 44	30077	143.32	123.53	19.78	1520	2.946	0.995	61105	6.89	2738	(25250)
TPBFC2- 45	20275	142.05	123.58	18.48	1097	2.946	0.995	61119	6.89	1976	(25275)
TPBFC2- 46	30260	143.67	123.31	20.36	1486	2.946	0.995	61035	6.89	2676	(31990)

TABLE N-20 (Continued)

Run Ident.		q'' (HM-2)	Temperatures			h Coef	Vel	G Flux	Re	Pr	Nu	q'' (HM-1)	
		Btu	Wall	Bulk	Diff	Btu	Ft	10 ⁴ Lbm				Btu	
		Hr-Ft ²	°F	°F	°F	Hr-Ft ² -°F	Sec	Hr-Ft ²				Hr-Ft ²	
Water with Essentially Zero Thermodynamic Quality													
TPBWC2-	1	50411	233.78	214.71	19.07	2644	0.494	0.106	11426	1.70	480	(0)	
TPBWC2-	2	34905	232.99	214.63	18.35	1902	0.491	0.106	11356	1.70	345	(0)	
TPBWC2-	3	61153	232.12	214.72	17.40	3514	0.998	0.215	23075	1.70	638	(0)	
TPBWC2-	4	51027	231.94	214.73	17.20	2966	1.001	0.215	23143	1.70	538	(0)	
TPBWC2-	5	41135	231.79	214.73	17.06	2411	0.998	0.215	23077	1.70	437	(0)	
TPBWC2-	6	30934	231.38	214.73	16.64	1859	0.998	0.215	23077	1.70	337	(0)	
TPBWC2-	7	25871	230.97	214.72	16.25	1592	0.998	0.215	23076	1.70	289	(0)	
TPBWC2-	8	18306	229.92	214.70	15.22	1203	0.994	0.215	23075	1.70	218	(0)	
TPBWC2-	9	71577	237.57	214.75	22.82	3136	2.005	0.431	46360	1.70	569	(0)	
TPBWC2-	10	61211	237.11	214.74	22.38	2735	2.008	0.432	46424	1.70	496	(0)	
TPBWC2-	11	51434	236.83	214.72	22.11	2326	2.005	0.431	46355	1.70	422	(0)	
TPBWC2-	12	40531	236.23	214.70	21.53	1883	2.005	0.431	46349	1.70	342	(0)	
TPBWC2-	13	30486	235.20	214.68	20.52	1485	2.005	0.431	46343	1.70	270	(0)	
TPBWC2-	14	70615	235.69	214.76	20.92	3375	2.005	0.431	46363	1.70	612	(0)	
TPBWC2-	15	70698	236.54	214.84	21.71	3257	2.008	0.432	46450	1.70	591	(0)	
TPBWC2-	16	61113	236.22	214.86	21.36	2861	2.008	0.432	46455	1.70	519	(0)	
TPBWC2-	17	50451	236.18	214.89	21.69	2326	2.008	0.432	46359	1.70	422	(0)	
TPBWC2-	18	40470	235.53	214.75	20.77	1946	2.008	0.432	46429	1.70	354	(0)	
TPBWC2-	19	30422	234.65	214.74	19.91	1528	2.005	0.431	46359	1.70	277	(0)	
TPBWC2-	20	70486	242.55	214.78	27.78	2538	1.999	0.430	46235	1.70	460	(0)	
TPBWC2-	21	64445	232.18	214.68	17.51	3710	2.008	0.432	46409	1.70	673	(0)	
TPBWC2-	22	35240	231.23	214.69	16.54	2131	1.999	0.430	46213	1.70	387	(0)	
TPBWC2-	23	34506	231.80	214.59	17.22	2004	1.999	0.430	46187	1.70	364	(0)	
TPBWC2-	24	64748	238.05	214.77	23.27	2702	2.005	0.431	46367	1.70	505	(0)	
TPBWC2-	25	55418	233.24	214.64	18.59	3518	3.001	0.646	69364	1.70	638	(0)	

TABLE n-20 (Continued)

Run Ident.	q'' (RH-2)	Temperatures			h Coef	Vel	G Flux	Re	Pr	Nu	q'' (RH-1)
	$\frac{\text{Btu}}{\text{Hr-Ft}^2}$	Wall $\circ\text{F}$	Bulk $\circ\text{F}$	Diff $\circ\text{F}$		$\frac{\text{Ft}}{\text{Sec}}$	$\frac{10^6 \text{ Lbm}}{\text{Hr-Ft}^2}$				$\frac{\text{Btu}}{\text{Hr-Ft}^2}$
TPBWC2- 26	56135	233.23	214.68	18.55	3026	3.001	0.646	69376	1.70	549	(0)
TPBWC2- 27	35106	231.99	214.68	17.32	2027	3.001	0.646	69376	1.70	368	(0)
TPBWC2- 28	45536	232.78	214.70	18.08	2519	3.001	0.646	69385	1.70	457	(0)
TPBWC2- 29	30663	231.31	214.68	16.64	1843	3.016	0.649	69709	1.70	334	(0)
TPBWC2- 30	64540	239.25	215.73	23.52	2745	3.965	0.852	92161	1.69	498	(0)
TPBWC2- 31	34767	233.33	214.68	18.65	1864	0.491	0.106	11359	1.70	338	(50209)
TPBWC2- 32	34880	233.44	214.64	18.80	1856	0.489	0.105	11291	1.70	337	(34984)
TPBWC2- 33	34832	233.30	214.64	18.66	1867	0.489	0.105	11291	1.70	339	(65228)
TPBWC2- 34	70804	236.77	214.83	21.94	3227	2.005	0.431	46380	1.70	585	(75764)
TPBWC2- 35	70774	236.83	214.83	22.00	3216	2.005	0.431	46379	1.70	584	(50362)
TPBWC2- 36	60768	236.58	214.61	21.97	2766	2.008	0.432	46392	1.70	502	(75900)
TPBWC2- 37	60847	236.76	214.67	22.09	2754	2.008	0.432	46407	1.70	500	(49879)
TPBWC2- 38	50967	236.78	214.55	22.23	2293	2.008	0.432	46377	1.70	416	(75732)
TPBWC2- 39	50825	236.47	214.68	21.79	2332	2.002	0.431	46276	1.70	423	(49785)
TPBWC2- 40	40347	234.88	214.65	20.23	1994	2.008	0.432	46405	1.70	362	(75859)
TPBWC2- 41	40367	234.99	214.62	20.37	1982	2.016	0.434	46597	1.70	360	(50298)
TPBWC2- 42	34903	231.40	214.68	16.73	2087	1.999	0.430	46209	1.70	379	(75861)
TPBWC2- 43	34973	231.69	214.63	17.05	2051	1.999	0.430	46198	1.70	372	(50388)

TABLE D-21

REDUCED DATA FOR ANNULAR FLOW BOILING OF FREON-113
ON HEAT METER HM-2

Run Ident.		q'' (HM-2)	Temperatures			h Coef	G Flux	Re	Pr	Nu	Thermo. Quality %
		Btu Hr-Ft ²	Wall °F	Bulk °F	Diff °F	Btu Hr-Ft ² -°F	10 ⁴ Lbm Hr-Ft ²				
TPXFC2-	1	25058	141.32	122.33	18.99	1319	0.326	19890	6.92	2371	1.99
TPXFC2-	2	17862	140.19	122.32	17.86	1000	0.326	19889	6.92	1797	2.00
TPXFC2-	3	10219	138.15	122.25	15.90	643	0.326	19880	6.92	1155	2.02
TPXFC2-	4	5194	134.07	122.26	11.81	440	0.326	19882	6.92	790	1.99
TPXFC2-	5	25266	141.38	122.33	19.05	1327	0.325	19799	6.92	2384	3.91
TPXFC2-	6	20254	140.58	122.27	18.32	1106	0.325	19793	6.92	1987	3.86
TPXFC2-	7	15334	139.53	122.24	17.30	886	0.325	19789	6.93	1593	3.88
TPXFC2-	8	7796	136.18	122.13	14.06	555	0.325	19780	6.93	996	3.83
TPXFC2-	9	22760	140.72	121.98	18.74	1215	0.323	19655	6.93	2181	6.08
TPXFC2-	10	12922	138.39	121.96	16.43	787	0.325	19767	6.93	1413	6.09
TPXFC2-	11	7790	135.77	122.30	13.47	579	0.325	19815	6.92	1040	5.84
TPXFC2-	12	22751	140.91	122.21	18.69	1217	0.321	19582	6.93	2187	8.13
TPXFC2-	13	15470	139.40	122.27	17.13	903	0.321	19588	6.92	1623	8.12
TPXFC2-	14	7742	135.45	122.22	13.23	585	0.322	19637	6.93	1051	8.16
TPXFC2-	16	25225	141.60	121.75	19.85	1271	0.656	39872	6.94	2282	1.86
TPXFC2-	17	20245	140.68	121.79	18.88	1072	0.656	39884	6.94	1925	1.84
TPXFC2-	18	15363	139.46	121.70	17.76	865	0.656	39861	6.94	1553	1.88
TPXFC2-	19	10265	137.35	121.57	15.78	650	0.657	39891	6.94	1168	1.85
TPXFC2-	20	5177	132.02	121.70	10.32	502	0.660	40093	6.94	901	1.85
TPXFC2-	21	2148	126.35	121.56	4.79	448	0.660	40060	6.94	804	1.88
TPXFC2-	22	25089	142.65	122.11	20.54	1221	0.655	39868	6.93	2194	4.00
TPXFC2-	23	17904	141.13	122.06	19.06	939	0.656	39916	6.93	1687	4.00
TPXFC2-	24	10379	137.93	122.13	15.80	657	0.655	39874	6.93	1180	3.99
TPXFC2-	25	29908	144.03	124.33	19.69	1519	0.990	61145	6.87	2738	1.75

