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TRANSIENT-BOILING HEAT TRANSFER
FROM SPHERES TO SODIUM

by

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TABLE OF CONTENTS

	<u>Page</u>
NOMENCLATURE	10
ABSTRACT	11
I. INTRODUCTION	12
II. LITERATURE REVIEW	14
A. Film Boiling	14
1. Vertical Surfaces	14
2. Horizontal Surfaces	15
3. Horizontal Cylinders	16
4. Spheres	16
B. The Minimum Heat Flux	18
C. Transition Boiling	18
D. The Critical Heat Flux	18
E. Nucleate Boiling of Liquid Metals	19
III. THEORETICAL ANALYSIS	21
A. Introduction	21
B. Film Boiling	23
1. Introduction	23
2. Basic Model and Equations	23
IV. EXPERIMENTAL APPARATUS	30
A. Equipment	30
1. Sodium Pot	31
2. Sphere-thermocouple Assembly	31
3. Induction Heater	31
4. Transport System	32
5. Glovebox	34
B. Instrumentation	35
1. Thermocouples	35
2. Transducers	35
3. Bias Circuit	36
4. Charge Amplifiers	36

TABLE OF CONTENTS

	<u>Page</u>
5. Power Amplifiers	36
6. Data-recording Oscillograph	36
7. Glovebox Controls	36
V. EXPERIMENTAL PROCEDURES	37
A. Choice of Experimental Apparatus.	37
B. Calibration of Instruments	38
1. Amplifiers	38
2. Galvanometers	38
3. Pool Pressure Transducer	38
4. Temperature Controller	38
C. Sodium Experiments	39
1. Variables Sought	40
2. Experimental Procedure	41
3. Difficulties and Remedies	42
VI. DATA REDUCTION	48
A. Discussion	48
B. Accuracy of the Data-reduction Method	51
VII. RESULTS AND DISCUSSION	54
A. General	54
1. Accuracy of the Experimental Results	54
2. Reproducibility	56
3. Effect of Initial Sphere Temperature	57
4. Effect of Sodium Subcooling	59
5. Effect of Depth	63
6. Effect of Sphere Diameter	64
B. The Film-boiling Region	65
1. Saturated Film-boiling Heat-transfer Coefficient, h_{fb}	69
2. Radiative Heat-transfer Coefficient, h_r	70
3. Natural-convection Heat-transfer Coefficient in the Liquid, h_c	70
C. The Leidenfrost Point	72

TABLE OF CONTENTS

	<u>Page</u>
D. The Transition Region	74
E. The Critical Heat Flux q_{crit}^*	77
VIII. GENERAL DISCUSSION, CONCLUSIONS, AND RECOMMENDATIONS	80
APPENDIXES	
A. Sodium-boiling Curves	84
B. Experimental Techniques	85
C. FORTRAN Computer Programs	87
1. SPHERE	87
2. WRETC.	89
D. Preliminary Experiments with Water	91
1. Experimental Apparatus.	91
2. Experimental Procedures.	92
3. Experimental Results	97
E. Sample Calculation	99
1. Physical Properties of Vapor	99
2. Physical Properties of Liquid	99
3. Saturated Film-boiling Coefficient, h_{fb}	100
4. Radiation Coefficient, h_r	100
5. Natural-convection Coefficient, h_c	101
6. Total Film Coefficient, h_t	101
F. Further Discussion on the Effect of Subcooling on the Boiling Curve	102
ACKNOWLEDGMENTS	104
REFERENCES	105

LIST OF FIGURES

<u>No.</u>	<u>Title</u>	<u>Page</u>
1.	A Typical Boiling Curve	21
2.	Single-dome Model for Subcooled-film Boiling from Small Spheres	23
3.	Multiple-dome Model for Subcooled-film Boiling from Large Spheres	23
4.	Schematic Diagram of the Experimental Apparatus	30
5.	The Different Sphere Sizes Used: 1-, 3/4-, and 1/2-in. Diameter	32
6.	Cross Section of Sphere-thermocouple Assembly	32
7.	Sphere-thermocouple Assembly Ready for the Experiment . . .	33
8.	Transport System for Sodium Experiments	33
9.	Dry Box and Part of the Instrumentation Used in the Sodium Experiments	34
10.	X-radiograph of a 0.040-in.-dia Thermocouple	43
11.	X-radiograph of a Used Sphere, Showing the Thermocouple . . .	44
12.	Gamma Radiograph of the Sphere of Fig. 11, Showing Sharper Boundaries	44
13.	Old Sphere Design	44
14.	New Sphere Design, Showing Extension Neck	44
15.	Damaged Sphere Surface (initial sphere temperature = 4200°F; sodium temperature = 1292°F)	45
16.	Radial and Time Steps for the Difference Equations	49
17.	Typical Oscillograph Record Showing the Various Boiling Regimes, Run 36	51
18.	The Boiling Curve for Run 36 as Determined by the Data-reduction Method	52
19.	Reproducibility of the Experimental Results, q/A in Btu/hr-ft^2 and $T_w - T_{\text{sat}}$ in °F	56
20.	Effect of Changing the Initial Sphere Temperature on the Boiling Curves of Sodium; $T_{\text{Na}} = 1607^\circ\text{F}$, q/A in Btu/hr-ft^2 , $T_w - T_{\text{sat}}$ in °F	58
21.	Temperature Distribution in the Sphere near the Leidenfrost Point, $T_{\text{Na}} = 1607^\circ\text{F}$	58

LIST OF FIGURES

<u>No.</u>	<u>Title</u>	<u>Page</u>
22.	Effect of Changing the Degree of Subcooling on the Boiling Curves; $D = 1.00$ in., q/A in Btu/hr-ft ² , and $T_w - T_{sat}$ in °F .	60
23.	Effect of Changing the Degree of Subcooling on the Boiling Curves; $D = 0.75$ in., q/A in Btu/hr-ft ² , and $T_w - T_{sat}$ in °F .	60
24.	Effect of Changing the Degree of Subcooling on the Boiling Curves; $D = 0.50$ in.	60
25.	Effect of Changing the Depth on the Boiling Curve; $T_{Na} = 1589^\circ\text{F}$, q/A in Btu/hr-ft ² , and $T_w - T_{sat}$ in °F.	63
26.	Effect of Changing the Sphere Diameter on the Boiling Curve; $T_{Na} = 1607^\circ\text{F}$, q/A in Btu/hr-ft ² , and $T_w - T_{sat}$ in °F.	64
27.	Correlation of Film-boiling Data for 1-in. Spheres, $\Delta T_{sc} = 11.6^\circ\text{F}$	68
28.	Correlation of Film-boiling Data for 1-in. Spheres, $\Delta T_{sc} = 20.6^\circ\text{F}$	68
29.	Correlation of Film-boiling Data for 1-in. Spheres, $\Delta T_{sc} = 29.6^\circ\text{F}$	68
30.	Correlation of Film-boiling Data for 1-in. Spheres, $\Delta T_{sc} = 38.6^\circ\text{F}$	68
31.	Correlation of Film-boiling Data for 1-in. Spheres, $\Delta T_{sc} = 56.6^\circ\text{F}$	68
32.	Estimated Time to Reach the Leidenfrost Point from Pressure Trace and Final Boiling Curve.	72
33.	Effect of the Degree of Subcooling on the Violent Interaction in the Transition Region.	74
34.	Condition of the Sodium Pot following a Violent Interaction, $T_{Na} = 1472^\circ\text{F}$	75
D.1.	Experimental Apparatus Used in Water Experiments.	91
D.2.	Sodium Pot	92
D.3.	Pool Sliding Gate	92
D.4.	Pool Cold Trap.	92
D.5.	Sphere-Thermocouple Assembly	93
D.6.	Transformer and Power Leads	93
D.7.	Induction Coil.	94

LIST OF FIGURES

<u>No.</u>	<u>Title</u>	<u>Page</u>
D.8.	Dropping System for Water Experiments	94
D.9.	Effect of Subcooling on the Boiling Curve of Water; $T_{init} \sim 1418^{\circ}\text{F}$	97
D.10.	Effect of Subcooling on the Boiling Curve of Water; $T_{init} \sim 1148^{\circ}\text{F}$	97
D.11.	Effect of Subcooling on the Boiling Curve of Water; $T_{init} \sim 839^{\circ}\text{F}$	98
D.12.	Effect of Initial Water Temperature on the Boiling Curve; $\Delta T_{sc} = 18^{\circ}\text{F}$	98
F.1.	Predicted Effect of Increasing the Degree of Subcooling to Very Large Values on the Boiling Curve of Sodium	102

LIST OF TABLES

<u>No.</u>	<u>Title</u>	<u>Page</u>
I.	Summary of Experiments and Experimental Variables.	41
A.1.	Summary of the Experimental Variables Which Correspond to the Boiling Curves.	84

A	surface area
a	constant
Bo	Bond number based on radius (Eq. 26)
Bo _D	Bond number based on diameter, 4Bo
b	constant
C	constant, also specific heat of tantalum in Eq. 53
C _p	specific heat of vapor
C _{p,l}	specific heat of liquid
D	diameter
d	constant, also diameter in Eq. 5
e	constant
f(R)	function of radius, Eq. 8
G(Bo)	optimization function (Ref. 22)
G _R	function defined by $p(p_g - p)gd^3\mu^2$
Gr	Grashof number
g	acceleration of gravity
G _C	gravitational constant in Newton law of motion
h	heat-transfer coefficient
h _C	convective heat-transfer coefficient
h _f , h _{fb}	saturated film boiling, heat-transfer coefficient
h _r , h _{rad}	radiative heat-transfer coefficient
h _t , h _{tot}	total, subcooled film, heat-transfer coefficient
K	thermal conductivity, parameter in Eqs. 56 and 57
L	characteristic length
Nu	Nusselt number
Nu*	modified Nusselt number (Eq. 29)
P	total pressure
P _d	pressure in vapor dome
P ₀	environmental pressure
Pr	Prandtl number
Q, q	heat-transfer rate
q"	heat flux
R	radius
R _d	radius of vapor dome
R ₀	radius of sphere
Ra	Rayleigh number GrPr
Ra*	modified Rayleigh number, $Ra \lambda^*/C_p \Delta T_s$
r	radial coordinate
s	entropy
T	temperature
T ₀	initial sphere temperature
T _b	bulk temperature of liquid
T _f	vapor-film temperature ($T_w + T_s/2$)
T _{fg}	liquid-film temperature ($T_s + T_b/2$)
T _s , T _{sat}	saturation temperature of liquid
T _w	wall temperature

NOMENCLATURE

ΔT_s , ΔT_{sat}	wall superheat, $T_w - T_s$
ΔT_{sc}	liquid subcooling, $T_s - T_b$
t	time
V _r	radial velocity component
V _θ	theta velocity component
α	thermal diffusivity of vapor
α _l	thermal diffusivity of liquid
ε	emissivity
β _g	coefficient of thermal expansion of liquid
Y	coefficient of dome size, see Fig. 2
$\bar{\Delta}$	average dome thickness, see Fig. 2
δ	gap thickness, see Fig. 2
θ	angular coordinate, also dimensionless time, Eq. 42
θ*	angular coordinate at vapor dome
Δθ	time step for data reduction, Eq. 48
λ	latent heat of vaporization
λ*	modified latent heat of vaporization ($\lambda + 0.5C_p \Delta T_s$)
λ _c	critical wavelength, $2\pi\sqrt{gR_d/(P_g - P)}$
λ _{md}	most dangerous wavelength, $\sqrt{30}\lambda_c$
μ	viscosity of vapor
μ _g	viscosity of liquid
ν	kinematic viscosity, μ/ρ
ν _g	kinematic viscosity of liquid
ρ	density of vapor, dimensionless radial coordinate, Eq. 42
ρ _g	density of liquid
ΔP	spatial step in data reduction, Eq. 46
σ	surface tension, also Stephan-Boltzmann constant, Eq. 35
φ	angular coordinate, see Fig. 2

Subscripts

c, crit	critical, also cylinder
D	diameter
d	diameter or dome
f, fb	saturated film
L	characteristic length
l	liquid
max	maximum
md	most dangerous
min	minimum
opt	optimum
rad	radiation
r	radial direction
θ	angular direction

Superscripts

— average value

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ABSTRACT

Transient, natural-convection pool boiling from spheres to subcooled sodium was studied. Hot tantalum spheres were submerged in sodium, and the surface temperature of the sphere was recorded, together with the pressure pulses that developed due to vapor growth and collapse. The experimental data were reduced by numerically solving the heat-conduction equation in the sphere, the end result being the boiling curves of sodium. The range of variables investigated was:

Sodium temperature:	392 to 1607°F
Sphere temperature:	2785 to 4281°F
Sphere diameters:	1.0, 0.75, and 0.50 in.
Sodium depth:	3.0 and 4.5 in.
Pressure:	atmospheric

This investigation showed that sodium subcooling has a large effect on the transient-boiling curve. The initial sphere temperature did not have an appreciable effect on the boiling curve as long as the initial regime was film boiling. An effect of changing the sphere diameter was observed only in the film boiling region.

The experimental data in the film boiling region are correlated by

$$h_t = h_{fb} + 0.88h_r + Kh_c(\Delta T_{sc}/\Delta T_{sat}),$$

where K depends on both the degree of subcooling and the sphere diameter.

In the transition region, violent interaction occurred. The degree of violence reached a maximum at sodium temperature in the range from 1320 to 1570°F. Pressure pulses as high as 5.7 atm were measured 12 in. below the sphere.

Nucleate-boiling data are presented in the transient-boiling curves of sodium at various experimental conditions.

I. INTRODUCTION

In analyzing the safety of fast reactors, one must consider two features in which they differ from thermal reactors, the short prompt-neutron lifetime and the higher fuel enrichment. This second feature increases the chance that a supercritical condition could be achieved by rearrangement of the fuel into a more compact geometry.