TABLE D-21 (Continued)

Run Ident.	q'' (HM-2) Btu Hr-Ft ²	Temperatures			h Coef Btu Hr-Ft ² -°F	G Flux 10 ⁶ Lbm Hr-Ft ²	Re	Pr	Nu	Thermo. Quality %
		Wall °F	Bulk °F	Diff °F						
TPXFC2- 26	22639	142.94	124.12	18.82	1203	0.990	61062	6.87	2168	1.83
TPXFC2- 27	15406	141.30	124.10	17.20	896	0.995	61349	6.87	1614	1.81
TPXFC2- 28	10380	139.09	124.03	15.06	689	0.990	61038	6.88	1242	1.80
TPXFC2- 29	8421	131.62	124.06	7.56	585	0.990	61052	6.88	1054	1.78
TPXFC2- 30	27916	145.95	126.34	19.61	1423	0.987	61725	6.81	2575	2.29
TPXFC2- 31	20503	144.66	126.39	18.27	1122	0.987	61749	6.81	2030	2.25
TPXFC2- 32	12845	142.85	126.41	16.45	781	0.987	61738	6.81	1413	2.36
TPXFC2- 33	6206	136.78	126.39	10.38	598	0.987	61731	6.81	1081	2.37
TPXFC2- 34	34552	147.96	128.24	19.72	1752	1.141	72176	6.76	3180	2.10
TPXFC2- 35	25062	146.81	128.41	18.41	1362	1.141	72248	6.76	2472	2.05
TPXFC2- 36	17652	145.59	128.33	17.26	1023	1.141	72216	6.76	1856	2.06
TPXFC2- 37	10202	142.54	128.33	14.21	718	1.141	72218	6.76	1303	2.06
TPXFC2- 38	7668	140.76	128.55	12.21	628	1.141	72290	6.75	1140	2.11
TPXFC2- 39	5051	137.32	128.56	8.77	576	1.145	72595	6.75	1046	2.10

TABLE D-22

ROHSENOW ADDITIVE CORRELATION DATA FOR FLOW BOILING
ON HEAT METER HM-1

Run Ident.	q^{Tot} Btu/(Hr-Ft ²)	q^{NB} Btu/(Hr-Ft ²)	q^{PC} Btu/(Hr-Ft ²)	Temp. diff. °F	h_{Meas} Btu/(Hr-Ft ² -°F)	h_{NB} Btu/(Hr-Ft ² -°F)	h_{PC} Btu/(Hr-Ft ² -°F)	$\frac{h_{Meas}}{h_{PC}}$	G Flux 10 ⁴ Lbm Hr-Ft ²	Reciprocal Martinelli Parameter
Freon-113 with Essentially Zero Thermodynamic Quality										
TPBFC1- 1	14517	12150	2367	16.78	865	724	141	6.1	0.329	0.000
TPBFC1- 2	14543	12192	2351	16.70	871	730	141	6.2	0.328	0.000
TPBFC1- 3	14383	12017	2366	16.81	856	715	141	6.1	0.328	0.000
TPBFC1- 4	14515	12157	2358	16.76	866	726	141	6.2	0.328	0.000
TPBFC1- 5	14371	12122	2349	16.69	867	726	141	6.2	0.328	0.000
TPBFC1- 6	14434	12081	2353	16.72	863	723	141	6.1	0.328	0.000
TPBFC1- 7	25271	21342	3929	18.02	1402	1184	218	6.4	0.663	0.000
TPBFC1- 8	14591	11055	3536	16.24	899	681	218	4.1	0.662	0.000
TPBFC1- 9	22720	18862	3858	17.70	1284	1066	218	5.9	0.663	0.000
TPBFC1- 10	14673	11144	3529	16.19	906	688	218	4.2	0.663	0.000
TPBFC1- 11	25356	21415	3941	18.05	1405	1186	218	6.4	0.665	0.000
TPBFC1- 12	14623	11080	3543	16.22	901	683	218	4.1	0.665	0.000
TPBFC1- 13	24857	20847	4010	18.38	1352	1134	218	6.2	0.664	0.000
TPBFC1- 14	14516	10927	3589	16.47	881	664	218	4.0	0.663	0.000
TPBFC1- 15	14384	10798	3596	16.50	872	654	218	4.0	0.663	0.000
TPBFC1- 16	25016	21036	3980	18.28	1369	1151	218	6.3	0.662	0.000
TPBFC1- 17	24842	20873	3969	18.24	1362	1144	218	6.3	0.661	0.000
TPBFC1- 18	13815	10229	3586	16.47	839	621	218	3.9	0.662	0.000
TPBFC1- 19	24356	20409	3947	18.14	1342	1125	218	6.2	0.661	0.000
TPBFC1- 20	24316	20314	4002	18.38	1323	1105	218	6.1	0.662	0.000
TPBFC1- 21	14540	10904	3636	16.68	872	654	218	4.0	0.663	0.000
TPBFC1- 22	25250	21098	5152	18.34	1377	1096	281	4.9	0.995	0.000

TABLE D-22 (Continued)

Run Ident.	$\frac{q^{Tot}}{Btu/(Hr-Ft^2)}$	$\frac{q^{WB}}{Btu/(Hr-Ft^2)}$	$\frac{q^{FC}}{Btu/(Hr-Ft^2)}$	Temp. Diff. °F	$\frac{h_{Meas}}{Btu/(Hr-Ft^2-°F)}$	$\frac{h_{WB}}{Btu/(Hr-Ft^2-°F)}$	$\frac{h_{FC}}{Btu/(Hr-Ft^2-°F)}$	$\frac{h_{Meas}}{h_{FC}}$	G Flux $\frac{10^6 Lbm}{Hr-Ft^2}$	Reciprocal Martinelli Parameter
TPBFC1- 23	25275	20128	5147	18.32	1380	1099	281	4.9	0.995	0.000
TPBFC1- 24	31990	26601	5389	19.18	1668	1387	281	5.9	0.995	0.000
Water with Essentially Zero Thermodynamic Quality										
TPBWC1- 1	50209	40491	9718	16.17	3105	2504	601	5.2	0.106	0.000
TPBWC1- 2	34984	25396	9588	16.01	2185	1586	599	3.6	0.105	0.000
TPBWC1- 3	65228	54892	10336	17.26	3778	3180	599	6.3	0.105	0.000
TPBWC1- 4	75764	47980	27784	19.30	3926	2486	1440	2.7	0.431	0.000
TPBWC1- 5	50362	24155	26207	18.20	2767	1327	1440	1.9	0.431	0.000
TPBWC1- 6	75900	47204	28696	19.92	3810	2370	1440	2.6	0.432	0.000
TPBWC1- 7	49879	22353	27526	19.11	2610	1170	1441	1.8	0.432	0.000
TPBWC1- 8	75732	46204	29528	20.50	3694	2254	1440	2.6	0.432	0.000
TPBWC1- 9	49785	21868	27917	19.41	2564	1126	1438	1.8	0.431	0.000
TPBWC1- 10	75859	46709	29150	20.23	3749	2308	1441	2.6	0.432	0.000
TPBWC1- 11	50298	21761	28537	19.76	2546	1101	1444	1.8	0.434	0.000
TPBWC1- 12	75920	46969	28951	20.11	3774	2335	1439	2.6	0.431	0.000
TPBWC1- 13	50548	22318	28230	19.63	2575	1137	1438	1.8	0.431	0.000
TPBWC1- 14	75976	46596	29380	20.42	3721	2282	1439	2.6	0.431	0.000
TPBWC1- 15	76013	46875	29138	20.28	3748	2311	1437	2.6	0.430	0.000
TPBWC1- 16	76167	46838	29329	20.42	3730	2294	1436	2.6	0.430	0.000
TPBWC1- 17	75533	40979	34554	24.00	3147	1707	1440	2.2	0.431	0.000
TPBWC1- 18	49385	16923	32462	22.53	2192	751	1441	1.5	0.432	0.000
TPBWC1- 19	49377	17069	32308	22.45	2200	760	1439	1.5	0.431	0.000
TPBWC1- 20	49308	16836	32472	22.56	2186	746	1439	1.5	0.431	0.000
TPBWC1- 21	75861	56721	19140	13.32	5695	4258	1437	4.0	0.430	0.000
TPBWC1- 22	50388	31363	19025	13.24	3805	2368	1437	2.6	0.430	0.000

TABLE D-23

ROHSENOW ADDITIVE CORRELATION DATA FOR FLOW BOILING
ON HEAT METER RM-2

Run Ident.	q^{Tot} Btu/(Hr-Ft ²)	q^{NB} Btu/(Hr-Ft ²)	q^{FC} Btu/(Hr-Ft ²)	Temp. Diff. °F	h_{Meas} Btu/(Hr-Ft ² -°F)	h_{NB} Btu/(Hr-Ft ² -°F)	h_{FC} Btu/(Hr-Ft ² -°F)	h_{Meas} h _{FC}	G Flux 10 ⁴ Lbm Hr-Ft ²	Reciprocal Martinelli Parameter
Freon-113 with Essentially Zero Thermodynamic Quality										
TPBFC2- 1	15282	12793	2489	17.66	865	724	141	6.1	0.329	0.000
TPBFC2- 2	10209	7901	2308	16.40	622	482	141	4.4	0.328	0.000
TPBFC2- 3	5087	3145	1942	13.80	369	228	141	2.6	0.328	0.000
TPBFC2- 4	3000	1269	1731	12.30	244	103	141	1.7	0.328	0.000
TPBFC2- 5	2028	488	1540	10.95	185	45	141	1.3	0.328	0.000
TPBFC2- 6	25961	21787	4174	19.16	1355	1137	218	6.2	0.663	0.000
TPBFC2- 7	20905	16897	4008	18.40	1136	918	218	5.2	0.663	0.000
TPBFC2- 8	13544	9856	3688	16.94	799	582	218	3.7	0.662	0.000
TPBFC2- 9	9461	5126	3335	15.29	553	335	218	2.5	0.664	0.000
TPBFC2- 10	5074	2201	2873	13.19	385	167	218	1.8	0.663	0.000
TPBFC2- 11	4097	1437	2660	12.21	336	118	218	1.5	0.663	0.000
TPBFC2- 12	2972	516	2456	11.27	264	46	218	1.2	0.663	0.000
TPBFC2- 13	1959	90	1968	9.04	217	10	218	1.0	0.663	0.000
TPBFC2- 14	24964	20841	4123	18.91	1320	1102	218	6.1	0.664	0.000
TPBFC2- 15	25121	20970	4151	19.07	1317	1099	218	6.1	0.662	0.000
TPBFC2- 16	17877	13948	3929	18.05	990	773	218	4.6	0.662	0.000
TPBFC2- 17	11019	7455	3564	16.38	673	455	218	3.1	0.662	0.000
TPBFC2- 18	7564	4315	3249	14.90	508	290	218	2.3	0.664	0.000
TPBFC2- 19	29837	24303	5534	19.76	1510	1230	280	5.4	0.990	0.000
TPBFC2- 20	20297	15133	5164	18.39	1104	823	281	3.9	0.995	0.000
TPBFC2- 21	12859	8122	4737	16.92	760	480	280	2.7	0.990	0.000
TPBFC2- 22	7790	3647	4143	14.75	528	247	281	1.9	0.995	0.000

TABLE D-23 (Continued)

Run Ident.	q^{Tot} Btu/(Hr-Ft ²)	q^{NB}	q^{FC}	Temp. Diff. °F	h Meas Btu/(Hr-Ft ² -°F)	h NB	h FC	$\frac{h \text{ Meas}}{h \text{ FC}}$	G Flux 10 ⁴ Lbm Hr-Ft ²	Reciprocal Martinelli Parameter
TPBFC2- 23	5066	1432	3634	12.94	391	111	281	1.4	0.995	0.000
TPBFC2- 24	3074	198	2876	10.24	300	19	281	1.1	0.995	0.000
TPBFC2- 25	34764	27935	6829	20.43	1702	1368	334	5.1	1.316	0.000
TPBFC2- 26	20355	14144	6211	18.54	1098	763	335	3.3	1.321	0.000
TPBFC2- 27	12831	7157	5674	16.97	756	422	334	2.3	1.316	0.000
TPBFC2- 28	6347	1756	4591	13.70	463	128	335	1.4	1.321	0.000
TPBFC2- 29	4048	347	3701	11.02	367	31	336	1.1	1.326	0.000
TPBFC2- 30	15291	12790	2501	17.75	862	721	141	6.1	0.329	0.000
TPBFC2- 31	10246	7955	2291	16.28	629	489	141	4.5	0.328	0.000
TPBFC2- 32	5065	3226	1839	13.07	387	247	141	2.8	0.328	0.000
TPBFC2- 33	3134	1699	1435	10.20	307	167	141	2.2	0.328	0.000
TPBFC2- 34	25111	20957	4154	19.07	1317	1099	218	6.0	0.663	0.000
TPBFC2- 35	25250	21096	4154	19.09	1323	1105	218	6.1	0.662	0.000
TPBFC2- 36	17801	13948	3953	18.15	981	763	218	4.5	0.663	0.000
TPBFC2- 37	17889	13970	3919	17.99	994	776	218	4.6	0.663	0.000
TPBFC2- 38	12750	9042	3708	16.99	750	532	218	3.4	0.665	0.000
TPBFC2- 39	12756	9039	3717	17.03	749	531	218	3.4	0.665	0.000
TPBFC2- 40	7690	4525	3165	14.52	530	312	218	2.4	0.664	0.000
TPBFC2- 41	7692	4480	3212	14.75	522	304	218	2.4	0.663	0.000
TPBFC2- 42	5076	2342	2734	12.55	404	187	218	1.9	0.663	0.000
TPBFC2- 43	5092	2394	2698	12.40	411	193	218	1.9	0.662	0.000
TPBFC2- 44	30077	24521	5556	19.78	1520	1239	281	5.4	0.995	0.000
TPBFC2- 45	20275	15086	5189	18.48	1097	817	281	3.9	0.995	0.000
TPBFC2- 46	30260	24543	5717	20.36	1486	1206	281	5.3	0.995	0.000

TABLE D-23 (Continued)

Run Ident.	$\frac{q^{Tot}}{Btu/(Hr-Ft^2)}$	$\frac{q^{NB}}{Btu/(Hr-Ft^2)}$	$\frac{q^{FC}}{Btu/(Hr-Ft^2)}$	Temp. Diff. °F	$\frac{h_{Meas}}{Btu/(Hr-Ft^2-°F)}$	$\frac{h_{NB}}{Btu/(Hr-Ft^2-°F)}$	$\frac{h_{FC}}{Btu/(Hr-Ft^2-°F)}$	$\frac{h_{Meas}}{h_{FC}}$	G Flux $\frac{10^6 Lbm}{Hr-Ft^2}$	Reciprocal Martinelli Parameter
Water with Essentially Zero Thermodynamic Quality										
TPBWC2- 1	50411	38917	11494	19.07	2644	2041	603	4.4	0.106	0.000
TPBWC2- 2	34905	23883	11022	18.35	1902	1301	601	3.2	0.106	0.000
TPBWC2- 3	61153	44922	16231	17.40	3514	2581	933	3.8	0.215	0.000
TPBWC2- 4	51027	34952	16075	17.20	2966	2032	934	3.2	0.215	0.000
TPBWC2- 5	41135	25221	15914	17.06	2411	1478	933	2.6	0.215	0.000
TPBWC2- 6	30934	15410	15524	16.64	1859	926	933	2.0	0.215	0.000
TPBWC2- 7	25871	10709	15162	16.25	1592	659	933	1.7	0.215	0.000
TPBWC2- 8	18306	4108	14198	15.22	1203	270	933	1.3	0.215	0.000
TPBWC2- 9	71577	38743	32834	22.82	3136	1698	1439	2.2	0.431	0.000
TPBWC2- 10	61211	28990	32221	22.38	2735	1296	1440	1.9	0.432	0.000
TPBWC2- 11	51434	19626	31808	22.11	2326	888	1439	1.6	0.431	0.000
TPBWC2- 12	40531	9561	30970	21.53	1883	444	1439	1.3	0.431	0.000
TPBWC2- 13	30466	963	29523	20.52	1485	47	1438	1.0	0.431	0.000
TPBWC2- 14	70616	40513	30103	20.92	3375	1936	1439	2.3	0.431	0.000
TPBWC2- 15	70698	39440	31258	21.71	3257	1817	1440	2.3	0.432	0.000
TPBWC2- 16	61113	30349	30764	21.36	2861	1421	1440	2.0	0.432	0.000
TPBWC2- 17	50451	19229	31222	21.69	2326	886	1439	1.6	0.432	0.000
TPBWC2- 18	40470	10557	29913	20.77	1948	508	1440	1.4	0.432	0.000
TPBWC2- 19	30422	1773	28649	19.91	1528	89	1439	1.1	0.431	0.000
TPBWC2- 20	70486	30596	39890	27.78	2538	1102	1436	1.8	0.430	0.000
TPBWC2- 21	64945	39740	25205	17.51	3710	2270	1440	2.6	0.432	0.000
TPBWC2- 22	35240	11494	23746	16.54	2131	695	1436	1.5	0.430	0.000
TPBWC2- 23	34506	9788	24718	17.22	2004	569	1436	1.4	0.430	0.000
TPBWC2- 24	64748	31268	33480	23.27	2782	1344	1439	1.9	0.431	0.000
TPBWC2- 25	65418	31052	34366	18.59	3518	1670	1848	1.9	0.646	0.000