There is a definite possibility for a positive reactivity coefficient due to sodium voiding in a sodium-cooled fast reactor. The voiding may occur as a result of a loss-of-coolant accident in which stagnant sodium is caused to boil in a coolant channel. The short prompt-neutron lifetime can cause the reactor to go on a rapid excursion, possibly faster than the response of the available feedback control system. The energy released may result in a meltdown of one or more of the fuel pins, with possible destructive interaction between the molten fuel and liquid sodium.

The destruction may cause a rearrangement of the core materials into a more compact geometry, providing a supercritical condition, which could result in further damage to the reactor core.

The safety analysis of the LMFBR, therefore, requires a better understanding of the energy-transfer process associated with the contact of hot fuel particles and the liquid-metal coolant, that follows a fuel-failure accident.

In the present investigation the mechanism of transient boiling-heat transfer associated with the interaction of hot metal particles with liquid sodium is studied. The experimental conditions (particle temperature, size of particles, depth of coolant, and coolant temperature) that may result in violent interaction were also studied.

A tantalum sphere was induction-heated to a high temperature. The hot sphere was rapidly immersed into a pool of subcooled sodium and kept submerged at a predetermined depth within the pool. The natural-convection, transient-boiling behavior on the sphere surface was observed. This was done through temperature, pressure, and vibration measurements while the hot sphere, at rest, passed through the various boiling regimes. The experimental procedure and instrumentation were tested earlier in preliminary experiments using stainless steel spheres and water.

Sphere diameters of 1.0, 0.75, and 0.50 in. were investigated. The spheres were heated to a range of temperatures between 2785 to 4281°F and dropped into sodium at a range of temperatures between 392 to 1607°F. The sodium levels at which the spheres were placed were 4.5 and 3.0 in.

Theoretical modeling of film boiling was conducted, and a correlation for the heat-transfer coefficient for this regime was proposed. This correlation accounts for the effect of heat transfer by conduction, convection,

and radiation across the vapor film, as well as convective heat transfer to the subcooled liquid. The experimental data are shown to be reasonably well represented by the proposed correlation.

The experimental results show that sodium subcooling has a very important effect on the transient-boiling curve. By varying the sodium temperature, essentially any initial boiling regime can be attained on the sphere surface over a wide range of initial sphere temperatures. Furthermore, at a certain degree of subcooling, film boiling did not occur.

Violent interaction, which was characterized by large pressure pulses and sodium expulsion, occurred for a certain range of sodium subcooling. In this range, a maximum for the degree of violence, as measured by the magnitude of the largest pressure pulse, was obtained.

Useful experimental techniques are presented to characterize the various boiling regimes. In particular, the onset of transition boiling has been accurately determined using the pressure-pulse records obtained during the study.

II. LITERATURE REVIEW

This chapter is not intended as a complete survey of all the literature reported about the subject of boiling-heat transfer. Only those investigations that are considered relevant to the present work are reviewed. This is a review, therefore, of literature pertinent to film, transition, and nucleate boiling, together with the regions of minimum and maximum heat fluxes. Special emphasis will be given to liquid metals in general, and to liquid sodium in particular. The effect of subcooling on various regions of the boiling curve will also be considered.

Investigation of the film-boiling region for metallic fluids is very limited. An extensive survey is made of the work done on nonmetallic fluids. Various models are presented for different geometries.

The minimum heat flux and the region of transition boiling have received very little attention in literature, as indicated by the small number of reported investigations. The critical heat flux has been studied more widely because of its importance in the operation and safety of nuclear reactors and boilers. Substantial work has been directed toward the critical heat flux in liquid metals.

The region of nucleate boiling is certainly the most thoroughly studied of the boiling-curve regions. No attempt is made to cover the theoretical work involved. The fact that nucleate-boiling heat transfer is influenced strongly by the surface condition of the heater makes it less attractive from the point of view of a generalized study of the boiling behavior of a particular fluid. Nevertheless, only experiments and correlations made for this region with liquid sodium are reviewed.

A. Film Boiling

1. Vertical Surfaces

Bromely¹ did his early work on the film boiling from vertical surfaces. The vapor formed by evaporation was assumed to flow laminarly upward under the influence of the buoyancy force. He assumed that the interface was smooth and that the vapor velocity was controlled by either a slip or a nonslip boundary condition. The equation obtained for the film-boiling coefficient was of the form

$$Nu_L = C_1 Ra_L^{*1/4}, \quad (1)$$

where Nu_L and Ra_L^* are the Nusselt and the modified Raleigh numbers based on the characteristic length L of the surface. The effect of radiation on the total coefficient was then included:

$$h_{tot} = h_{fb} + (3/4)h_r. \quad (2)$$

Equation 1 predicted film boiling for the short lower portion of the surface fairly well. Other investigators²⁻⁶ used the same laminar-flow approach with more refinement to take into account convective effects.

Tachibana⁴ also studied the subcooling effect. He concluded that the heat-transfer rate increases markedly with the degree of subcooling. Sparrow and Cess³ considered the effect of subcooled liquid on laminar film boiling. The result of their investigation indicated that for very large subcooling the heat-transfer rate is essentially equal to that for pure free convection (no boiling). They confirmed Tachibana's findings that the surface heat flux increases with subcooling.

Frederking and Hopenfeld⁷ treated the laminar two-phase boundary layers in the subcooled problem. They derived simple equations for the two limiting cases of saturated and highly subcooled film boiling.

Hsu and Westwater^{8,9} noted the presence of a wave profile at the interface which invalidates the basic assumption of laminar flow of the vapor. They proposed a model assuming the onset of a turbulent film instability at a critical Reynolds number. This approach was modified by other investigators,¹⁰ and it appears that the waves on the interface are an important factor in analyzing film boiling.

2. Horizontal Surfaces

Film boiling from horizontal surfaces was studied by Berenson,¹¹ who assumed that Taylor instability causes breakup of waves in a cell pattern. The vapor generated is considered to flow laminarly into vapor domes which are spaced at a distance equivalent to the most dangerous wavelength λ_{md} . His final equation was of the form

$$Nu_L = C_2 Ra_L^{*1/4}, \quad (3)$$

where L is a characteristic length proportional to λ_{md} . A similar equation was derived by Hamill and Baumeister,¹² who showed by a maximization principle that λ_{md} is the optimal spacing between domes for the maximum heat-transfer coefficient. It is worthwhile to mention that they derived the constant in Eq. 3 from purely theoretical considerations. Ruckenstein¹³ also analyzed the same problem. He used Taylor's instability to determine bubble spacing and the frequency of bubble departure.

All the above investigations neglected the contribution of radiation in deriving expressions for film-boiling coefficients. Weighted radiation coefficients were then added to calculate a total heat-transfer coefficient. Hamill and Baumeister¹⁴ considered the simultaneous effect of conduction, convection, and radiation on the vapor film, together with the effect of subcooling of the liquid film. They obtained a general graphical solution for

the problem and a particular one for the range of interest. One of their important conclusions is that beyond a certain degree of subcooling, film boiling is impossible. This is in agreement with Sparrow and Cess,³ who suggested that for very large subcooling, natural convection is the only mechanism for heat transfer. Although the last statement precludes the presence of boiling at large degrees of subcooling, Hamill and Baumeister did not rule out the possibility of another form of boiling replacing the film-boiling mechanism. No data have been presented, so far, to confirm the above predictions, possibly because of the limited range of subcooling for the commonly used fluids.

3. Horizontal Cylinders

Bromely¹ studied this problem and derived an equation similar to Eq. 1, with the characteristic length being the tube diameter:

$$Nu_d = C_1 Ra_d^{*1/4}. \quad (4)$$

Laminar flow was assumed and the buoyancy force balanced the viscous force. Breen and Westwater¹⁵ suggested that Eq. 4 was not valid for very small tubes, where the surface force is important, or for very large tubes, where turbulence might set in. They proposed a semiempirical equation which fits all cylinder diameters:

$$Nu_L = [C_3(L/d) + C_4](Ra_L^*/C_5 L)^{1/4}. \quad (5)$$

Baumeister and Hamill¹⁶ derived an equation based on the same functional groupings. Their equation was in good agreement with the data supplied by Breen and Westwater.

Subcooled film boiling was investigated by Siviour and Ede.¹⁷ They assumed that the heat is lost from the surface by two completely independent heat-transport mechanisms. In the vapor layer, the problem is equivalent to the saturated film-boiling case, while in the subcooled liquid heat is transferred by natural convection. This approach is an oversimplification of the problem since it ignores the coupling of the two mechanisms at the liquid-vapor interface.

4. Spheres

Film boiling from spheres has been studied by several investigators.¹⁸⁻²³ Frederking and Clark¹⁹ derived an expression similar to that of Bromely (Eq. 4). But they found that their experimental data could be correlated by

$$Nu_d = C_6 Ra_d^{*1/3}. \quad (6)$$

Hendricks and Baumeister²³ observed that the above expression does not predict the film-boiling data for small spheres. They analyzed film boiling from small and large spheres, and were able to correlate the available data (see Eq. 34).

Subcooled film boiling of water from spheres was observed by Bradfield²⁴ and Walford.²⁵ Jacobson and Shair²⁶ observed film boiling from a 1/2-in. steel sphere to flowing subcooled water with no net vapor formation.

Except for mercury, there has been little data for liquid metals in the film boiling regime. Lyon²⁷ made an extensive study of boiling heat transfer to liquid metals. He experienced film boiling only for pure mercury and cadmium, and attributed this to their nonwetting characteristics. Bonilla²⁸ and Lin²⁹ studied the pool boiling of mercury with and without wetting agents.

Poppendiek³⁰ carried out analytical and experimental studies of the vaporization of potassium droplets suspended over a hot surface by a vapor film. He concluded that the process of quiet, stable film boiling of potassium drops was the same as that of water and some organic fluids.

A study of stable, pool film boiling from an alkali metal was reported by Padilla and Balzhiser.³¹ In this investigation, potassium was charged to a heater plate at 1600°F, and the film-boiling data were collected at pressures between 2 and 300 mm Hg. The data were correlated with the expression

$$\text{Nu}_L = C_7 \text{Ra}_L^{*1/4} \quad (7)$$

The expression is similar to those used to correlate data from nonmetallic fluids. This indicates that film boiling in alkali metals might not be too dissimilar from that observed in common fluids.

Witte^{32,33} studied the heat transfer from 1/2-in. tantalum spheres to liquid sodium in forced convection. With sphere temperatures of 3500°F and a sodium temperature of 572°F he was not able to detect film boiling. He attributed this behavior to liquid superheat and concluded that the sphere had undergone single-phase, forced-convection cooling. However, he had no experimental means to conclude that other forms of boiling did not actually take place. It is possible, according to the predictions of Hamill and Baumeister,¹⁴ that his experiments were conducted at a subcooling level beyond that required for sustaining film boiling.

To the best knowledge of the author, no data have been presented in the literature for pool film boiling from sodium at atmospheric pressure.

B. The Minimum Heat Flux

Berenson¹¹ modified an approach by Zuber³⁴ to calculate the minimum heat flux on a flat surface. Berenson used the concept of two-phase Taylor instability and extended his analysis to include the minimum temperature difference. Padilla and Balzhiser³¹ found that their data did not agree with Berenson's expressions.

Lienhard and Sun³⁵ showed that it is possible to relate the minimum heat flux on a cylindrical configuration, $(q''_{\min})_{\text{cyl}}$, to that for the flat plate developed by Berenson by the expression

$$(q''_{\min})_{\text{cyl}} = K(q''_{\min})_{\text{plate}} f(R), \quad (8)$$

where $f(R)$ includes the effect of cylinder radius and the liquid properties.

C. Transition Boiling

There has been relatively little investigation of the transition boiling region, a possible reason being a lack of practical interest in this particular region. The difficulty of sustaining this type of boiling on electrically heated test sections is probably another reason.

Berenson³⁶ studied this region extensively. He concluded that transition boiling is a combination of unstable film boiling and unstable nucleate boiling, each of which alternately exists at a given location of the heating surface. Berenson demonstrated that the surface finish of his test section affected the boiling curve in the transition region. This substantiated his assumption that solid-liquid contact exists during transition boiling.

Bankoff and Mehra³⁷ proposed a quenching theory for transition boiling. They assumed that substantially all of the heat transfer occurs by periodic quenching of the solid surface when directly contacted by the liquid. During the short time of liquid-solid contact, they assumed a linear flow of heat in an infinite-composite solid. In this manner, they were able to obtain an expression of the heat flux to the liquid per contact. A modified model was proposed by Aoki and Welty.³⁸ They suggested both quenching and nucleate boiling were responsible for the heat transferred during transition boiling.

D. The Critical Heat Flux

Expressions for the critical heat flux were derived by Kutateladze,³⁹ and by Zuber and Tribus,³⁴ in which q''_{crit} was a function only of the physical properties of the liquid and vapor. They made the assumptions that the total energy supplied by the heater is used to produce vapor at the heater surface and that none of the energy is convected away from the heater to produce vapor

elsewhere. This assumption is correct at low pressures, and it was suggested that an extra term be added to account for the convective effects. Correlations for the critical heat flux in pool boiling of sodium were given by Caswell and Balzhiser,⁴¹ Subbotin *et al.*,⁴² and by Noyes and Lurie.⁴³ The three correlations included the effect of pressure and were all in good agreement. Caswell and Balzhiser⁴¹ proposed an empirical correlation, based only on the physical properties of liquid and vapor, which is applicable to all liquid metals.

The critical heat flux for sodium seems to be dependent on the nature of the heating surface. The data provided by Subbotin *et al.*⁴² show that a variation of 30% or more in the critical heat flux for sodium occurs between a stainless steel and a nickel-based alloy surface.

The effect of subcooling was investigated by Ivey and Morris^{44,45} who presented data for water on stainless steel tubes. They confirmed previous predictions that a linear relationship exists between the critical heat flux and the degree of subcooling.

E. Nucleate Boiling of Liquid Metals

The heat-transfer rate to boiling sodium was measured at atmospheric pressure by Lyon, Foust, and Katz²⁷ using a cylindrical heater. Carnack and Forster⁴⁶ showed that these rates could be represented by the equation derived by Forster and Zuber.⁴⁷

Noyes⁴⁸ conducted experiments at pressures of 1 to 8 psia using cylindrical heaters. He detected no effect due to pressure. His correlation of heat flux and surface superheat does not include pressure as a parameter. In later experiments,^{42,49,50} a pressure effect was found which caused the wall superheat to increase at constant heat flux as the pressure was reduced.