TABLE D-23 (Continued)

Run Ident.	$q^{*}Tot$ Btu/(Hr-Ft ²)	$q^{*}NB$ Btu/(Hr-Ft ²)	$q^{*}PC$ Btu/(Hr-Ft ²)	Temp. Diff. °F	h Meas Btu/(Hr-Ft ² -°F)	h NB Btu/(Hr-Ft ² -°F)	h PC Btu/(Hr-Ft ² -°F)	h Meas h PC	G Flux 10 ⁴ Lbm Hr-Ft ²	Reciprocal Martinelli Parameter
TPBWC2- 26	56135	21851	34284	18.55	3026	1178	1848	1.6	0.646	0.000
TPBWC2- 27	35106	3102	32004	17.32	2027	179	1848	1.1	0.646	0.000
TPBWC2- 28	45536	12118	33418	18.08	2519	670	1848	1.4	0.646	0.000
TPBWC2- 29	30663	166	30842	16.64	1843	10	1854	1.0	0.649	0.000
TPBWC2- 30	64540	12791	51749	23.52	2745	544	2201	1.2	0.852	0.000
TPBWC2- 31	34767	23565	11202	18.65	1864	1263	601	3.1	0.106	0.000
TPBWC2- 32	34880	23632	11248	18.80	1856	1257	598	3.1	0.105	0.000
TPBWC2- 33	34832	23669	11163	18.66	1867	1269	598	3.1	0.105	0.000
TPBWC2- 34	70804	39232	31572	21.94	3227	1788	1439	2.2	0.431	0.000
TPBWC2- 35	70774	39115	31659	22.00	3216	1778	1439	2.2	0.431	0.000
TPBWC2- 36	60768	29142	31626	21.97	2766	1327	1440	1.9	0.432	0.000
TPBWC2- 37	60847	29040	31807	22.09	2754	1315	1440	1.9	0.432	0.000
TPBWC2- 38	50967	18967	32000	22.23	2293	853	1439	1.6	0.432	0.000
TPBWC2- 39	50825	19505	31320	21.79	2332	895	1437	1.6	0.431	0.000
TPBWC2- 40	40347	11222	29125	20.23	1994	555	1440	1.4	0.432	0.000
TPBWC2- 41	40367	10963	29404	20.37	1982	538	1444	1.4	0.434	0.000
TPBWC2- 42	34903	10885	24018	16.73	2087	651	1436	1.5	0.430	0.000
TPBWC2- 43	34973	10486	24487	17.05	2051	615	1436	1.4	0.430	0.000

TABLE D-24

ROUSEMON ADDITIVE CORRELATION DATA FOR ANNULAR FLOW BOILING OF FREON-113
ON HEAT METER HM-2

Run Ident.	$q^{*}Tot$ Btu/(Hr-Ft ²)	$q^{*}NB$ Btu/(Hr-Ft ²)	$q^{*}FC$ Btu/(Hr-Ft ²)	Temp. Diff. °F	h Meas Btu/(Hr-Ft ² -°F)	h NB Btu/(Hr-Ft ² -°F)	h FC Btu/(Hr-Ft ² -°F)	$\frac{h \text{ Meas}}{h \text{ FC}}$	G Flux $\frac{10^6 \text{ Lbm}}{\text{Hr-Ft}^2}$	Reciprocal Martinelli Parameter
TPXPC2- 1	25058	22392	2666	18.99	1319	1179	140	9.4	0.326	0.280
TPXPC2- 2	17862	15355	2507	17.86	1000	860	140	7.1	0.326	0.280
TPXPC2- 3	10219	7987	2232	15.90	643	502	140	4.6	0.326	0.283
TPXPC2- 4	5194	3536	1658	11.81	440	299	140	3.1	0.326	0.280
TPXPC2- 5	25266	22600	2666	19.05	1327	1187	140	9.5	0.325	0.522
TPXPC2- 6	20254	17691	2563	18.32	1106	966	140	7.9	0.325	0.517
TPXPC2- 7	15334	12913	2421	17.30	886	746	140	6.3	0.325	0.520
TPXPC2- 8	7796	5828	1968	14.06	555	415	140	4.0	0.325	0.514
TPXPC2- 9	22760	20146	2614	18.74	1215	1075	139	8.7	0.323	0.798
TPXPC2- 10	12922	10622	2300	16.43	787	647	140	5.6	0.325	0.798
TPXPC2- 11	7790	5904	1886	13.47	579	438	140	4.1	0.325	0.765
TPXPC2- 12	22751	20152	2599	18.69	1217	1078	139	8.8	0.321	1.054
TPXPC2- 13	15470	13088	2382	17.13	903	764	139	6.5	0.321	1.051
TPXPC2- 14	7742	5899	1843	13.23	585	446	139	4.2	0.322	1.057
TPXPC2- 16	25225	20926	4299	19.85	1271	1054	217	5.9	0.656	0.264
TPXPC2- 17	20245	16155	4090	18.88	1072	855	217	4.9	0.656	0.262
TPXPC2- 18	15363	11517	3846	17.76	865	649	217	4.0	0.656	0.267
TPXPC2- 19	10265	6844	3421	15.78	650	434	217	3.0	0.657	0.264
TPXPC2- 20	5177	2934	2243	10.32	502	284	217	2.3	0.660	0.263
TPXPC2- 21	2148	1106	1042	4.79	448	231	217	2.1	0.660	0.267
TPXPC2- 22	25089	20645	4444	20.54	1221	1005	216	5.6	0.655	0.535
TPXPC2- 23	17904	13776	4128	19.06	939	723	217	4.3	0.656	0.536
TPXPC2- 24	10379	6961	3418	15.80	657	441	216	3.0	0.655	0.533
TPXPC2- 25	29908	24389	5519	19.69	1519	1239	280	5.4	0.990	0.244

TABLE D-24 (Continued)

Run Ident.	q^{Tot} Btu/(Hr-Ft ²)	q^{NB}	q^{PC}	Temp. Diff. °F	h Meas Btu/(Hr-Ft ² -°F)	h NB	h PC	h Meas h PC	G Flux 10 ⁴ Lbm Hr-Ft ²	Reciprocal Martinelli Parameter
TPXPC2- 26	22639	17364	5275	18.82	1203	922	280	4.3	0.990	0.256
TPXPC2- 27	15406	10572	4834	17.20	896	615	281	3.2	0.995	0.253
TPXPC2- 28	10380	6161	4219	15.06	689	409	280	2.5	0.990	0.252
TPXPC2- 29	4421	2303	2118	7.56	585	305	280	2.1	0.990	0.248
TPXPC2- 30	27916	22421	5495	13.61	1423	1143	280	5.1	0.987	0.308
TPXPC2- 31	20503	15383	5120	18.27	1122	842	280	4.0	0.987	0.303
TPXPC2- 32	12845	8237	4608	16.45	781	501	280	2.8	0.987	0.316
TPXPC2- 33	6206	3297	2909	10.38	598	318	280	2.1	0.987	0.318
TPXPC2- 34	34552	20499	6053	19.72	1752	1445	307	5.7	1.141	0.280
TPXPC2- 35	25062	19410	5652	18.41	1362	1055	307	4.4	1.141	0.274
TPXPC2- 36	17652	12353	5299	17.26	1023	716	307	3.3	1.141	0.275
TPXPC2- 37	10202	5839	4363	14.21	718	411	307	2.3	1.141	0.274
TPXPC2- 38	7668	3918	3750	12.21	628	321	307	2.0	1.141	0.281
TPXPC2- 39	5051	2353	2698	8.77	576	268	308	1.9	1.145	0.279