Marto and Rohsenow⁵¹ used a flat surface heater to boil sodium. They noticed that the system pressure, the surface finish of the heater, and the gas content of sodium had a considerable influence on the results obtained. They also observed large temperature fluctuations on the heater surface. The same observation was made by Ivashkevitch⁴² and is consistent with the known instabilities of nucleation activities. However, Ivashkevitch found only a slight pressure effect.

Petukhov *et al.*⁵² used electron bombardment to heat their cylindrical surface. The effect of pressure was included in their correlation. Borishanskii *et al.*⁵³ correlated their data and those of Noyes⁴⁸ and Lyon⁴⁵ for pressures close to atmospheric. They recommended a general correlation taking into account the effect of pressure.

Collier and Kosky⁴⁰ examined the various correlations of the data for nucleate boiling of sodium. They found considerable differences between

the results of various workers. These differences were not as large as expected in view of the different geometries, gas content, surface materials, and finishes used. They concluded that the scattering of the data and of the correlations are no greater than observed with nonmetallic fluids. The Forster-Zuber equation⁴⁷ was recommended for estimating the nucleate-boiling curve with a tolerance of $\pm 10^\circ\text{F}$.

The amount of liquid superheat required to nucleate the liquid alkali metals has been presented by several investigators.^{27,54-56} There is considerable disagreement in these data (superheats varied between 5 and 820°C). Singer and Holtz⁵⁷ conducted several studies to explain this discrepancy. They showed or postulated that there are several factors that affect the incipient-boiling liquid superheat. Among the important ones are: the pressure-temperature history of the heating surface, the liquid velocity, the system pressure, heat flux, liquid purity, dissolved gas content, surface conditions, nuclear radiation, heating method, and the length of time of operation. In a recent study, an electromagnetic pump was used to pressurize sodium in order to eliminate the presence of an inert gas. In this manner, Holtz⁵⁸ was able to demonstrate that the partial pressure of inert gas in the surface cavities and the system pressure are key parameters which influence the incipient-boiling superheats in sodium.

III. THEORETICAL ANALYSIS

A. Introduction

The present and other studies²² indicate that the problem of quenching single hot bodies in cold liquids is by no means simple. If a solid sphere is heated and dropped into a liquid maintained at the saturation temperature, boiling may occur at the sphere surface. The temperature to which the sphere is originally heated is an important factor in determining whether or not boiling will take place and what type of boiling is to occur first.

Fig. 1. A Typical Boiling Curve

The boiling curve (see Fig. 1) shows the various regions encountered in studying the boiling operation and the terminology that is often used to describe them. It is customary to arrive at the boiling curve by a steady-state procedure. The temperature of the surface adjacent to a liquid is increased in steps. At each step, the resultant heat flux is measured and a boiling curve, similar to the one shown, is determined. In a quenching experiment, the surface is at the high tem-

perature originally. As cooling occurs, the same boiling regions are traversed in the opposite direction. Although the general characteristics of the boiling curve are independent of the method of approach, it has been shown⁵⁹ that the two are not equivalent. This is true, at least, for the transition and nucleate boiling regions. The time span encountered during transient boiling in these two regions is rather small. The assumption of a quasi-steady-state boiling is therefore not a realistic one.

In the film-boiling region, however, transient-boiling data may well represent those taken in steady-state experiments.⁵⁹ Heat-transfer rates are usually low and surface cooling is relatively slow. The time scales involved are of the order of seconds (as will be shown in the present study with both liquid sodium and water). The assumption of a quasi-stationary boiling in the film region is justified. The quenching experiments can thus be utilized to generate film-boiling data by conducting relatively simple experiments.

Film boiling will result if the sphere is heated to a temperature above the "Leidenfrost point," then immersed in the liquid. The sphere surface is completely blanketed by a vapor film with no liquid-solid contact. As the surface temperature drops below the Leidenfrost point, a vapor film cannot be sustained by the rate of vapor formation, and rapid intermittent solid-liquid contact takes place. The chaotic nature of boiling in this "transition region" is due to the formation of vapor at rates high enough to obstruct the liquid from contacting the surface over a short

period of time, but not at rates high enough to balance the hydrostatic pressure due to the weight of the liquid. This instability causes the vapor blanket to collapse. The liquid rushing to fill in the gap contacts the hot surface, and vapor is rapidly formed again. Vapor accumulates until it blocks the liquid, then collapses, and so on.

Eventually, the surface temperature drops to a value at which no film type of boiling can be maintained, and complete liquid-solid contact is reached. The boiling mechanism now changes to the nucleate type, in which surface cavities act as nucleation sites for the formation of vapor bubbles.

For a sphere heated to a high temperature and then immersed in a pool of saturated liquid, film boiling can be initiated. The vapor moves up in the surrounding vapor film towards the top part of the sphere and is eventually released in the form of large bubbles. The film thickness around the sphere will increase from bottom to top due to the continuous addition of vapor generated at the liquid-vapor interface. This means that the resistance to heat flow will be smallest at the lower part of the sphere and will increase to a maximum at the top. This implies that the heat-transfer rate will be maximum at the lowest point of the sphere. The surface temperature therefore drops fastest at that point, and the eventual breakage of the film would probably start there. While the lower portion of the sphere is undergoing transition-type boiling, the upper part might still be hot enough to sustain stable film boiling. Consequently, when nucleate boiling starts at the lower part, transition and/or film boiling might still exist on the upper parts of the sphere.

The preceding analysis indicates the difficulty of the exact theoretical modeling of transient boiling from a sphere. In addition to the fact that more than one boiling regime can exist on the sphere surface, the inclusion of time as a variable adds to the complexity of the problem. The presence of a subcooled liquid (a liquid at a lower temperature than the boiling point) surrounding the sphere, as is the case in the present study, only makes the theoretical problem less tractable. Convective heat transfer to the subcooled liquid at the vapor-liquid interface enhances the process of heat removal, and the time scale is further reduced. As a first approximation, one assumes that only one boiling regime exists at a time on the sphere surface. In the present study, no attempt will be made to go beyond this first approximation. It is assumed that a quasi-steady state exists during film boiling. This is not unreasonable, as was mentioned earlier, and will allow the use of a steady-state analysis of the film-boiling problem. The transition- and nucleate-boiling regions will not be analyzed in any rigorous mathematical fashion. Heat transfer in these two regions is influenced by several unknowns, such as the pressure-temperature history and the condition of the sphere surface, which are beyond the scope of this study. Existing correlations will be used later for comparison with correlations of the present data in the respective regimes.

B. Film Boiling

1. Introduction

The physical model used for film boiling from a submerged sphere is essentially that used by Berenson¹¹ for flat surfaces, by Hamill and Baumeister for flat surfaces¹² and wires,¹⁶ and by Hendricks and Baumeister for spheres.²²

A mechanism is assumed for film boiling which is based on several experimental observations. The vapor film completely surrounds the sphere with a vapor gap thickness slightly varying over the effective area of the sphere. Near the top, vapor collects in a single dome in the case of small spheres (see Fig. 2), and in multiple domes in the case of large spheres (see Fig. 3). These domes act, in effect, as hydrodynamic sinks where vapor is collected. As the amount of vapor in the dome increases, the dome grows in size and eventually a large bubble of vapor is released. The model used in the present analysis assumes the presence of a time-average configuration in which all the velocity, pressure, and temperature fields are in steady state.

Fig. 2. Single-dome Model for Subcooled-film Boiling from Small Spheres. ANL Neg. No. 900-1003 (top half).

Fig. 3. Multiple-dome Model for Subcooled-film Boiling from Large Spheres. ANL Neg. No. 900-1003 (bottom half).

2. Basic Model and Equations

Consider the single-dome case in Fig. 2. A solid sphere at a uniform surface temperature T_w is immersed in a subcooled liquid at T_b . The surface is maintained at a high enough temperature to sustain a stable film. Heat is transferred from the sphere surface to the vapor-liquid interface by conduction and convection through the vapor. Radiative interchange between the solid and liquid surfaces are assumed to take place with no absorption in the vapor (transparent medium). The heat reaching the

vapor-liquid interface is partly used to form vapor and partly convected to the subcooled liquid. The proportion depending mainly upon the degree of subcooling of the liquid. The liquid evaporated at the interface flows around the sphere to the vapor reservoir (dome). The reservoir begins at the point designated $(R_0 + \delta, \theta^*)$ in Fig. 2. The single-dome model appears to represent film boiling from small spheres.²² However, for large-diameter spheres the single dome appears to give way to multiple domes, such as occur off a horizontal plate facing upward.

The following assumptions are made in developing the single- or multiple-dome models:

1. Rotational and vibrational effects due to the sphere are small.
2. The flow of vapor around the sphere is laminar, and surface capillary waves, while influencing the boundaries, do not significantly alter the heat transfer or the laminar nature of the flow.
3. The vapor-gap thickness is constant.
4. The model has complete symmetry in the ϕ coordinate.
5. Because of the low Reynolds number, the flow of vapor is both laminar and incompressible. The inertia and body forces in the Navier-Stokes equations as applied to the vapor are neglected.
6. The velocity and temperature profiles are in a steady state.
7. At any instance of time, the surface of the sphere is at an average temperature T_w , the interface at T_s , and the bulk of the fluid at T_b .
8. The properties of the vapor are evaluated at the film temperature

$$T_f = \frac{T_w + T_s}{2} \quad (9)$$

9. The convective terms in the energy equation as applied to the vapor are neglected; however, the latent heat of vaporization is corrected to accommodate this assumption. The correction is usually of the form

$$\lambda^* = \lambda + 0.5C_p(T_w - T_s). \quad (10)$$

a. Governing Equations in the Vapor Film. The governing equations in the vapor film are:²²

(1) Momentum

$$0 = -\frac{g_c}{\rho} \frac{\partial P}{\partial r} + \nu \left(\nabla^2 V_r - \frac{2V_r}{r^2} - \frac{2}{r^2} \frac{\partial V_\theta}{\partial \theta} - \frac{2}{r^2} V_\theta \cot \theta \right); \quad (11)$$

$$0 = -\frac{g_c}{\rho} \frac{1}{r} \frac{\partial P}{\partial \theta} + \nu \left(\nabla^2 V_\theta + \frac{2}{r^2} \frac{\partial V_r}{\partial \theta} - \frac{V_\theta}{r^2 \sin^2 \theta} \right); \quad (12)$$

where

$$\nabla^2 = \frac{1}{r^2} \frac{\partial}{\partial r} \left(r^2 \frac{\partial}{\partial r} \right) + \frac{1}{r^2 \sin \theta} \frac{\partial}{\partial \theta} \left(\sin \theta \frac{\partial}{\partial \theta} \right). \quad (13)$$

(2) Energy

$$0 = \frac{\partial}{\partial r} \left(r^2 \frac{\partial T}{\partial r} \right). \quad (14)$$

(3) Continuity

$$0 = \frac{1}{r^2} \frac{\partial}{\partial r} (r^2 V_r) + \frac{1}{r \sin \theta} \frac{\partial}{\partial \theta} (V_\theta \sin \theta). \quad (15)$$

b. Constraints

(1) The boundary conditions are separated into two categories, which represent the single-vapor dome and multiple-vapor dome regions:

Region	Surface	Single-dome Model	Multiple-dome Model
Thin Film	$r = R_0$	$V_r = 0, V_\theta = 0, T = T_w$	(16)
	$\theta < \theta^*$	$V_r = V_r(\delta), V_\theta = 0, T = T_s$	(17)
	$r = R_0 + \delta$		
	$\theta = 0$		
	$R_0 \leq r \leq R_0 + \delta$	$V_\theta = 0$	(18)
	$\theta = \theta^*$		
	$r = R_0 + \delta$	$P = P_{\text{dome}}$	(19)
Vapor Dome	$\theta^* \leq \theta < \pi$	Heat transfer by conduction	Heat transfer similar to film boiling off a horizontal plate facing upwards

In Fig. 2 θ^* is defined as the transition from the thin film to the vapor dome, δ is the vapor thickness, and P_d is the dome pressure. Since δ , θ^* , and $V_r(\delta)$ are unknowns, three additional mathematical constraints are required to make the problem in the vapor film tractable.

(2) An additional constraint requires that the sphere be in static equilibrium. The condition for static equilibrium requires that the average pressure $\bar{P}$ in the vapor at the liquid-vapor interface resulting from the weight of the supported liquid, surface tension, and atmospheric pressure balance the average pressure due to flow. Summation of the forces acting at the interface gives

$$\int_0^{2\pi} \int_0^{\theta^*} P|_{R_0+\delta} (R_0 + \delta)^2 \sin \theta \, d\theta \, d\phi = \bar{P} \int_0^{2\pi} \int_0^{\theta^*} (R_0 + \delta)^2 \sin \theta \, d\theta \, d\phi. \quad (20)$$

(3) The second constraint requires that the rate of entropy production be a maximum. This is based on a thermodynamic hypothesis suggested earlier.¹² It was assumed that there is an optimum size of the dome that maximizes the rate of entropy production (ds/dT). The hot surface is the heat source, and the liquid-vapor interface is the heat sink. If both the wall and the liquid-vapor interface are at constant temperatures, ds/dT will be maximum if the rate of heat transfer is maximum. This maximizes also the total heat-transfer coefficient h_t (see Eq. 54). As θ^* is the parameter which controls the rate of exchange of energy,

$$\frac{\partial h_t}{\partial \theta^*} = 0; \quad \frac{\partial^2 h_t}{\partial \theta^{*2}} < 0. \quad (21)$$

(4) The momentum and energy equations for the vapor as well as those of the liquid can be coupled by an energy balance at the liquid-vapor interface, where $r = R_0 + \delta$:

$$\underbrace{-\rho \lambda V_r(R_0 + \delta, \theta)}_{\text{vapor production}} - \underbrace{\left(K \frac{\partial T}{\partial r}\right)_l}_{\text{heat conducted to liquid}} = \underbrace{\left(-K \frac{\partial T}{\partial r}\right)_v}_{\text{heat conducted through vapor}} + \underbrace{h_{\text{rad}}(T_w - T_s)}_{\text{heat radiated through vapor}}. \quad (22)$$

A set of equations for the liquid has to be written with the respective assumptions and boundary conditions. The completely defined problem is then solved for the total heat-transfer coefficient from the sphere, in relation to the heat lost by conduction, and radiation in the vapor as well as that convected in the liquid.

The mathematical solution of this problem is obviously complicated, and no attempt to solve it was made in the present study.

A solution of the subcooled film-boiling problem on a flat, horizontal surface was obtained over a certain range of parameters.¹⁴ This has the form

$$h_t = h_f + 0.88h_r + 0.12h_c \frac{\Delta T_{sc}}{\Delta T_s} \quad (23)$$

A form similar to Eq. 23 will be adopted in the present study to correlate the measured total heat-transfer coefficient h_t , the saturated film boiling coefficient h_f , radiation coefficient h_r , the convective coefficient in the liquid h_c , and the ratio of liquid subcooling to surface superheat $\Delta T_{sc}/\Delta T_s$.

c. Saturated Film-boiling Coefficient h_f . For film-boiling studies, spheres were divided into two categories.²²

(1) Small Spheres. These are characterized by the single-dome model. The criterion for a sphere to be small is that the circumference of the dome be less than "the most dangerous wavelength" λ_{md} , where

$$R_0 \theta^* \leq \lambda_{md} \quad (24)$$

For the case of a heavy liquid over a lighter fluid, the most dangerous wavelength represents the wavelength of a small sinusoidal unstable disturbance, which has the fastest rate of amplitude growth of all possible unstable disturbances. From hydrodynamic stability theory, λ_{md} is defined as

$$\lambda_{md} = 2 \sqrt{3} \pi R_0 Bo^{-1/2}, \quad (25)$$

where Bo is the Bond number:

$$Bo = \frac{(\rho_l - \rho) g R_0^2}{\sigma g_c} = \frac{\text{buoyancy forces}}{\text{surface-tension forces}} \quad (26)$$

It follows that for small spheres

$$Bo \leq \left(\frac{2\pi \sqrt{3}}{\theta^*} \right)^2 \quad (27)$$

The solution for this problem is of the form²²

$$Nu^* = \frac{1}{4} \left[\frac{-2Ra^* G(Bo)}{3} \right]^{1/4} + \gamma^{-1} (1 + \cos \theta^*) \csc \theta^*, \quad (28)$$

where

$$Nu^* = 2 \left(\frac{h_f R_0}{K} - 1 \right), \quad (29)$$

$$Ra^* = \frac{\rho(\rho_L - \rho) g D^3 \lambda^*}{K \mu \Delta T_s}, \quad (30)$$

γ is a coefficient of dome size shown in Fig. 2, and $G(Bo)$ is a function of the Bond number.

(2) Large Spheres. When the Bond number is greater than that given by Eq. 27, multiple domes replace single domes (see Fig. 3). The expression for the modified Nusselt number, Nu^* , given by Eq. 28, includes the heat transfer from the dome region. Since the final expression (Eq. 28) is a superposition of solutions for the dome region and for the rest of the sphere, it is not expected to hold when multiple domes replace the single-dome model.

It was assumed²² that film coefficient of the multiple-dome region is proportional to that of the flat surface, derived earlier as

$$Nu_L = hL/K = \text{const}(Ra_L^*)^{1/4}, \quad (31)$$

where L replaces D for the Ra^* expression (Eq. 30). The resulting expression for large spheres becomes

$$\frac{Nu^*}{(Ra^* \sqrt{Bo})^{1/4}} = \frac{1}{4} \left[-\frac{2G(Bo)}{3\sqrt{Bo}} + 0.21(1 + \cos \theta^*) \right]. \quad (32)$$

(3) Generalized Solution. The solution for large and small spheres was matched at a value of $R_0 \theta^* / \lambda_c = 1$, where λ_c is a critical wavelength:

$$\lambda_c = \lambda_{md} / \sqrt{3}. \quad (33)$$

The linear combination of the two solutions produced a general solution applicable to both small and large spheres:²³

$$Nu = 2 + \frac{1}{4} \left[-\frac{2Ra^* G(Bo)}{3} \right]^{1/4} + \left[0.177(Ra^* \sqrt{Bo_D})^{1/4} + \csc \theta^* \right] (1 + \cos \theta^*) \quad (34)$$

d. Radiative Heat-transfer Coefficient h_r . The heat-transfer coefficient by radiation is determined based on the assumption of two concentric spheres, the outer sphere being the liquid-sodium interface. This is considered a specular surface, meaning an incident ray striking it is

reflected in a symmetric fashion about the surface normal. This is a reasonable assumption since liquid sodium has a mirrorlike appearance. The coefficient is given⁶⁰ by the following expression, assuming the vapor in the gap to be transparent (nonabsorbent) to thermal radiation:

$$h_r = \frac{\sigma}{\frac{1}{\epsilon_1(T_w)} + \frac{1}{\epsilon_2(T_s)} - 1} \left(\frac{T_w^4 - T_s^4}{T_w - T_s} \right), \quad (35)$$

where ϵ_1 and ϵ_2 are the emissivities of the sphere surface and the liquid sodium at their respective temperatures.

Data of liquid-alkali-metal emissivities are generally scarce. Only two references^{61,62} reported data on liquid-sodium emissivities. In both cases, a thin film of liquid sodium covering surfaces of other metals was used for the measurement.

In the present study, the relationship between emissivity and thermal conductivity⁶³ of metals used is given by Eq. 62.

e. Convective Heat-transfer Coefficient in the Liquid, h_c . In general,

$$Nu = a(GrPr^c)^b. \quad (36)$$

For natural-convection heat transfer in liquid metals,

$$Nu = h_c D / K = d(GrPr^2)^e. \quad (37)$$

IV. EXPERIMENTAL APPARATUS

A. Equipment

In general, the experimental apparatus consisted of:

- (1) an electrically heated pot containing sodium;
- (2) a thermocouple-instrumented test sphere;
- (3) an induction-coil heater;
- (4) a transport system to move and locate the hot sphere to a pre-determined depth;
- (5) a glovebox enclosure to provide an inert-gas (argon) atmosphere for the test apparatus.

There were two arrangements of the apparatus. One was used in the preliminary experiments with water (see Fig. D.1) and the other for experiments with sodium (see Fig. 4). The differences were basically in the transport system and the sphere-thermocouple assembly.

Fig. 4

Schematic Diagram
of the Experimental
Apparatus. ANL Neg.
No. 900-1005.

1. Sodium Pot

The sodium pool, contained in the pot shown in Fig. D.2, was a 12-in.-high, 3-in. stainless steel pipe, closed at the lower end. (All equipment parts in contact with sodium are made of Type 304 stainless steel.) A sliding gate on the top flange of the pot controlled the release of sodium vapor to the glovebox. The gate also actuated the transport mechanism when fully open. It was controlled by a double-acting pneumatic cylinder* using argon gas at 15 psig, admitted through two three-way solenoid valves. Four thermocouple wells were welded to the side of the pot to measure the sodium temperature at various depths and were located 2 in. apart.

A cold trap (see Fig. D.4) was welded to the lower end of the pot through a reducer. It was 6 in. long and made of 1.25-in. pipe. The bottom end of the trap was sealed off by a threaded cap and used to drain the pool. A quartz pressure transducer was mounted in a side arm near the lower end of the trap. A thermocouple to measure the temperature of the transducer face was located on the opposite side. A cooling jacket on the cold trap maintained the temperature of the transducer within set limits. The jacket was cooled with helium or argon gas. Details of the pot, gate, and cold trap are shown in Figs. D.2 and D.3.

The pot was heated by two 12-in.-high, semicylindrical resistance heaters of 230-V, 3200-W capacity. These were suitable for operation up to 1200°C. The heater was insulated radially with a 2.5-in.-thick asbestos cylinder and axially with a 1-in. asbestos plate. It was contained in a cylindrical stainless steel box, of 10-in. ID and 14-in. height. The heater leads came through the bottom of the box. The electrical power was controlled by an autotransformer.

2. Sphere-thermocouple Assembly

The spheres used during the sodium experiments were made of tantalum (supplied by Norton Co.). The diameters of the spheres were 1.00, 0.75, and 0.50 in., supported by tubes of 0.082-in. OD and 0.01-in. wall thickness or of 0.125-in. OD and 0.015-in. wall thickness (see Fig. 5). The thermocouples used were tungsten-5% rhenium/tungsten-26% rhenium with compacted BeO insulator and tantalum sheath. The thermocouple sheaths were of 0.025-, 0.040-, or 0.063-in. OD, respectively. The use of both exposed and grounded hot junctions was attempted. The sphere-thermocouple assembly for one of the combinations is shown in Figs. 6 and 7.

3. Induction Heater

An induction coil was used to heat the sphere. It was made of 1/4-in. square copper tubing. The five-turn coil was shaped cylindrically, being of 1 3/8-in. ID and 2 in. high. The power leads were connected through

*Bimba Manufacturing Company, Type 045-DX.

the walls of the glovebox to a transformer, then to a 15 kW-450 kHz induction generator. The coil was cooled by silicone oil in accordance with safety requirements established for sodium operations. The transformer and power leads are shown in Fig. D.6, and the induction coil in Fig. D.7.

4. Transport System

The transport system for the sodium experiments is shown in Fig. 8. This was basically a pneumatic cylinder,* 15 in. long. The

Fig. 5. The Different Sphere Sizes Used: 1-, 3/4-, and 1/2-in. Diameter

Fig. 6. Cross Section of Sphere-thermocouple Assembly

*Double-end rod, double-acting, Bimba model 090-DXDEH.

Fig. 7. Sphere-thermocouple Assembly Ready for the Experiment

Fig. 8. Transport System for Sodium Experiments

hollow-plunger shaft was of 1/4-in. OD and 3/16-in. ID, and had a maximum stroke of 12 in. The pneumatic cylinder was mounted on a square flange bolted to the top flange of the glovebox. The sphere assembly was attached to the plunger shaft using an adaptor. The wires of the sphere thermocouple were led through the adaptor and ended with a male ceramic connector. The thermocouple extension wires connected to the power amplifier, passed through the walls of the glovebox, and led through the hollow shaft and the adaptor, terminating in a female ceramic connector which was taped to the adaptor. Argon was used as the driving fluid. The operation of the pneumatic cylinder was controlled by three three-way solenoid valves. A mechanical shock developed when the piston of the transport pneumatic cylinder reached the end of the stroke. This was minimized by stopping the supply of the driving gas before the end of the downward stroke. During the remainder of the stroke, the sphere fell under the effect of both inertia and gravity forces. A microswitch was inserted in the transport circuit and mounted on the upper end of the pneumatic cylinder. When the upper end of the plunger shaft was 2 in. from the end of the stroke, the microswitch was actuated, closing the solenoid valve that fed argon and keeping the exhaust valve open. The argon gas pressure in all experiments was 20 psig. It was independently fed to and exhausted directly inside the glovebox.

5. Glovebox

A glovebox was used to provide an inert-gas-atmosphere enclosure. The box, shown in Fig. 9, was 5 x 3 ft at the bottom and 5 x 2 ft

Fig. 9. Dry Box and Part of the Instrumentation Used in the Sodium Experiments

at the top, and was 3 ft above ground. There was a front inclined window for viewing and a top one for lighting. The box had three gloves and a 24-in.-long side port of 12-in. diameter. Some modifications of the box were made to fit the experimental apparatus. These included incorporation of a stainless steel box on the bottom to accommodate the cold trap and a welded flange on the top to adapt a 36-in.-high Lucite box for use with the transport system. Two extra gloves were installed on the front window. These, in addition to the existing three gloves, facilitated the handling of the experimental apparatus. All electrical connections were made through hermetically sealed receptacles. Gases were admitted and exhausted through modified double ports on the walls of the

box. The power leads of the induction coil passed through a flange bolted to the back wall of the box.

B. Instrumentation

Parts of the experimental apparatus used for measurement or control of the experimental variables will be considered briefly in the following paragraphs.

1. Thermocouples

a. Sphere-surface Temperature Measurement. The thermocouples in the tantalum spheres used for the sodium experiments were W-5% Re/W-26% Re,* sheathed with tantalum and insulated with compacted BeO. All thermocouples used had grounded hot junctions. The sheath diameters were 0.030, 0.040, or 0.063 in.

b. Sodium-pool-temperature Measurement and Control. The thermocouples used to measure and control the sodium temperature were Chromel-Alumel.* The thermocouple sheaths were stainless steel, of 1/8-in. OD. The thermostatic controller (1200°C range) regulated the power input to the sodium heater.

2. Transducers

a. Sodium-pool Pressure Measurements. The pressure pulses due to vapor formation and collapse on the sphere surface were measured using a quartz piezoelectric pressure transducer (Kistler Model 606L), mounted on the cold leg, below the cooling jacket. This had a range of 0-30 psi and a maximum pressure of 300 psi. The sensitivity was not affected at operating temperatures up to 400°F.

b. Argon-atmosphere Pressure Measurement. A second Kistler pressure transducer (Model 606L) was suspended 16 in. over the pool to measure the pressure pulses during violent boiling in some of the sodium experiments.

c. Sonic Detectors. During the water experiments, a sonic transducer was used to detect the boiling sound. This was a phonograph-crystal pickup, cartridge, and a sapphire needle, thermally insulated and enclosed in a box. The transducer was mounted on a stainless steel plate securely attached to the upper flange of the pot.

Due to higher pool temperatures in the sodium experiments, the sonic detector did not function satisfactorily. It was replaced by a

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*Manufactured by Continental Sensing, Inc.

Kistler quartz piezoelectric force transducer (Model 933A), bolted to the lower side of the plate. The other end of the transducer hung freely. The force transducer acted in effect as an accelerometer and functioned successfully in subsequent experiments.