TABLE D-25

CHRONOLOGICAL CROSS REFERENCE LISTING OF ALL EXPERIMENTS

Run	Date	Hour	HN-1	Expt.	HN-2	Expt.
Flcw Boiling Experiments						
WN002	08-02-73	1330			SPCWN2-	1
WN004	08-03-73	0945			SPCWN2-	2
WN007	08-03-73	1310			SPCWN2-	3
WN009	08-03-73	1500			SPCWN2-	4
WN011	08-09-73	1245			SPCWN2-	5
WN012	08-09-73	1330			SPCWN2-	6
WN014	08-09-73	1550			SPCWN2-	7
WN017	08-10-73	1045	SPCWN1-	1		
WN019	08-10-73	1200	SPCWN1-	2		
WN020	08-10-73	1245	SPCWN1-	3		
WN024	08-13-73	1315			SPCWN2-	8
WN025	08-13-73	1430			SPCWN2-	9
WN028	08-17-73	1145			SPCWN2-	10
WN030	08-17-73	1400			SPCWN2-	11
WN032	08-17-73	1610			SPCWN2-	12
WN033	08-20-73	1015			SPCWN2-	13
WN035	08-20-73	1300	SPCWN1-	4		
WN046	08-21-73	1445	SPCWN1-	5		
WN057	08-23-73	1345			SPCWN2-	14
WN072	08-28-73	1230			SPCWN2-	15
WN074	08-28-73	1440			SPCWN2-	16
WN075	08-28-73	1520			SPCWN2-	17
WN081	09-04-73	1540			SPCWN2-	18
WN083	09-05-73	1045			SPCWN2-	19
WN084	09-05-73	1200			SPCWN2-	20
WN128	09-21-73	2030			SPCWN2-	21
WN129	09-21-73	2100			SPCWN2-	22
WN130	09-21-73	2130			SPCWN2-	23
WN131	09-21-73	2200			SPCWN2-	24
WN152	09-24-73	2240			SPCWN2-	25
WN153	09-24-73	2315			SPCWN2-	26
WN156	09-26-73	1000			SPCWN2-	27
WN167	09-26-73	1325			SPCWN2-	28
WN168	09-26-73	1500			SPCWN2-	29
WN169	10-01-73	1030			SPCWN2-	30
WN170	10-01-73	1115			SPCWN2-	31
WN171	10-01-73	1300			SPCWN2-	32
WN172	10-01-73	1330			SPCWN2-	33
WN173	10-01-73	1430	SPCWN1-	6	SPCWN2-	44
WN174	10-01-73	1530	SPCWN1-	7	SPCWN2-	45

TABLE D-25 (Continued)

Run	Date	Hour	MM-1 Expt.	MM-2 Expt.
WM175	10-01-73	1620	SPCM1- 8	SPCM2- 46
WM176	10-02-73	0940		SPCM2- 34
WM177	10-02-73	1045		SPCM2- 35
WM178	10-02-73	1200		SPCM2- 36
WM179	10-02-73	1300	SPCM1- 9	SPCM2- 37
WM180	10-02-73	1400	SPCM1- 10	SPCM2- 47
WM181	10-02-73	1445	SPCM1- 10	SPCM2- 48
WM182	10-02-73	1530	SPCM1- 11	SPCM2- 49
WM183	10-02-73	1615		SPCM2- 38
WM184	10-02-73	1645		SPCM2- 39
WM185	10-02-73	1730		SPCM2- 40
WM186	10-02-73	1800		SPCM2- 41
WM187	10-02-73	1835	SPCM1- 12	SPCM2- 50
WM188	10-02-73	1915	SPCM1- 13	SPCM2- 51
WM189	10-02-73	2000	SPCM1- 14	SPCM2- 52
WM190	10-02-73	2040	SPCM1- 15	SPCM2- 53
WM195	10-03-73	1240		SPCM2- 42
WM203	10-03-73	1740		SPCM2- 43
WM212	10-04-73	1635	SPCM1- 16	
WM213	10-04-73	1715	SPCM1- 17	
WM001	10-17-73	1700		SPCM2- 9
WM002	10-17-73	1730		TPBWC2- 10
WM003	10-17-73	1810		TPBWC2- 11
WM004	10-17-73	1850		TPBWC2- 12
WM005	10-17-73	1925		TPBWC2- 13
WM006	10-17-73	1945		SPCM2- 1
WM007	10-17-73	2015		SPCM2- 2
WM008	10-17-73	2045		SPCM2- 3
WM009	10-18-73	0930		TPBWC2- 14
WM010	10-18-73	1000	TPBWC1- 4	TPBWC2- 14
WM011	10-18-73	1030	TPBWC1- 5	TPBWC2- 35
WM012	10-18-73	1100		TPBWC2- 15
WM013	10-18-73	1210		TPBWC2- 16
WM014	10-18-73	1300	TPBWC1- 6	TPBWC2- 36
WM015	10-18-73	1330	TPBWC1- 7	TPBWC2- 37
WM016	10-18-73	1400		TPBWC2- 17
WM017	10-18-73	1550	TPBWC1- 8	TPBWC2- 38
WM018	10-18-73	1630	TPBWC1- 9	TPBWC2- 39
WM019	10-18-73	1710		TPBWC2- 18
WM020	10-18-73	1745	TPBWC1- 10	TPBWC2- 40
WM021	10-18-73	1830	TPBWC1- 11	TPBWC2- 41
WM022	10-18-73	1915		TPBWC2- 19
WM023	10-19-73	1950	TPBWC1- 12	SPCM2- 16
WM024	10-19-73	2030	TPBWC1- 13	SPCM2- 17
WM025	10-18-73	2100	TPBWC1- 14	SPCM2- 18
WM026	10-18-73	2130	TPBWC1- 15	SPCM2- 19
WM027	10-18-73	2200	TPBWC1- 16	SPCM2- 20

TABLE D-25 (Continued)

Run	Date	Hour	HH-1	Expt.	HH-2	Expt.
WC028	10-19-73	0945			SPCWC2-	4
WC029	10-19-73	1030	TPBWC1-	17	SPCWC2-	21
WC030	10-19-73	1100	TPBWC1-	18	SPCWC2-	22
WC031	10-19-73	1130	TPBWC1-	19	SPCWC2-	23
WC032	10-19-73	1200	TPBWC1-	20	SPCWC2-	24
WC033	10-19-73	1230			SPCWC2-	5
WC034	10-19-73	1300			SPCWC2-	6
WC035	10-19-73	1330			SPCWC2-	7
WC036	10-19-73	1400			SPCWC2-	8
WC037	10-19-73	1515			TPBWC2-	20
WC038	10-23-73	1145			TPBWC2-	21
WC039	10-23-73	1215			TPBWC2-	22
WC040	10-23-73	1240	TPBWC1-	21	TPBWC2-	42
WC041	10-23-73	1310	TPBWC1-	22	TPBWC2-	43
WC042	10-23-73	1345			TPBWC2-	23
WC043	10-23-73	1440			TPBWC2-	25
WC044	10-23-73	1500			TPBWC2-	26
WC045	10-23-73	1530			TPBWC2-	27
WC046	10-23-73	1550			SPCWC2-	9
WC047	10-23-73	1620	SPCWC1-	1	SPCWC2-	14
WC048	10-23-73	1700	SPCWC1-	2	SPCWC2-	15
WC049	10-23-73	1720			SPCWC2-	10
WC050	10-23-73	1815			TPBWC2-	28
WC051	10-23-73	1840			TPBWC2-	29
WC052	10-23-73	1930			TPBWC2-	3
WC053	10-23-73	2000			TPBWC2-	4
WC054	10-23-73	2030			TPBWC2-	5
WC055	10-23-73	2100			TPBWC2-	6
WC056	10-23-73	2130			TPBWC2-	7
WC057	10-23-73	2215			TPBWC2-	8
WC058	10-24-73	0935			TPBWC2-	24
WC059	10-24-73	1025			TPBWC2-	30
WC060	10-24-73	1050			SPCWC2-	11
WC061	10-24-73	1140			SPCWC2-	12
WC062	10-24-73	1230			SPCWC2-	13
WC063	10-24-73	1320			TPBWC2-	1
WC064	10-24-73	1350			TPBWC2-	2
WC065	10-24-73	1430	TPBWC1-	1	TPBWC2-	31
WC066	10-24-73	1500	TPBWC1-	2	TPBWC2-	32
WC067	10-24-73	1540	TPBWC1-	3	TPBWC2-	33
PC001	11-06-73	1050			TPBWC2-	6
PC002	11-06-73	1120			TPBWC2-	7
PC003	11-06-73	1230			TPBWC2-	8
PC004	11-06-73	1330			TPBWC2-	9
PC005	11-06-73	1415			TPBWC2-	10
PC006	11-06-73	1450			TPBWC2-	11

TABLE D-25 (Continued)