3. Bias Circuit

The zero level of the signal from the sphere thermocouple to the power amplifier was controlled by a bias circuit, the adjustable range of which was 0-50 mV. This allowed a maximum gain on the amplifier of 1000 to be utilized.

4. Charge Amplifiers

The output signals of all the transducers used were amplified using Kistler (Model 503) charge amplifiers. The amplified signal was fed to the power amplifiers.

5. Power Amplifiers

The output of the sphere thermocouple was amplified using a DANA (Model 3400) power amplifier, with a maximum gain of 1000. For all the other signals, Honeywell (Model Accudata 117) amplifiers were used.

6. Data-recording Oscillograph

Temperature and pressure data were recorded using a Mid-western oscillograph (Model 1210). This had the capacity of simultaneously recording on 50 channels. The recording paper used was Du Pont LINO-WRIT 7B direct-writing paper.

7. Glovebox Controls

a. Pressure Switches. The argon pressure in the glovebox was controlled by two pressure switches. One switch admitted argon to the box when the pressure was lower than 0.6 in. of water and turned on an alarm signal if the pressure was higher than 1.3 in. The other exhausted the gas when the pressure was higher than 0.9 in. and turned on the alarm signal at box pressures lower than 0.3 in.

b. Oxygen Analyzer. The concentration of oxygen in the glovebox atmosphere was determined by a Westinghouse, Model 203 Oxygen Gauge. The concentration was read directly in ppm on the instrument dial. The output was also continuously recorded on a recorder.

c. Water Analyzer. A probe was installed on the wall of the glovebox to measure the vapor pressure of water in the argon atmosphere. The measurements were read on the instrument dial.

V. EXPERIMENTAL PROCEDURES

A. Choice of Experimental Apparatus

In the present study, spheres were chosen to simulate fuel particles. Although the geometrical configuration of fragmented, molten materials may deviate from spherical,⁶⁴ recent fragmentation experiments with molten UO_2 in sodium indicate that resulting fragments have a spherical appearance.⁶⁵ It is reasonable to assume that if fragmentation of molten fuel takes place in the liquid state, the molten fragments will assume a spherical configuration if no forces other than gravity and surface tension are acting on it.⁶⁶ Thus, the sphere bears a close resemblance to particles that would form during molten fuel-coolant interaction.

It is difficult to select a particular sphere size to represent an actual case. Postulated fuel-failure accidents associated with vapor explosions involve a single fuel pin, all the fuel in a subassembly, or even all the fuel in the core.⁶⁷ In fast reactors, large amounts of reactivity are potentially available from any phenomena that moves the fuel into geometrical arrangements of higher average fuel density. This might happen if the fuel material melted and settled with the fuel cladding, or it might result upon escape of molten fuel from the fuel cladding and reassembly into a more dense configuration possibly near the center or bottom of the core.⁶⁸ A suggestion was made⁶⁹ to conduct fuel-coolant interaction experiments in which quantities greater than 1 kg of molten fuel (equivalent to a UO_2 sphere of 2.26-in. diameter) are dumped into sodium, in order to extrapolate present information about vapor explosions to larger quantities of fuel. On the other hand, the mean diameter of particles resulting from fragmentation experiments in which molten UO_2 was dumped into sodium were of the order of 250 μ .⁷⁰ The sphere sizes chosen for the present work (diameters of 1, 3/4, and 1/2 in.), although they are closer to the larger end of the scale, are within the range of previously postulated fuel-failure accidents. From a practical point of view, the choice of relatively large spheres has several advantages. The sphere is easier to instrument, has a large enough heat capacity to sustain boiling for a reasonably long time, and the geometrical error introduced by the presence of the thermocouple is reduced. In addition, the heat conducted away by the thermocouple wires and sheath is reduced.

A sodium-pot diameter of 3 in. was believed reasonable. For such a diameter in comparison with sphere diameters of 1/2, 3/4, and 1 in., the wall effects may be neglected. A larger pool diameter may have been preferred from the viewpoint of wall effects. However, it was necessary to limit the heater and insulation to reasonable dimensions.

The effective pot height of 12 in. gave enough flexibility in changing the sphere depth in sodium, thereby eliminating the need for an unduly long sphere-supporting tube.

The remaining dimensions of the experimental apparatus were generally based on those for the sphere and the sodium pot.

Spheres used in this study were made of tantalum, due to its high melting point ($\sim 5430^{\circ}\text{F}$). Both tungsten and rhenium have higher melting points (~ 6180 and 5750°F , respectively), but are difficult to machine. It was possible to heat the spheres to temperatures up to 4281°F , the limiting factor being the usable range of the thermocouples installed (W-5% Re/W-26% Re).

It would have been desirable to use UO_2 to make the spheres. This would have been one step forward in simulating fuel-coolant interaction. It is not currently possible, however, to manufacture a UO_2 sphere that will keep its configuration on heating and quenching, let alone welding a thermocouple to its surface. Experimental considerations, therefore, dictated the use of metallic spheres.

B. Calibration of Instruments

1. Amplifiers

The data amplifiers used in the present work were dynamically tested for noise level and gain accuracy. A square-wave calibration signal from a Tektronics oscilloscope was applied to the amplifier inputs and the amplified signal measured using the same oscilloscope.

2. Galvanometers

Galvanometers were selected according to their frequency response, maximum deflection, and stability. Calibration was carried out using the appropriate amplifiers. The external resistances (series and shunt) of the galvanometers were adjusted such that the maximum deflection of the galvanometer coincided with the maximum output voltage of the amplifier. The internal calibrator of the charge amplifier was used to calibrate the data amplifier-galvanometer system. The accuracy of the calibration was checked using a digital voltmeter.

3. Pool Pressure Transducer

The Kistler Model 606L pressure transducer and associated amplifiers were calibrated against a Heisse precision, Bourdon-tube pressure gauge. Pressures ranging from 0 to 35 psig were applied. The output signal proved to be linear in accordance with the manufacturers calibration charts.

4. Temperature Controller

The accuracy and operation of the temperature controller and control thermocouple were checked. Sodium temperature was increased

stepwise from 390 to 1607°F in steps. At each step, the pool temperature was measured directly using an external (C-A) thermocouple and the thermocouple calibration supplied by the manufacturer. The directly measured temperature was plotted against the temperature indicated by the controller.

C. Sodium Experiments

The experience gained in the preliminary runs with water (see Appendix D) helped to determine the final setup of the experimental apparatus used in the sodium experiments. These experiments constitute the major part of the present work.

The modifications made earlier (see Appendix D) were satisfactory for preliminary experiments. This is especially true since the glovebox was open and various components accessible for repair or replacement. But the arrangement was not adequate for the sodium experiments. A new arrangement was possible when a new type of pneumatic cylinder* was introduced by the Bimba Company. A different transport system was designed based on that pneumatic cylinder. The new design eliminated most of the problems encountered in the water experiments.

The idea of using a pneumatic system for the transport mechanism was considered earlier. With the pneumatic cylinders available at that time, it was not possible to cope with the thermocouple extension wires. It is necessary that the extension wires travel with the transport system. If they hang below the pneumatic cylinder, they may interfere and fall into the induction coil, causing arcing (and damage to the wires). The new type of pneumatic cylinder has a hollow shaft which can adequately accommodate the extension wires.

Pneumatic cylinders have certain advantages in a transport system: (a) they can be adapted for automatic operation; (b) the speed of travel of their plunger can be controlled by adjusting the pressure of the driving gas; (c) the size of the inlet and outlet openings of the gas can add to the control of transport speed and rate of deceleration; (d) a pneumatic cylinder does not need added alignment or travel guides. Thus simplicity and flexibility can be combined in using a pneumatic transport system.

Mechanical impact was virtually eliminated in the following manner. As argon gas was admitted to the upper end of the pneumatic cylinder, the sphere was transported downwards. At some time during its travel, the driving force on the transport system was stopped. However, the sphere continued to travel at lower speed. This significantly reduced the final impact when the sphere stopped.

A bleed valve was installed on the exhaust end of a pneumatic cylinder. A microswitch was mounted on the upper end of the cylinder such

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*Bimba Model 090-DXDEH, double end-rod, double acting.

that the height above the cylinder was adjustable. The flow of gas to the cylinder was controlled by a three-way solenoid valve. This solenoid valve closed if the microswitch was activated. A circular plate was mounted at the end of the upper shaft of the pneumatic cylinder (the cylinder had a double-end shaft). During the downward stroke of the shaft, the circular plate actuated the microswitch, stopping the argon supply and opening the cylinder inlet to the atmosphere. This occurred at a height determined by the vertical position of the switch. The sphere continued to move under the influence of gravity and inertia forces. The final mechanical impact was practically eliminated by manipulating several variables, such as the driving-gas pressure, the valve opening on the exhaust end, and the position at which the microswitch is actuated.

The transport speed of the sphere was slightly compromised in this manner, particularly in the last 2 in. of travel. The average speed of transport in the argon atmosphere was 2.2 ft/sec. Except for the last 2 in., the sphere traveled in sodium at 2.45 ft/sec. During the last 2 in., the average speed was 1.24 ft/sec.

1. Variables Sought

A total of 43 experiments were conducted. The data obtained in 23 experiments were satisfactory in terms of stability of thermocouple output. Sphere-thermocouple failure was the main reason for not using the data obtained in the rest of the runs.

Several variables were investigated: the sodium temperature, the initial sphere temperature, depth of the sphere in sodium, and the size of the sphere. The sodium temperature was varied between 392 and 1607°F ($T_{\text{sat}} = 1618.6^\circ\text{F}$). The initial temperature of the sphere ranged from 2785 to 4281°F. The level of sodium above the center of the sphere, at rest, was 4.5 in. in most of the experiments. Sodium levels of 3.5 and 3.0 in. were also investigated. The spheres used were 1, 3/4, or 1/2 in. in diameter.

The sodium used was of CP grade. It was replaced once during the experiments.

Emphasis was given to the effect of sodium temperature. Experience with water indicated that this variable is rather important in determining the boiling-heat-transfer mechanism.

Six experiments were duplicated to check reproducibility of and confidence in the obtained data. The effect of the initial sphere temperature was investigated in four experiments, and the depth of the sphere in sodium was varied in two runs. All but six experiments were conducted using the 1-in.-dia sphere. The variables used and the experiments made are summarized in Table I.

TABLE I Summary of Experiments and Experimental Variables

Sodium Temperature, °F	Initial Sphere Temperature, °F							
	2700 to 2900	2900 to 3100	3100 to 3300	3300 to 3500	3500 to 3700	3700 to 3900	3900 to 4100	4100 to 4300
392			x					
752						x	x	
932								x
1112								x
1292						x		
1381								z
1392								xx
1544								xxx
1562								xx††x
1571								z
1580							x	z
1589							xx*	x†
1598							x	xtxxxx**
1607	x			x	x	x	xt††	xxz

x runs made with 1-in spheres and 4.5-in depth in sodium

z runs which did not produce satisfactory temperature trace

* runs at depth = 3.0 in

** runs at depth = 3.5 in

† 0.75-in spheres

†† 0.50-in spheres

2. Experimental Procedure

The sodium in the pot was maintained in the liquid state at all the times. The temperature was adjusted so that the sodium does not solidify in the cold leg. This procedure is known to reduce the wear and corrosion on the pressure transducer. A temperature of 392°F (200°C) is suitable for this purpose.

Prior to conducting any experiment, certain variables were adjusted such as the sodium level, gas pressures, oscillograph, and amplifier settings. The oxygen and water contents of the argon atmosphere were checked and corrected. (A detailed step-by-step procedure is given in Appendix B). In those experiments in which sodium was heated to a high temperature (near saturation), the heating process was slow. For example, 60 min were used to heat the sodium to a steady-state temperature of 1470°F. It is important to know the length of time required to heat sodium to a certain temperature, so that, the sphere is not heated for an undue period of time. During heating, the temperature in the cold trap increased.

However, the temperature at the face of the pool pressure transducer must be kept below 400°F. Argon or helium gas was circulated through

the cooling jacket for this purpose. The rate of flow of the gas depended on the temperature in the cold leg, which was monitored by the thermocouple opposite the face of the transducer.

After setting all the experimental conditions to their desired values, the sphere was positioned at the center of the heating coil, being kept in place by closing the exhaust solenoid valve. The sphere was heated by slowly increasing the power input to the induction generator. During an experiment, the heating of the sphere was programmed in steps to maintain a uniform temperature throughout, while increasing that temperature. The final temperature of the sphere was to be reached after the sodium attained a steady-state temperature. The output of the sphere thermocouple was monitored on a DVM (digital voltmeter). When the final temperature was reached, the generator power was turned off. The thermocouple output was switched to the power amplifier and the pilot switch actuated. This actuated the oscillograph transport mechanism and a time-delay relay. Shortly after the paper reached steady speed (approx 0.6 sec), the time-delay relay switched on the transport solenoid valve. The gate on the top of the pot was opened and the sphere transported into the sodium pot. The solenoid valve was closed when the sphere was 2 in. above its final position. The sphere moved down at a slower speed until the end of the transport stroke. The ejection of the recording paper was stopped automatically if a predetermined length had been set. If, however, the length of the cooling period was not known, the oscillograph transport system did not stop until the pilot switch button was released. The sphere was raised up at the end of the experiment and the gate closed. Cooling continued until the sodium pool reached 392°F. The sphere, which was usually tarnished, was removed from the box for inspection. After cleaning and polishing, the sphere was reinstalled in the box, ready for another experiment.

3. Difficulties and Remedies

Many difficulties were faced during the course of the sodium experiments. This was expected due to the wide range of the variables investigated, especially the sodium and sphere temperatures. Only those difficulties of significance will be discussed.

The majority of problems were connected with the sphere-thermocouple assembly. At the early stages of design, one of the main concerns was to locate the hot junction of the sphere thermocouple as close as possible to the surface. This shortens the response time and allows for an accurate interpretation of the thermocouple output. In order to minimize the disturbance of the heat-flow pattern inside the sphere, the diameter of the thermocouple sheath has to be relatively small. The above two conditions were met by choosing a thermocouple with an exposed hot junction and a sheath of 0.030-in. diameter. Many attempts were made to electron-beam weld the hot junction to the surface. The thermocouple wires (~0.005 in. in diameter) broke every time. After repeated failure, this type of thermocouple was changed to one with a sheathed, grounded hot junction. This

meant some compromise of the distance of the thermocouple junction from the surface. For the data-reduction method used, this distance is important for the calculation of the temperature distribution in the sphere. It is therefore necessary to control this distance and keep it as small as practicable.

The manufacturer (Continental Sensing) specifies this distance to be within the dimensions of the sheath radius. X-radiographs showed that this distance varied between half to twice the sheath diameter (see Fig. 10). By use of the radiographs it was possible to measure the distance accurately and to make the necessary adjustments in the subsequent welding operation. During electron-beam welding, some tantalum was added to seal the sphere-thermocouple hole completely. The sphere was subsequently polished to remove the extra material.

Fig. 10. X-radiograph of a 0.040-in.-dia Thermocouple. ANL Neg. No. 116-1111.

Although the positions of the thermocouple wires were known within about 0.010 in., the calculation used required a better accuracy. This was done by taking radiographs of the sphere section containing the thermocouple.

Each tantalum sphere was used until it failed, due to thermocouple, support tube, or surface damage. The sphere was then cut in three pieces with two parallel cuts. The center disc, about 1/8 in. thick, contained the thermocouple. (Efforts to cut thinner discs or to grind or end-mill the 1/8-in. disc resulted in damage of the brittle thermocouple weld.) The disc was then cast in a mold of synthetic polymer to protect the fragile thermocouple tip at the surface and the spherical configuration. A metallurgical-polishing wheel was then used to grind the disc to reduce its thickness. This was necessary since tantalum is extremely opaque to X-rays. Any reduction in the thickness improved the resolution of the obtained radiographs of the disc and thermocouple. Gamma radiography was used since it gives clearer indication of edges than X-rays (see Figs. 11 and 12).

The supporting tubes used originally, of 0.082-in. OD and 0.062-in. ID, did not withstand the thermal shock resulting from quenching. Breakage took place invariably near the position of welding to the sphere. Larger size tubes, of 0.125-in. OD and 0.095-in. ID, were stronger but eventually fractured the same position. The method of manufacturing the

spheres was changed. This allowed a 1/4-in.-long tube (neck) to be an integral part of the sphere (see Figs. 13 and 14). The neck was drilled to fit the supporting tube, which was crimped in position. A very thin weld was made at the end of the neck for the purpose of sealing. This design proved quite satisfactory.

Fig. 11. X-radiograph of a Used Sphere, Showing the Thermocouple. ANL Neg. No. 116-1112.

Fig. 12. Gamma Radiograph of the Sphere of Fig. 11, Showing Sharper Boundaries. ANL Neg. No. 116-1113.

Fig. 13. Old Sphere Design. ANL Neg. No. 116-1114.

Fig. 14. New Sphere Design, Showing Extension Neck

In spite of the improvements of the sphere-thermocouple assembly, the thermocouples were very short-lived, due mainly to the failure of the wires to withstand the thermal stresses resulting from repeated heating and cooling. Heavier wires were used. This required use of thermocouples with larger sheath diameters of 0.040 in. in most runs and of 0.063 in. in few experiments.

Another problem resulted from working with highly subcooled sodium. The severity of the thermal shock in these experiments resulted in surface damage of the sphere (see Fig. 15). This problem was solved

Fig. 15. Damaged Sphere Surface (initial sphere temperature = 4200°F; sodium temperature = 1292°F)

partly by annealing the sphere during heating at 2200°F for 30 min. Annealing in this manner removed the mechanical stresses gained during the machining and drilling operations. If the sphere has been previously quenched, the induced thermal stresses were also reduced. On the other hand, annealing enhanced grain growth. This was not desirable since grain growth reduced the strength of the surface of the sphere and, hence, its resistance to damage due to thermal shock. The manufacturer of the tantalum rod from which the spheres were made was requested to supply rods which had undergone the maximum possible cold work, thus reducing the grain size of the metallic structure. This was done by cold drawing from bars having much larger diameters. In addition, fewer experi-

ments were conducted with the highly subcooled sodium. The combination resulted in a better performance of the sphere, and surface damage was no longer encountered.

Some of the problems resulted from the heated sodium, especially at temperatures near saturation. As the sodium temperature increases, its vapor pressure increases. Thus, more sodium vapor would escape from the top of the pot. Were this vapor to get into the heating coil while the sphere was heated, the gap between the sphere and coil would become conductive. If an electric arc were formed, the coil and the sphere surface could be damaged. A fan was installed to blow argon over the cover.

The gas carried away the sodium vapor before it reached the coil. This was satisfactory only at lower sphere and sodium temperatures. The problem was finally solved by physically separating the sphere from the coil. A quartz tube, 3 in. long, was inserted in the coil. Enough clearance was left so that the sphere did not touch the tube. In this manner, it was possible to carry out all the experiments with no arcing.

The sodium temperature in the cold leg had to be kept below 400°F at all times. This was accomplished by circulating argon through the cooling jacket at rates depending on the sodium temperature. At pool temperatures higher than 1300°F, it was not possible to cool the cold leg at the desired temperature with argon at a pressure of 50 psi. It was not feasible to use higher pressures because of design considerations. Helium gas proved to be a more satisfactory coolant and was used in all subsequent runs.

In the majority of the experiments, it was desired that the sodium level be maintained constant. After heating the sodium to its steady-state temperature, the sodium level was adjusted; this was not difficult until temperatures higher than 900°F are reached. At 1100°F, the sodium vapor made it difficult to locate the surface visually. When the temperature was higher than 1500°F, the sodium vapor pressure made it undesirable to open the gate for any length of time. For these temperatures, the amount of expansion of liquid sodium was calculated, and the sodium level was adjusted at 392°F to give the required value at the final temperature. The calculations were checked at the lower temperatures using a level measure made for this purpose and were found to hold closely.

The condensation of sodium vapor on the lower side of the gate and in the slide slots constituted a difficulty. This was reduced by insulating the upper side of the gate with a 1/4-in.-thick plate of transite (asbestos cement). Further, the design of the gate transport system was simplified to allow for the easy removal of the gate for cleaning after each experiment. The sliding gate plate was originally made of a 1/16-in.-thick stainless steel plate, which slid into a matching frame also made of stainless steel. After prolonged operation, it was difficult to slide the similar metals past each other; the contact surfaces were scratched and had to be polished. The sliding plate was replaced by one made of 1/32-in. tantalum.

In order to detect the boiling sound, a phonograph pickup cartridge and needle were originally used. This detector was sensitive to vibrations caused by the surrounding machinery. The effect of the vibrations was reduced by inserting rubber gaskets under the legs of the glovebox. Sections of the copper tubing connecting the box to the purification system were replaced by pieces of reinforced rubber hose.

Prolonged heating periods of the sodium pot had the effect of decreasing the sensitivity of the cartridge. Use of thermal insulation did not improve operation, and the cartridge was finally discarded. This was

replaced by a force transducer mounted as an accelerometer. The transducer proved satisfactory for high-temperature operation.

Control of the oxygen and water-vapor contents of the argon atmosphere was extremely important to prevent reaction with the sodium and tantalum. The oxygen analyzers originally used were of the "Thermox Probe One" model. This type failed repeatedly, especially on prolonged operation. In spite of simple design, the heaters, temperature controls, and chemical cells were quite unreliable. These were finally replaced with a Westinghouse, Model 203, oxygen gauge, which operated satisfactorily.

It is interesting to mention that although during the experiments sphere diameters were varied from 1.0 to $1/2$ in., it was not necessary, however, to change the size of the induction coil of $1\frac{3}{8}$ -in. ID in order to heat the differently sized spheres. The loss in coupling between the sphere and the coil was compensated for by the decrease in heat lost by radiation as the sphere surface area decreased.

VI. DATA REDUCTION

A. Discussion

The data obtained in a typical experiment consisted of a record of: the temperature variation at the sphere surface, measured by the thermocouple; the pressure pulses due to vapor formation and collapse, measured by the pressure transducer contacting the sodium; and the sonic vibrations, recorded by the sonic transducer. All three measurements were recorded optically on direct-printing paper together with timing marks. Since the main concern in this investigation was the heat-transfer process at the surface of the sphere, the temperature measurement was of particular importance. Furthermore, it is this variable that required detailed analysis in order to obtain meaningful information that can be interpreted easily.

The purpose of the data reduction was to transfer the temperature-time data into surface-heat flux/time data. One approach is to determine that heat content of the sphere as a function of time. Short time intervals are chosen and the heat lost during each period is calculated. Once this is known, it is possible to determine the average rate of heat loss from the sphere over that period.

To determine the heat content of the sphere, the temperature distribution inside the sphere has to be known. This is obtained by solving the time-dependent equation for heat conduction in the sphere:

$$\frac{\partial T}{\partial t} = \alpha \left(\frac{\partial^2 T}{\partial r^2} + \frac{2}{r} \frac{\partial T}{\partial r} \right), \quad (38)$$

with the appropriate initial and boundary conditions:

$$t \leq 0: T = T_0, \quad r \leq R_0 \quad (39)$$

$$t > 0: T = T_w, \quad r = R_0 \quad (40)$$

$$t \geq 0: \frac{\partial T}{\partial r} = 0, \quad r = 0 \quad (41)$$

using the dimensionless time and radius

$$\theta = \alpha t / R_0^2; \quad \rho = r / R_0.$$

Equation 38 becomes

$$\frac{\partial T}{\partial \theta} = \frac{1}{\rho^2} \frac{\partial}{\partial \rho} \left(\rho^2 \frac{\partial T}{\partial \rho} \right) \quad (42)$$

with boundary conditions

$$\theta \leq 0: T = T_0, \quad \rho \leq 1 \quad (43)$$

$$\theta > 0: T = T_w, \quad \rho = 1 \quad (44)$$

$$\theta \geq 0: \frac{\partial T}{\partial \rho} = 0, \quad \rho = 0 \quad (45)$$

Equation 42 was solved using a finite-difference procedure suggested in Ref. 71. The sphere was divided into $(M-1)$ annuli of equal width. This constitutes M nodes, as shown in Fig. 16. The following finite-difference approximations to the partial derivatives were used:

For $i = 1$,

$$\frac{1}{\rho^2} \frac{\partial}{\partial \rho} \left(\rho^2 \frac{\partial T}{\partial \rho} \right) = \frac{6}{\Delta \rho^2} (T_{2,j} - T_{1,j}); \quad (46)$$

For $i > 1$,

$$\frac{1}{\rho^2} \frac{\partial}{\partial \rho} \left(\rho^2 \frac{\partial T}{\partial \rho} \right) = \frac{1}{(i-1) \Delta \rho^2} [i T_{i+1,j} - 2(i-1) T_{i,j} + (i-2) T_{i-1,j}]; \quad (47)$$

For $i \geq 1$,

$$\frac{\partial T}{\partial \theta} = \frac{T_{i,j+1} - T_{i,j}}{\Delta \theta}. \quad (48)$$

Here i refers to position-step number and j to time-step number; $i = 1$ refers to the center and $i = M$ refers to the surface of the sphere.

Fig. 16

Radial and Time Steps for
the Difference Equations.
ANL Neg. No. 900-999.

The set of equations used for computation was: for $i = 1$, using Eqs. 46 and 48 with boundary condition 45:

$$T_{1,j+1} = \left(6 \frac{\Delta\theta}{\Delta\rho^2}\right) (T_{2,j} - T_{1,j}) + T_{1,j}; \quad (49)$$

for $2 \leq i \leq (M-1)$,

$$T_{i,j+1} = T_{i,j} + \frac{\Delta\theta}{\Delta\rho^2} \left[\left(\frac{i}{i-1}\right) T_{i+1,j} - 2T_{i,j} + \left(\frac{i-2}{i-1}\right) T_{i-1,j} \right]; \quad (50)$$

for $i = M$, boundary condition 44 becomes

$$T_{M,j} = T_w; \quad (51)$$

for $j = 1$, the initial condition 43 becomes

$$T_{i,1} = T_0. \quad (52)$$

Once the time step $\Delta\theta$ and the spatial step $\Delta\rho$ were specified, the problem was completely formulated (Eqs. 49-52), and computation of the spatial temperature distribution at each time step achieved.

From the temperature distribution, the heat content of each annulus was computed. The summation of the heat contents of the $(M-1)$ annuli gave the average heat content of the sphere at time θ . The difference between the heat contents computed between times θ and $\theta + \Delta\theta$ gave the heat lost from the sphere during $\Delta\theta$. The time assigned to this step is taken as $\theta + \Delta\theta/2$ and the surface temperature is taken as $[T_w(\theta) + T_w(\theta + \Delta\theta)]/2$.

The problem was programmed in FORTRAN IV for use with the CDC-6400 computer. Details of the program "SPHERE" and the assumptions made are given in Appendix C.

The surface temperature was obtained as a function of time, using the Model 400 data scaler chart reader, to read the points off the temperature-time trace. Because of the high degree of accuracy, the points could be read directly as volts-output from the amplifier. From knowledge of the amplifier gain and the thermocouple bias, the thermocouple output was computed. The sphere-surface temperature was obtained using the manufacturer's calibration data for the thermocouple.

The time interval between two data points is not necessarily constant. The interval depends on the rate of change of temperature. When the sphere-surface temperature is changing slowly, as in the film-boiling region, long time intervals are used. In the nucleate-boiling region, on

the other hand, the temperature changes rapidly, and relatively short time intervals are chosen. In either case, it is necessary that the temperature be linear between any two points. This is extremely important for the accuracy of the computer program used, since it allows for linear interpolation to small temperature changes between the measured data points.