Run	Date	Hour	RR-1 Expt.	RR-2 Expt.
PC007	11-06-73	1530		TPBFC2- 12
PC008	11-06-73	1615		TPBFC2- 13
PC009	11-06-73	1645		SPCFC2- 1
PC010	11-06-73	1720		SPCFC2- 2
PC011	11-07-73	0930		TPBFC2- 14
PC012	11-07-73	1120	TPBFC1- 7	TPBFC2- 34
PC013	11-07-73	1215	TPBFC1- 8	TPBFC2- 35
PC014	11-07-73	1245		TPBFC2- 15
PC015	11-07-73	1330		TPBFC2- 16
PC016	11-07-73	1400	TPBFC1- 9	TPBFC2- 36
PC017	11-07-73	1430	TPBFC1- 10	TPBFC2- 37
PC018	11-07-73	1500		TPBFC2- 17
PC019	11-07-73	1545	TPBFC1- 11	TPBFC2- 38
PC020	11-07-73	1615	TPBFC1- 12	TPBFC2- 39
PC021	11-08-73	0930		TPBFC2- 18
PC022	11-08-73	1010	TPBFC1- 13	TPBFC2- 40
PC023	11-08-73	1045	TPBFC1- 14	TPBFC2- 41
PC024	11-08-73	1130	TPBFC1- 15	TPBFC2- 42
PC025	11-08-73	1240	TPBFC1- 16	TPBFC2- 43
PC026	11-08-73	1315	TPBFC1- 17	SPCFC2- 11
PC027	11-08-73	1415	TPBFC1- 18	SPCFC2- 12
PC029	11-08-73	1520	TPBFC1- 19	SPCFC2- 13
PC030	11-08-73	1555		SPCFC2- 3
PC031	11-08-73	1620		SPCFC2- 4
PC032	11-09-73	1215	TPBFC1- 20	SPCFC2- 14
PC033	11-09-73	1330	TPBFC1- 21	SPCFC2- 15
PC034	11-09-73	1500	SPCFC1- 1	SPCFC2- 9
PC035	11-09-73	1610	SPCFC1- 2	SPCFC2- 10
PC036	11-12-73	1020		TPXFC2- 16
PC037	11-12-73	1100		TPXFC2- 17
PC038	11-12-73	1150		TPXFC2- 18
PC039	11-12-73	1230		TPXFC2- 19
PC040	11-12-73	1320		TPXFC2- 20
PC041	11-12-73	1405		TPXFC2- 21
PC042	11-12-73	1445		TPXFC2- 22
PC043	11-12-73	1530		TPXFC2- 23
PC044	11-12-73	1610		TPXFC2- 24
PC045	11-13-73	0945		TPBFC2- 1
PC046	11-13-73	1020	TPBFC1- 1	TPBFC2- 30
PC047	11-13-73	1105	TPBFC1- 2	TPBFC2- 31
PC048	11-13-73	1125		TPBFC2- 2
PC049	11-13-73	1215		TPBFC2- 3
PC050	11-13-73	1250	TPBFC1- 3	TPBFC2- 32
PC051	11-13-73	1340	TPBFC1- 4	TPBFC2- 33
PC052	11-13-73	1405		TPBFC2- 4
PC053	11-13-73	1440		TPBFC2- 5

TABLE D-25 (Continued)

Run	Date	Hour	HH-1	Expt.	HH-2	Expt.
PC054	11-13-73	1515	TPBPC1-	5	SPCPC2-	16
PC055	11-13-73	1540	TPBPC1-	6	SPCPC2-	17
PC056	11-13-73	1615			SPCPC2-	5
PC057	11-14-73	1015			TPXPC2-	1
PC058	11-14-73	1035			TPXPC2-	2
PC059	11-14-73	1100			TPXPC2-	3
PC060	11-14-73	1130			TPXPC2-	4
PC061	11-14-73	1210			TPXPC2-	5
PC062	11-14-73	1235			TPXPC2-	6
PC063	11-14-73	1300			TPXPC2-	7
PC064	11-14-73	1325			TPXPC2-	8
PC065	11-14-73	1400			TPXPC2-	9
PC066	11-14-73	1440			TPXPC2-	10
PC067	11-14-73	1500			TPXPC2-	11
PC068	11-14-73	1540			TPXPC2-	12
PC069	11-14-73	1600			TPXPC2-	13
PC070	11-14-73	1640			TPXPC2-	14
PC072	11-15-73	0920			TPBPC2-	19
PC073	11-15-73	1000	TPBPC1-	22	TPBPC2-	18
PC075	11-15-73	1100			TPBPC2-	20
PC074	11-15-73	1030	TPBPC1-	23	TPBPC2-	15
PC076	11-15-73	1120			TPBPC2-	21
PC077	11-15-73	1145			TPBPC2-	22
PC078	11-15-73	1210			TPBPC2-	23
PC079	11-15-73	1235			TPBPC2-	24
PC081	11-15-73	1340			SPCPC2-	6
PC082	11-15-73	1400	TPBPC1-	24	TPBPC2-	16
PC083	11-15-73	1450			TPBPC2-	25
PC084	11-15-73	1510			TPBPC2-	26
PC085	11-15-73	1540			TPBPC2-	27
PC086	11-15-73	1605			TPBPC2-	28
PC087	11-15-73	1640			TPBPC2-	29
PC088	11-15-73	1720			SPCPC2-	7
PC089	11-15-73	1800			SPCPC2-	8
PC103	11-19-73	1405			TPXPC2-	25
PC104	11-19-73	1430			TPXPC2-	26
PC105	11-19-73	1505			TPXPC2-	27
PC106	11-19-73	1535			TPXPC2-	28
PC107	11-19-73	1610			TPXPC2-	29
PC108	11-19-73	1700			TPXPC2-	30
PC109	11-19-73	1730			TPXPC2-	31
PC110	11-19-73	1800			TPXPC2-	32
PC111	11-19-73	1830			TPXPC2-	33
PC112	11-20-73	0915			TPXPC2-	34
PC113	11-20-73	0950			TPXPC2-	35
PC114	11-20-73	1015			TPXPC2-	36

TABLE D-25 (Continued)

Run	Date	Hour	HH-1	Expt.	HH-2	Expt.
PC115	11-20-73	1050			T2KPC2-	37
PC116	11-20-73	1115			T2KPC2-	38
PC117	11-20-73	1150			T2KPC2-	39
Pool Boiling Experiments						
CAL02	12-04-73	1245	HPBPC1-	1	HPBPC2-	1
CAL03	12-04-73	1345			HPBPC2-	2
CAL04	12-04-73	1450			HPBPC2-	3
PFC01	12-05-73	1045	HPBPC1-	2	HPBPC2-	4
PFC03	12-05-73	1605	HPBPC1-	3	HPBPC2-	5
PFC04	12-05-73	1710	HPBPC1-	4	HPBPC2-	6
PFC05	12-05-73	1810	HPBPC1-	5	HPBPC2-	7
PFC06	12-05-73	1915	HPBPC1-	6	HPBPC2-	8
PFC07	12-05-73	2025	HPBPC1-	7	HPBPC2-	9
PFC08	12-05-73	2130	HPBPC1-	8	HPBPC2-	10
CAL06	12-06-73	1015	HPBPC1-	9	HPBPC2-	11
PFC09	12-06-73	1115	HPBPC1-	10	HPBPC2-	12
PFC10	12-06-73	1220	HPBPC1-	11	HPBPC2-	13
PFC11	12-06-73	1320	HPBPC1-	12	HPBPC2-	14
PFC12	12-06-73	1415	HPBPC1-	13	HPBPC2-	15
PFC13	12-06-73	1500	HPBPC1-	14	HPBPC2-	16
PFC14	12-06-73	1535	HPBPC1-	15	HPBPC2-	17
PFC15	12-06-73	1620	HPBPC1-	16	HPBPC2-	18
PFC16	12-06-73	1700	HPBPC1-	17	HPBPC2-	19
PFC17	12-06-73	1740	HPBPC1-	18	HPBPC2-	20
PFC18	12-06-73	1830	HPBPC1-	19	HPBPC2-	21
PFC19	12-06-73	1910	HPBPC1-	20	HPBPC2-	22
PFC20	12-06-73	2000	HPBPC1-	21	HPBPC2-	23
CAL07	12-07-73	0830	HPBPC1-	22	HPBPC2-	24
PFC21	12-07-73	1015	HPBPC1-	23	HPBPC2-	25
PFC22	12-07-73	1050	HPBPC1-	24	HPBPC2-	26
PFC23	12-07-73	1130	HPBPC1-	25	HPBPC2-	27
PFC24	12-07-73	1210	HPBPC1-	26	HPBPC2-	28
PFC25	12-07-73	1245	HPBPC1-	27	HPBPC2-	29
PFC26	12-07-73	1330	HPBPC1-	28	HPBPC2-	30
PFC27	12-07-73	1410	HPBPC1-	29	HPBPC2-	31
PFC28	12-07-73	1500	HPBPC1-	30	HPBPC2-	32
PFC29	12-07-73	1530	HPBPC1-	31	HPBPC2-	33
PFC30	12-07-73	1620	HPBPC1-	32	HPBPC2-	34
PFC33	12-12-73	0820	VPBPC1-	1		
PFC34	12-12-73	0940			VPBPC2-	1
PFC35	12-12-73	1030			VPBPC2-	2
PFC36	12-12-73	1120			VPBPC2-	3
PFC37	12-12-73	1200			VPBPC2-	4
PFC38	12-12-73	1315			VPBPC2-	5

TABLE D-25 (Continued)