Because of the large number of points that have to be read for each experiment, a FORTRAN IV program "WRETC" was written that will accept the voltage output of the amplifier together with the amplification and bias, and prints as an output the surface-temperature data. This is done by interpolation through linear portions of the thermocouple calibration curve. This program is given in detail in Appendix C. Finally, the calculated heat flux from program "SPHERE" together with a time-averaged surface temperature were used as an input to another program, "GRAPH," which plots the logarithm of the heat flux q/A versus the logarithm of the surface superheat ($T_w - T_s$) using the CALCOMP subroutine. A typical temperature trace and its final CALCOMP plot are shown in Figs. 17 and 18, respectively.

B. Accuracy of the Data-reduction Method

In any data-reduction technique, several probable sources of inaccuracy arise. The present method involved approximating a partial differential equation by a difference equation. Errors were due mainly to truncation of the higher-order partial derivatives of temperature with respect to both space and time. Formulae for the order of magnitude of errors can be easily derived, but not easily evaluated because of the unknown values of the higher partial derivatives.

Fig. 17. Typical Oscillograph Record Showing the Various Boiling Regimes, Run 36. ANL Neg. No. 900-1000.

Fig. 18 The Boiling Curve for Run 36 as Determined by the Data-reduction Method

A method was used to evaluate the accuracy of the computed heat fluxes, based on the FORTRAN program "SPHERE" actually used to compute these fluxes. It was assumed that the truncation errors changed as the time steps and/or the spatial steps were changed. Theoretically speaking, the shorter these steps, the better is the accuracy, since this will simulate more closely the partial derivatives. Nevertheless, this is not an unlimited option, and the value of $(\Delta\theta/\Delta\rho^2)$ has to be smaller than a certain limit for the computer program to be stable. This actually limits the choice of $\Delta\rho$ at a given $\Delta\theta$. The time step is easier to change. Because of its presence in the numerator, it can be made as small as is desired. This is also restricted in the sense that interpretation of the temperature-time data becomes inaccurate at very small time intervals.

Bearing this in mind, the temperature-time trace in a typical experiment was divided into intervals in which the temperature changed linearly with time. These linear portions were then divided into smaller time intervals by interpolation such that the temperature drop during each subdivision was approximately 0.3°F . This close following of the temperature ensured accuracy and uniformity. Furthermore, it was independent of the manner by which the temperature changed with time. The sphere was normally divided into 11 nodes (10 annuli), and in six cases into nine nodes (8 annuli). Eleven nodes were considered accurate enough, as will be discussed below, and the choice of nine nodes for some runs was mainly to account for the thermocouple position. This will be discussed in Sec. VII.

The stability of the computer program with respect to $\Delta\rho$ was checked. The time step was chosen in accordance with the condition mentioned above. The number of nodes was increased from 5 to 21 in two-node steps. The computer program became unstable at values higher than 15 nodes. The value of $\Delta\theta/\Delta\rho^2$ at 15 was 0.278 and at 17 was increased to 0.363. It was thus assumed that $\Delta\theta/\Delta\rho^2$ should be less than 0.3 for stability.

The value of the heat flux was checked at three corresponding points. One point was chosen at the peak heat flux, one at the beginning of the cooling process, and a third point arbitrarily chosen somewhere in the middle of the data points. The difference between successive tests decreased as the number of nodes increased (which is what would be expected). The maximum difference between 13 and 15 nodes was 0.2%, and that between nine and 15 was 1%. Thus, the estimated truncation error due to the choice of the spatial step should be less than 1.5%.

Next, the time step $\Delta\theta$ was changed from the current value $(\Delta\theta)^*$ to $5(\Delta\theta)^*$ in five steps, keeping the number of nodes at 11. The change in heat-flux values was found to be less than 0.01% when $(\Delta\theta)^*$ increased to $2(\Delta\theta)^*$. At $3(\Delta\theta)^*$ and higher, $\Delta\theta/\Delta\rho^2$ became larger than 0.3, and the computer program became unstable. Thus, the value of $(\Delta\theta)^*$ could be doubled with no appreciable loss of accuracy. The truncation error in this interval can be assumed negligible.

The effect of changing both $\Delta\theta$ and $\Delta\rho$ was determined, keeping $\Delta\theta/\Delta\rho^2$ constant. When the spatial increment $\Delta\rho$ was doubled and $\Delta\theta$ quadrupled, the maximum difference was found to be 1.5% at the peak heat flux.

The maximum error due to the data-reduction method can be estimated at $\pm 1.5\%$ with an average value of approximately $\pm 1\%$. This will be combined with other possible sources of error in the next chapter, and a confidence limit will be established for the accuracy of the computed heat fluxes.

VII. RESULTS AND DISCUSSION

In this chapter, major experimental results will be discussed as well as the factors which are believed to influence them significantly. One section is devoted to the definition and estimation of the sources of error that affect the computed results. This is important, because it builds confidence in the experimental and computed results. It also provides a means of identifying and separating areas of uncertainty due to uncontrollable effects. This is enhanced by the section discussing reproducibility of the results of "identical" experiments.

After discussing the accuracy and reproducibility of results, an analysis of these results is conducted. The effect of the experimental variables on the boiling curve was analyzed with the help of individual or combinations of the boiling curves of several experimental runs.*

The individual regimes of the boiling curve (Fig. 1) are then discussed with particular emphasis on the effect of subcooling on the respective regions. Because of the strong dependence of the nucleate boiling region on surface condition and several other factors which are strictly controlled in the present study, this region will not be discussed. However, the experimental results, as shown in the boiling curves,* may be of interest for further study of this region, with the understanding that the inherent scatter makes these results only approximate.

A. General

1. Accuracy of the Experimental Results

Several sources of error are inherent in deriving the final value of the heat flux. Although an accurate estimate of the known contributing factors is very difficult, an effort was made to get a reasonable bound on the inaccuracies due to such factors as:

- a. thermocouple-calibration data;
- b. the uncertainty in determining the exact thermocouple position in the sphere;
- c. the electronic circuitry, such as the bias circuit, the amplifier, and the recording instrumentation;
- d. calibration of the recorder;
- e. uncertainty in the "data scale" chart reader;
- f. truncation error in the difference equation;
- g. the effect of temperature on the physical properties of tantalum.

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*A compilation of these boiling curves may be examined in copies of the thesis filed with Northwestern University and at University Microfilms, Ann Arbor, Mich.

The calibration data for the W-5%-Re/W-26% Re thermocouple supplied by the manufacturer were used; these are reported to be within $\pm 1\%$ accuracy. The uncertainty in determining the exact position of the thermocouple wires with respect to the sphere surface was estimated to be ± 0.002 in. To determine its effect on the computed heat flux, a typical set of computations was made while changing the thermocouple position in the computer program "SPHERE" in steps of ± 0.002 in. The maximum error would occur at the peak heat flux. This was found to be $\pm 1\%$ between any two positions 0.002 in. apart.

The error due to electronics was estimated to be less than $\pm 1\%$ and that due to calibration $\pm 1\%$. The chart reader had an accuracy of $\pm 0.05\%$ of full scale, that is, an average of $\pm 0.1\%$. The reproducibility of reading one data point was $\pm 0.4\%$ on the average. The full scale, in most of the experiments, represented 10 mV, giving a maximum error of $\pm 0.2\%$. In five runs, the full scale represented 33 mV and in two runs 100 mV; this produced a maximum error of $\pm 0.7\%$ and $\pm 2.0\%$, respectively. On the average, the error due to reading the data from the charts did not exceed $\pm 0.5\%$.

The truncation error due to the difference equations was shown in Sec. VI to be less than $\pm 1.5\%$. The computer program "SPHERE" (see Appendix C) takes into account the dependence of the specific heat of tantalum on temperature. The functional dependence used was obtained by fitting the specific-heat data⁷² to a second-degree polynomial by a least-squares fit:

$$C(T) = 3.764 \times 10^{-2} - (2.203 \times 10^{-6}) T + (1.080 \times 10^{-9}) T^2 \text{ Btu/lb} \cdot ^\circ\text{F} \quad (53)$$

valid for $2000 \leq T \leq 4000^\circ\text{F}$. The maximum and average deviations were 1% and 0.5%, respectively.

Another factor involving physical properties of tantalum is the thermal diffusivity ($\alpha = K/\rho C$). The physical properties K , ρ , and C change with temperature in such a manner that the variation in α is $\pm 1\%$ in the range of interest (2000-4000°F). The term α appears in the difference equation implicitly in $\Delta\theta/\Delta\rho^2$. The factor $\Delta\theta/\Delta\rho^2$ affects only the term that accounts for the change of temperature over one time step (see Eq. 50). Since this change is less than 1°F , the effect of temperature on α can be neglected. The value of α used in the computation corresponds to that at 3000°F.

An estimate of the total error due to the previously mentioned effects can now be made. The sum of all contributions gives a maximum error of $\pm 7\%$. Allowance may be made for some unusual variations, such as thermocouple-trace oscillation in the transition- and nucleate-boiling regions, and for very rapid temperature variations at high degrees of

subcooling, at which some of the assumptions used in the computations are not strictly valid. Therefore, a ± 10 and $\pm 5\%$ are reasonable estimates of the maximum and average errors in the computed heat fluxes, respectively.

2. Reproducibility

Two sets of experiments were conducted, each at similar conditions of sphere size, sodium temperature, and depth. The initial sphere

Fig. 19. Reproducibility of the Experimental Results, q/A in $Btu/hr-ft^2$ and $T_w - T_{sat}$ in $^\circ F$. ANL Neg. No. 900-1001.

temperature for one set varied by 2.5% for the other. Experimental runs 8 and 9 were conducted in sodium at $752^\circ F$, using 1-in. spheres at a depth of 4.5 in. The initial sphere temperatures were 4039 and $4132^\circ F$, respectively. The plot of the heat flux versus surface superheat (the increase of wall temperature over saturation temperature of sodium) is shown in Fig. 19. In those two runs, only transition- and nucleate-type boiling were achieved. The dotted line at the beginning of run 8 is due to the initial period of entrance of the sphere into sodium. During that period, forced-convection boiling takes place. This is accompanied by higher heat-transfer rates than in the natural-convection-boiling regime. The average scatter of data between the two runs, as indicated by the solid lines, varies from ± 5 to $\pm 16\%$. The latter value

cannot be accounted for by the error analysis previously made. In that analysis, only factors that can be accounted for are discussed.

One important factor that can have a strong effect on both transition- and nucleate-boiling heat transfer is the nature of the heating surface. This is not accounted for in the present investigation. Following each experiment, the sphere was removed from the glovebox and cleaned of adhering sodium. A thin dark film usually existed on the surface. This was polished out with emery paper and the final finish made with alumina powder. Although the surfaces appeared identical, the actual microscopic nature is not known. It has been reported⁷³ that changing the surface finish of a heated 2-in. vertical cylinder changes the pool-boiling heat flux of a n-pentane by an order of magnitude in the regions of transition and nucleate boiling. Furthermore, the surface superheat at the critical heat flux can decrease 500% by changing the surface from "mirror-like" to "Emery 60" finish.

This can be explained by the fact that during both transition and nucleate boiling, liquid-surface contact takes place. The boiling heat transfer in these two regions, therefore, depends on the nature of the surface.

Such magnitudes of variation were not encountered in the present investigation. This can be due to a fair degree of reproducibility of the sphere-surface finish. Yet these variations emphasize the sensitivity of boiling heat transfer in the transition and nucleate regimes to small variations of the surface roughness. The average variation of the critical heat flux in runs 8 and 9 was $\pm 5\%$, and the superheat varied by $\pm 2.5\%$.

The results of runs 24 and 29 are shown in Fig. 19. These were conducted in 1607°F sodium with initial sphere temperatures of 4195°F , other conditions being identical. The maximum scatter in the film region is $\pm 4.5\%$, within the estimated error due to factors other than surface effects. The surface superheats at the Leidenfrost point are within $\pm 1\%$ from their average value. However, the critical heat fluxes have an average scatter of $\pm 30\%$, whereas the superheat scatter is $\pm 25\%$.

It can be concluded that the experimental results were fairly reproducible in those areas of the boiling curve where no dependence on surface finish is expected. The scatter in these areas was found to be within the estimated experimental accuracy. When surface effects are present, the scatter increases with increased dependence of the boiling mechanism on the nature of the surface.

Although an effort was made to reproduce the surface finish, average variations of $\pm 17\%$ in the heat flux and $\pm 15\%$ in the wall superheat were encountered in both the transition and nucleate regions.

3. Effect of Initial Sphere Temperature

The induction heater used to heat the spheres did not have fine enough control to reproduce exactly the initial sphere temperature in two similar experiments. It is, therefore, important to determine the effect of small variations in the initial temperature on the boiling curve.

The effect of more pronounced differences between initial sphere temperatures in separate runs is also important to know. This makes it possible to compare different experiments when variables other than the sphere temperature are also changing.

Six experiments were conducted (runs 24-29) using 1-in. spheres in 1607°F sodium. The initial temperature varied between 4195 and 2785°F .

Results of these experiments are shown in Fig. 20. In spite of the large variation of initial temperature (1410°F), the similarity between

the boiling curves is noticeable. The maximum scatter of the heat flux in the film-boiling region was $\pm 7\%$. This indicates a nearly identical cooling behavior of the sphere in all these experiments, regardless of the initial temperature. The temperature distribution in the sphere at the time step before the Leidenfrost point is shown in Fig. 21 for all but run 28.

Fig. 20

Effect of Changing the Initial Sphere Temperature on the Boiling Curves of Sodium; $T_{Na} = 1607^\circ F$, q/A in $Btu/hr-ft^2$, $T_w - T_{sat}$ in $^\circ F$. ANL Neg. No. 900-996.

Fig. 21
Temperature Distribution in the Sphere near the Leidenfrost Point, $T_{Na} = 1607^\circ F$

The temperature distribution was identical in all cases to within less than $\pm 1\%$. The maximum difference in temperature between any two experiments at the same spatial position was $38^\circ F$, to be compared with an initial difference of $1410^\circ F$. It may be concluded that in the film-boiling region the initial sphere temperature does not influence, to any appreciable extent, the cooling behavior of the hot sphere.