Run	Date	Hour	HH-1	Expt.	HH-2	Expt.
PFC39	12-12-73	1430			VPBPC2-	6
PFC40	12-12-73	1455			VPBPC2-	7
PFC41	12-12-73	1620			VPBPC2-	8
PFC42	12-12-73	1700			VPBPC2-	9
PFC43	12-12-73	1740			VPBPC2-	10
PFC44	12-12-73	1820			VPBPC2-	11
PFC45	12-12-73	1855			VPBPC2-	12
PFC46	12-12-73	1940			VPBPC2-	13
PFC47	12-12-73	2020			VPBPC2-	14
PFC48	12-12-73	2055			VPBPC2-	15
PFC49	12-12-73	2135			VPBPC2-	16
PFC50	12-13-73	0830	VPBPC1-	2	VPBPC2-	17
PFC51	12-13-73	0920			VPBPC2-	18
PFC52	12-13-73	0955			VPBPC2-	19
PFC53	12-13-73	1045	VPBPC1-	3		
PFC54	12-13-73	1120	VPBPC1-	4		
PFC55	12-13-73	1210	VPBPC1-	5		
PFC56	12-13-73	1240	VPBPC1-	6		
PFC57	12-13-73	1320	VPBPC1-	7		
PFC58	12-13-73	1405	VPBPC1-	8		
PFC59	12-13-73	1450	VPBPC1-	9		
PFC60	12-13-73	1525	VPBPC1-	10		
PFC61	12-13-73	1610	VPBPC1-	11		
PFC62	12-13-73	1705	VPBPC1-	12		
PFC63	12-13-73	1755	VPBPC1-	13		
PFC64	12-14-73	0820	VPBPC1-	14	VPBPC2-	20
PFC65	12-14-73	0930	VPBPC1-	16	VPBPC2-	34
PFC66	12-14-73	1010	VPBPC1-	17	VPBPC2-	35
PFC67	12-14-73	1045	VPBPC1-	18	VPBPC2-	36
PFC68	12-14-73	1130	VPBPC1-	19	VPBPC2-	37
PFC69	12-14-73	1210	VPBPC1-	20	VPBPC2-	38
PFC70	12-14-73	1250	VPBPC1-	21	VPBPC2-	39
PFC71	12-14-73	1330	VPBPC1-	22	VPBPC2-	40
PFC72	12-14-73	1400	VPBPC1-	23	VPBPC2-	41
PFC73	12-14-73	1440	VPBPC1-	24	VPBPC2-	42
PFC74	12-14-73	1515	VPBPC1-	25	VPBPC2-	43
PFC75	12-14-73	1615	VPBPC1-	26	VPBPC2-	44
PFC76	12-15-73	1200	VPBPC1-	15	VPBPC2-	21
PFC77	12-15-73	1600			VPBPC2-	22
PFC01	12-18-73	0820	VPBPC1-	1	VPBPC2-	1
PFC02	12-18-73	1000			VPBPC2-	2
PFC03	12-18-73	1100			VPBPC2-	3
PFC04	12-18-73	1120			VPBPC2-	4
PFC05	12-18-73	1145			VPBPC2-	5
PFC06	12-18-73	1215			VPBPC2-	6
PFC07	12-18-73	1245			VPBPC2-	7

TABLE D-25 (Continued)

Pan	Date	Hour	HH-1	Expt.	HH-2	Expt.
PWC08	12-18-73	1315			VPBWC2-	8
PWC09	12-18-73	1350			VPBWC2-	9
PWC10	12-18-73	1450			VPBWC2-	10
PWC11	12-18-73	1520			VPBWC2-	11
PWC12	12-18-73	1540			VPBWC2-	12
PWC13	12-18-73	1610			VPBWC2-	13
PWC15	12-19-73	0830	VPBWC 1-	2	VPBWC2-	14
PWC16	12-19-73	0950			VPBWC2-	15
PWC17	12-19-73	1045			VPBWC2-	16
PWC18	12-19-73	1130	VPBWC 1-	3	VPBWC2-	19
PWC19	12-19-73	1215	VPBWC 1-	4	VPBWC2-	20
PWC20	12-19-73	1240	VPBWC 1-	5		
PWC21	12-19-73	1320	VPBWC 1-	6	VPBWC2-	21
PWC22	12-19-73	1355	VPBWC 1-	7	VPBWC2-	22
PWC23	12-19-73	1430	VPBWC 1-	8	VPBWC2-	23
PWC24	12-19-73	1500			VPBWC2-	17
PWC25	12-19-73	1540	VPBWC 1-	9	VPBWC2-	24
PWC26	12-19-73	1615	VPBWC 1-	10	VPBWC2-	25
PWC27	12-19-73	1655	VPBWC 1-	11	VPBWC2-	26
PWC28	12-19-73	1730	VPBWC 1-	12	VPBWC2-	27
PWC29	12-19-73	1800	VPBWC 1-	13	VPBWC2-	28
PWC30	12-20-73	0840	VPBWC 1-	14	VPBWC2-	18
PWC31	12-21-73	1000	VPBWC 1-	1	VPBWC2-	1
PWC32	12-21-73	1100	VPBWC 1-	2	VPBWC2-	2
PWC33	12-21-73	1140	VPBWC 1-	3	VPBWC2-	3
PWC34	12-21-73	1225	VPBWC 1-	4	VPBWC2-	4
PWC35	12-21-73	1305	VPBWC 1-	5	VPBWC2-	5
PWC36	12-21-73	1350	VPBWC 1-	6	VPBWC2-	6
PWC37	12-21-73	1430	VPBWC 1-	7	VPBWC2-	7
PWC38	12-21-73	1505	VPBWC 1-	8	VPBWC2-	8
PWC39	12-21-73	1615	VPBWC 1-	9	VPBWC2-	9
PWC40	12-21-73	1715	VPBWC 1-	10	VPBWC2-	10
PWC41	12-21-73	1815	VPBWC 1-	11	VPBWC2-	11
PWC42	12-22-73	0830	VPBWC 1-	12	VPBWC2-	12
PWC43	12-22-73	1000	VPBWC 1-	13	VPBWC2-	13
PWC44	12-22-73	1035	VPBWC 1-	14	VPBWC2-	14
PWC45	12-22-73	1120	VPBWC 1-	15	VPBWC2-	15
PWC46	12-22-73	1215	VPBWC 1-	16	VPBWC2-	16
PWC47	12-22-73	1305	VPBWC 1-	17	VPBWC2-	17
PWC48	12-22-73	1340	VPBWC 1-	18	VPBWC2-	18
PWC49	12-22-73	1410	VPBWC 1-	19	VPBWC2-	19
PWC50	12-22-73	1450	VPBWC 1-	20	VPBWC2-	20
PWC51	12-22-73	1540	VPBWC 1-	21	VPBWC2-	21
PWC52	12-22-73	1640	VPBWC 1-	22	VPBWC2-	22

APPENDIX E

ESTIMATE OF ERRORS

Errors of principal concern are those associated with heat flux, heat transfer coefficient, and the temperature difference. The functional relationships between these parameters are:

$$\Delta T_{\text{sat}} = T_w - T_{\text{sat}} \quad , \quad (\text{E-1})$$

$$q' = -k \text{ grad } T \quad , \quad (\text{E-2})$$

and

$$h = q' / \Delta T_{\text{sat}} \quad , \quad (\text{E-3})$$

where grad T is the gradient of the temperature distribution within the heat meter, evaluated at the surface.

A convenient method of computing the fractional error in these is to take the logarithm of both sides of the equations and then differentiate to obtain:

$$\frac{d(\Delta T_{\text{sat}})}{\Delta T_{\text{sat}}} = \frac{d(T_w - T_{\text{sat}})}{\Delta T_{\text{sat}}} \quad , \quad (\text{E-4})$$

$$\frac{dq'}{q'} = - \left(\frac{dk}{k} + \frac{d(\text{grad } T)}{\text{grad } T} \right) \quad , \quad (\text{E-5})$$

and

$$\frac{dh}{h} = \frac{dq'}{q'} - \frac{d(T_w - T_{\text{sat}})}{\Delta T_{\text{sat}}} \quad . \quad (\text{E-6})$$

Replacing the differentials with finite values and considering only the sum of absolute values for the individual error components to obtain a maximum error, the fractional error relations become:

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$$\frac{\delta \dot{Q}_{\text{sat}}}{\dot{Q}_{\text{sat}}} = \frac{\delta T_w}{T_w} + \frac{\delta T_{\text{sat}}}{T_{\text{sat}}} \quad , \quad (\text{E-5})$$

$$\frac{\delta q''}{q''} = \frac{\delta k}{k} + \frac{\delta \text{grad } T}{\text{grad } T} \quad , \quad (\text{E-6})$$

and

$$\frac{\delta h}{h} = \frac{\delta k}{k} + \frac{\delta \text{grad } T}{\text{grad } T} + \frac{\delta T_w}{T_w} + \frac{\delta T_{\text{sat}}}{T_{\text{sat}}} \quad . \quad (\text{E-7})$$

These may be used to estimate the maximum expected error; the probable error will, of course, be smaller.

The above expressions were used to estimate errors for four of the experiments to give a range of typical values expected; high and low heat flux experiments for both water and Freon were used for this purpose.

The equation used to calculate k [Eq. (1-1)] was estimated¹²² to be accurate within $\pm 6\%$. Thus, $\delta k/k = 0.06$. The thermocouples were calibrated over the temperature range of interest against standard indicators known to be accurate to $\pm 0.05^\circ\text{F}$ (see Appendix A). Based on the discussion in Appendix A, a value of 0.1°F appears to be a conservative estimate for the error in measured fluid temperature. The standard deviation obtained from the linear least-squares analysis of the measured temperatures within the heat meter were used for estimates of δT_w and δT_{sat} . Table E-1 tabulates the data used to obtain the error estimates given in Table E-2. The tabulated data have been rounded off from the higher precision values used in the actual calculations; hence, no discrepancies might appear apparent in the tabulation.

TABLE B-1
DATA FOR ESTIMATION OF ERRORS

	Freon Tests		Water Tests	
	Low Flux	High Flux	Low Flux	High Flux
Experiment No.	HPBFCE-001	HPBFCE-025	VPEWCE-001	VPEWCE-015
q'' (Btu/hr·ft ²)	512	19,138	1036	66,337
h (Btu/hr·ft ² ·°F)	70	1,825	680	4,987
ΔT_{sat} (°F)	11.53	21.44	1.59	14.46
T_{sat} (°F)	116.47	117.55	211.32	211.42
δT_{sat} (°F)	0.10	0.10	0.10	0.10
T_w (°F)	128.00	138.00	212.41	225.88
δ_w (°F)	0.05	0.20	0.03	0.30
grad T (°F/in.)	0.30	14.23	0.40	22.46
δ grad T (°F/in.)	0.04	0.17	0.03	0.33

TABLE E-2
RANGE OF ESTIMATED ERRORS

Experiment No.	q'' (Btu/hr·ft ²)	ΔT_{sat} (°F)	h (Btu/hr·ft ² ·°F)	Fractional Error			Error		
				$\frac{\delta q''}{q''}$	$\frac{\delta \Delta T_{sat}}{\Delta T_{sat}}$	$\frac{\delta h}{h}$	q'' (Btu/hr·ft ²)	ΔT_{sat} (°F)	h (Btu/hr·ft ² ·°F)
HPBFC2-003	812	11.53	70	0.151	0.013	0.164	123	0.15	11
HPBFC2-025	39,138	21.44	1825	0.000	0.013	0.103	3515	0.29	188
VPBWC2-001	1,095	1.59	689	0.089	0.084	0.173	97	0.13	119
VPBWC2-015	66,330	14.46	4587	0.033	0.034	0.067	2202	0.49	307

APPENDIX F

ALTERNATIVE ROHSENOW CORRELATION OF BOILING DATA

The Rohsenow pool boiling correlating equation was given earlier as Eq. (2-23); it is repeated here for convenience as:

$$\frac{C_{p,l} \Delta T_{\text{sat}}}{h_{fg}} = C_{sf} \left(\frac{q''}{\mu_l h_{fg}} \sqrt{\frac{g_c \sigma}{g(\rho_l - \rho_g)}} \right)^r \left(\frac{C_p \mu}{k} \right)_l^{1.7} \quad (\text{F-1})$$

Although it was originally suggested³² that the bubble Reynolds modulus exponent, r , should have a value of $1/3$ for most fluids to agree with the generally observed proportionality of $q'' \propto \Delta T_{\text{sat}}^3$, the availability of more extensive experimental data has led to the suggestion³³ that r should be permitted to take on different values (consistent with the surface-fluid factor, C_{sf}) to obtain better agreement with experimental observations. [Likewise, the Prandtl modulus exponent, 1.7 in the above equation, assumes the value of 1.0 for water³³ and remains 1.7 for other fluids.]

Equation (F-1) is usually written for correlating purposes in the form:

$$\frac{C_{p,l} \Delta T_{\text{sat}}}{h_{fg} N_{\text{Pr}}^{1.7}} = C_{sf} N_{\text{Re},b}^r \quad (\text{F-2})$$

If the left side of the equation is called x and $N_{\text{Re},b}$ is called y , taking the logarithm of both sides results in:

$$\ln x = r \ln y + \ln C_{sf} \quad (\text{F-3})$$

Equation (F-3) plots as a linear function on log-log coordinates. Thus, if the experimental data are cast in the form represented by x and y , a

linear regression analysis can be used to determine values for the slope (r) and the constant ($\ln C_{sf}$) in Eq. (F-3) that minimize the deviations in x . This is the method used by most boiling researchers to determine values reported for r and C_{sf} ; similarly, the values reported in Chapter V were determined as above. This ignores a fundamental premise of least-squares analysis; namely, that the parameter with the smaller experimental error (in reality assumed to be insignificant compared to the error in the other parameter) is taken to be the independent parameter and the deviations of the parameter with the larger experimental error are minimized in the least-squares sense.

Comparison of the maximum experimental errors in q'' and ΔT_{sat} (expressed as fractional errors) given in Table E-2 reveals that ΔT_{sat} in general has smaller fractional experimental errors than q'' . Thus, regression analysis applied to Eq. (F-3) violates the above mentioned premise. Formally, analysis of variance requires under these circumstances that Eq. (F-2) be written in the form:

$$y = (x/C_{sf})^{1/r} \quad (F-4)$$

Taking the logarithm of both sides results in another linear equation:

$$\ln y = \frac{1}{r} \ln x - \frac{1}{r} \ln C_{sf} \quad (F-5)$$

which should be used when the x parameter has a smaller experimental error than that associated with y .

The experimental data presented in Figs. V-15, V-16, V-23, and V-28 of Chapter V were analyzed, using Eq. (F-5), to determine alternative best fit values for C_{sf} and r under these rules. Table F-1 gives the results and a comparison with the values determined in Chapter V; Figs. F-1 through F-4 show the data plotted with the best fit

TABLE F-1

SUMMARY OF CORRELATING PARAMETERS AND STANDARD DEVIATIONS DETERMINED BY
ANALYSIS OF VARIANCE OF EQS. (F-3) AND (F-5)

Data Set	Eq. (F-3)			Eq. (F-5)		
	C_{sf}	r	Std. Dev. of $\ln x$	C_{sf}	r	Std. Dev. of $\ln y$
<u>Freon Pool Boiling</u>						
HM-1 Vertical	0.00275	0.132	0.017	0.00275	0.133	0.126
HM-1 Horizontal	0.00265	0.078	0.049	0.00273	0.113	0.524
HM-2 Vertical	0.00285	0.093	0.017	0.00286	0.096	0.176
HM-2 Horizontal	0.00267	0.076	0.046	0.00276	0.112	0.525
Combined Data	0.00272	0.092	0.049	0.00279	0.125	0.469
<u>Water Pool Boiling</u>						
HM-1 Vertical	0.00910	0.273	0.012	0.00911	0.274	0.043
HM-1 Horizontal	0.00897	0.063	0.053	0.00964	0.148	0.553
HM-2 Vertical	0.00861	0.155	0.015	0.00863	0.158	0.093
HM-2 Horizontal	0.00947	0.074	0.061	0.01035	0.165	0.524
Combined Data	0.00898	0.114	0.106	0.01089	0.324	0.545
Reference 33	0.0097	0.10	—	—	—	—
<u>Freon Flow Boiling</u>						
Low Quality (~0%) Tests	0.00262	0.144	0.050	0.00266	0.156	0.336
Annular Flow Tests	0.00276	0.192	0.030	0.00279	0.213	0.147

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Fig. F-1. Alternative Rohsenow pool boiling correlation for Freon-113.

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Fig. F-2. Alternative Rohsenow pool boiling correlation for water.

Fig. F-3. Alternative Rohsenow correlation of Freon flow boiling results.

Fig. F-4. Alternative Rohsenow correlation of Freon annular flow boiling.

correlating line. These figures are to be compared with Figs. V-15, V-16, V-23, and V-28, respectively, of Chapter V to show the difference in the correlating methods.

Table F-2 gives the distribution of the deviations from the best fit line as determined by the proper analysis of variance of the data; i.e., Eq. (F-5).

It is noted that little difference results in the values of C_{sf} as determined by the two analyses; the most noticeable effect is in the values of r for pool boiling of water. This is a result of the wide scatter in the data. The flow boiling data showed closer agreement than did the "true" pool boiling data. Considering the fact that the flow boiling data were obtained indirectly (by differences), the narrow data band (indicated by the Table F-2) gives additional support for the suitability of the Rohsenow additive technique for correlating Freon flow boiling data.

TABLE F-2

TABULATION OF PERCENT DEVIATION OF DATA FROM CORRELATION

Percent Deviation of Data from Correlation	Percent of Freon Pool Boiling Data within Deviation Percentage	Percent of Water Pool Boiling Data within Deviation Percentage	Percent of Freon Flow Boiling Data within Deviation Percentage	Percent of Freon Annular Flow Boiling Data within Deviation Percentage
10	17.7	10.9	28.6	46.4
20	38.0	21.8	58.6	89.3
30	51.9	36.4	77.1	96.4
40	67.1	60.0	84.3	100.0
50	75.9	74.5	90.0	100.0
> 50	24.1	25.5	10.0	0