This conclusion is not difficult to explain. As will be seen later, the calculated film-boiling, heat-transfer coefficients were rather small, usually less than $100 \text{ Btu/hr-ft}^2\text{-}^\circ\text{F}$. This implies a relatively slow cooling process [Biot number $(\text{hr/K}) < 0.1$], and the sphere behaves in an almost isothermal fashion.

It can be seen in Fig. 20 that run 28 differs in its boiling-curve characteristics from the other runs. The initial sphere temperature of this run was 2785°F , only 45°F higher than the average surface temperature at the Leidenfrost point (2740°F), where the vapor film is about to break down. The instability of the film caused the initial cooling behavior of the sphere to be different from that induced by the presence of a stable film for an appreciable length of time. It can be seen in Table A.1 that the total cooling in run 28 was much shorter than that for runs 24-27.

The scatter in the data points in the film region is slightly higher than the estimated overall error for this region ($\pm 5\%$). One of the previously unaccounted sources of error in the experiments is a slight uncertainty in the sodium temperature of $\pm 4^\circ\text{F}$. This can introduce an error which will be discussed in the section on the effect of subcooling. In general, the maximum scatter in the transition and nucleate regions is about $\pm 50\%$ with an average scatter of $\pm 25\%$. This is not an unusually large scatter in those areas (as was discussed previously).

The cooling pattern of the hot sphere in run 28 was noticeably different from that of runs 24-29. This is attributed to the fact that the initial sphere temperature in run 28 was in the vicinity of the Leidenfrost point. The sphere, thus, encountered transition boiling immediately. Knowing the chaotic nature of transition and the dependence of the boiling mechanism in the transition- and nucleate-boiling regimes on the nature of the surface, one may deduce that similar hot spheres, having different initial temperatures, will probably have a different boiling behavior when introduced into liquid sodium such that the initial regime is either transition or nucleate boiling.

4. Effect of Sodium Subcooling

The term subcooling refers to the status of a liquid below its saturation point. Subcooling is measured by the difference between the temperature of the liquid and its saturation temperature under the prevailing pressure. The degree of subcooling is high when the liquid temperature is low and is zero at the saturation temperature.

The effect of subcooling on the boiling characteristics of sodium was investigated in the present study (see Table I). Subcooled-sodium temperatures vary from 392 to $1607 \pm 4^\circ\text{F}$. In most of the runs, the initial sphere temperature falls in the range from 4000 to 4300°F .

In those experiments in which stable film boiling was established, it is possible to compare their respective boiling curves. This is justified in Sec. VII.A.3, bearing in mind the limitations assigned to the transition and nucleate regions.

Fig. 22. Effect of Changing the Degree of Subcooling on the Boiling Curves; $D = 1.00$ in., q/A in Btu/hr-ft², and $T_w - T_{sat}$ in °F

The results of several runs are shown in Fig. 22. Run 24 represents the average of runs 24-29, which were conducted at the same sodium temperature. Because the effect of subcooling is the most thoroughly studied variable in this investigation, only the general effects will be discussed. More detailed analysis is left to respective sections.

The sodium temperature in run 24 was $1607 \pm 4^\circ F$, that is, $12^\circ F$ below saturation. The behavior of sodium at this temperature is not expected to differ appreciably from satu-

rated sodium. This run will be taken, therefore, as a basis for comparison. The results of experiments with 3/4- and 1/2-in. spheres are shown in Figs. 23 and 24, respectively.

Fig. 23. Effect of Changing the Degree of Subcooling on the Boiling Curves; $D = 0.75$ in., q/A in Btu/hr-ft², and $T_w - T_{sat}$ in °F

Fig. 24. Effect of Changing the Degree of Subcooling on the Boiling Curves; $D = 0.50$ in.

It can be seen from Fig. 22 that there was a definite shift of the boiling curve toward higher heat fluxes as the degree of subcooling increased. Raising the sodium temperature from 392 to 1607°F had the effect of decreasing the heat flux, in comparable regions, by more than an order of magnitude (4×10^7 to 3×10^6 Btu/hr-ft² at the critical heat flux).

The wall superheat also changed by about the same order. Thus, the boiling curve shifted toward higher heat fluxes and wall superheats by increasing the degree of subcooling. Similar behavior has been observed in the preliminary experiments with water (see Figs. D.9-D.11).

In addition to the mechanism of vapor formation in saturated boiling, convective heat transfer takes place in subcooled boiling. This is attributed to the added $\Delta T_{sc} = T_{sat} - T_{bulk}$, which exists between the saturated liquid-vapor interface and the subcooled liquid.

The preceding statement should be valid whether the boiling regime is nucleate or film. In this respect, transition boiling is expected to follow the same trend since the mechanism previously assumed is that of alternate film- and nucleate-type boiling.

Another factor that can be as (or even more) important than the temperature-induced convection is the convection due to turbulence. It has been observed in experiments with both water and sodium that the degree of the turbulence due to bubble formation and collapse increases markedly as subcooling is increased. This is probably due to the rapid collapse of bubbles forming in a highly subcooled-liquid field followed by an accelerated liquid moving in to occupy the vacancy left by the collapsed bubble. In transition boiling, in which intermittent vapor-film formation takes place, subcooling will shorten the life of the vapor film due to increased rate of condensation at the vapor-liquid interface.

The combination of a large ΔT_{sc} and the accompanying turbulence at the heating surface can account for the observed shift of the boiling curve toward higher heat-transfer rates.

The above two factors can also explain the increase in the surface superheat with the degree of subcooling. In the region of subcooled film boiling, two competing events take place at the vapor-liquid interface. The heat energy that reaches the interface by conduction, convection, and radiation is dissipated by convection into the subcooled liquid and by the formation of vapor which is necessary to sustain the vapor film. It is reasonable to assume that, as a large proportion of the heat reaching the interface is dissipated into the liquid, a larger amount of heat has to be transferred from the surface in order to sustain a stable vapor film. Since all processes of radiation, conduction, and convection are increasing functions of the wall temperature, a higher wall temperature (higher superheat) will be required.

In the nucleate-boiling region, a larger wall temperature is required to heat the subcooled-liquid layer contacting the surface to a temperature where nucleation is possible. In addition, there are the competing processes of the heat transferred to liquid and that required to sustain the minimum necessary vapor pressure inside the bubble.

Since the effect of increased subcooling is to increase wall superheat in both film and nucleate regions, the transition region will be accordingly characterized by an increased wall superheat.

It follows from the preceding argument that for a given wall temperature, as the degree of subcooling is increased, it becomes more difficult to sustain film boiling. The subcooled liquid, acting as a heat sink, condenses a larger proportion of the vapor necessary to keep a certain film thickness. Since the equilibrium film thickness corresponding to a certain wall temperature decreases with subcooling, the vapor film will collapse as soon as the forces (due to the external pressure, liquid heat, and surface tension) acting on it become unbalanced.

In the present investigation, at a wall temperature of approximately 4150°F, film boiling was not observed when the sodium temperature was 1544°F (75°F subcooled). At the next highest sodium temperature used, 1562°F (57°F subcooled), a film-boiling region was obtained. The latter was not initially stable, as indicated by both the pressure and sonic transducers. An initial short period of transition-type boiling was obtained, which changes rapidly into film boiling. A possible explanation is that this unstable period is due to the turbulence created by the sphere entering the liquid and stopping shortly thereafter.

Usually, a limited degree of vibration occurred as the sphere stopped. This was damped rapidly by the liquid, but could momentarily disturb the vapor film, which was already near its instability limit (due to high subcooling).

It is interesting to note that in their development of a theory for subcooled film boiling from horizontal plates, Hamill and Baumeister¹⁴ predicted that beyond a certain value of a certain heat-transfer parameter, it is impossible to sustain film boiling. This parameter includes heat-transfer coefficients due to radiation, saturated film boiling, and natural convection in the liquid, as well as the ratio of liquid subcooling to the wall superheat. They did not report any data to verify their prediction. This point will be discussed further in the section on film boiling.

The above discussion indicates that subcooling has an important effect on the boiling curve. For the same wall superheat, the boiling curve was shifted in the direction of higher heat flux. If the heat flux is kept constant, increased subcoolings cause the wall superheat to increase.

The general effect of increased subcooling, therefore, was to shift the boiling curve toward higher heat fluxes and higher wall superheats, as can be seen in Fig. 22.

An indirect consequence of the above conclusions is that for each wall superheat, there is a critical degree of subcooling which corresponds to a film thickness which is incapable of balancing the forces acting on it. It is reasonable to assume that film boiling cannot be sustained on the surface at a subcooling higher than that critical value. For sodium at atmospheric pressure and a wall superheat of 2530°F, this critical subcooling was found to be between 57 and 75°F.

5. Effect of Depth

The depth of interest in this study was the height of liquid sodium above the center of the immersed sphere. All the runs but one were conducted at a depth of 4.5 ± 0.1 in, using the 1-in. spheres.

Runs 35 and 36 were identical except that the depth was 3 ± 0.1 in. in run 36. Both runs were conducted in 1589°F sodium with initial temperatures of 3980 and 3995°F, respectively. The results are shown in

Fig. 25. Effect of Changing the Depth on the Boiling Curve; $T_{Na} = 1589^\circ\text{F}$, q/A in Btu/hr-ft^2 , and $T_w - T_{sat}$ in $^\circ\text{F}$. ANL Neg. No. 900-898.

Fig. 25. In the film region, the run made at 4.5 in. gave a heat flux higher by 12% than that at 3.0 in. (run 36). This is 7% higher than the expected scatter in this region and may indicate a trend towards higher heat fluxes as the depth is increased.

Pressure has the effect of increasing the degree of subcooling by increasing the saturation temperature, and also possibly reducing the thickness of the vapor film. Thus, pressure can be expected, in a broad sense, to have the same effect on film boiling as that of subcooling. It is unreasonable to expect that a change of 1.5 in. of sodium can have an appreciable effect on the saturation point of sodium or on the thickness of the vapor film.

But whatever little effect this may have, the observed trend of the boiling curve is in the right direction. Except for a small part of the transition region, the remaining portions of the boiling curves for the two runs are

remarkably identical. Knowing the extent of the scatter in the transition and nucleate regions, it must be quite a coincidence to get such reproducibility.

Firm conclusions that may be extended to large depths cannot be reached from the two preceding experiments. However, it can be seen that a small variation of depth will not have an appreciable effect on the boiling curve. This establishes confidence in the present investigation; since the uncertainty in measuring depths is of the order of ± 0.1 in.

6. Effect of Sphere Diameter

The effect of sphere diameter was investigated in runs 24, 38, and 41, conducted under similar conditions. Sphere diameters were 1.0, 0.75, and 0.50 in., respectively. The resulting heat fluxes are shown in Fig. 26.

Fig. 26. Effect of Changing the Sphere Diameter on the Boiling Curve; $T_{Na} = 1607^\circ\text{F}$, q/A in Btu/hr-ft^2 , and $T_w - T_{sat}$ in $^\circ\text{F}$

Runs 24 and 38 showed a close similarity in the film-boiling region. The correlation suggested by Frederking and Clark¹⁹ (Eq. 6) does not have a dependence of the film heat-transfer coefficient on the sphere diameter D . However, their originally derived equation (similar to Eq. 4) does have an inverse dependence as $D^{1/4}$. This means that the heat flux is higher for smaller spheres by a ratio proportional to $(D_{\text{large}}/D_{\text{small}})^{1/4}$.

Hendricks and Baumeister's equation (Eq. 34) also predicts a dependence on the diameter. Simplifying the equations to apply for large or small spheres only,²³ they suggest two expressions. For large spheres,

there is no dependence of the heat-transfer coefficient on the radius; for small spheres the dependence is roughly proportional to $(1/D)^{1/4}$.

The criterion for small and large spheres (see Eq. 27) shows that both the 1- and 3/4-in. spheres are considered large; for the 1/2-in. sphere, $Bo < (2\pi\sqrt{3}/8^*)$, and hence is considered small.² The above argument explains the fact that no measurable difference exists between heat fluxes for the 1- and 3/4-in. spheres. For the 1/2-in. sphere, the minimum heat flux is 13% higher than that for the 1-in. sphere. The ratio of $(D)^{-1/4}$

of the 1/2-in. to that for the 1-in. spheres is 1.19. The difference between 13 and 19% is within the expected limits of accuracy for this region.

The results in this section indicate that the size of the sphere does not appreciably affect the film-boiling heat flux for spheres larger than 3/4-in. diameter. For spheres smaller than 3/4 in., a dependence on the diameter exists. The dependence can be reasonably accounted for by the factor $(1/D)^{1/4}$. This is probably valid only so long as the vapor film thickness is small in comparison with the sphere diameter ($\delta/R \ll 1$). Since the heat flux due to radiation is independent of the diameter, and since the heat flux due to convection in the liquid is proportional to $(1/D)^{1/4}$ (see Eq. 65), the above discussion applied only when the contribution of radiation is small compared to the total rate of heat transfer.

B. The Film-boiling Region (Line AB in Fig. 1)

The film-boiling region is quite characteristic from an experimental point of view. In contrast with nucleate boiling, in which the cooling process usually takes a fraction of a second, film boiling is rather slow. The duration of film boiling is higher by approximately two orders of magnitude than nucleate boiling. Cooling periods of 60-70 sec were frequently encountered in experiments when the subcooling was low ($\approx 12^\circ\text{F}$) and the initial sphere temperature high ($\approx 4200^\circ\text{F}$).

This large difference in the cooling rates is reflected in the appearance of the temperature trace shown in Fig. 17. This shows that when the experimental conditions are such that film, transition, and nucleate boiling take place, it is not difficult to distinguish their respective regions from the relative rate of temperature drop. The pressure trace which appears above the temperature line in Fig. 17 is that registered by the sodium-pool pressure transducer. Again, it can be noticed that the film boiling is the most "quiet" among the three regimes. The pressure pulses that accompany vapor growth and collapse are relatively nonexistent when compared with transition or nucleate boiling.

This relative quiet, combined with a low rate of cooling, makes it easy to identify the film-boiling region. This is believed to be an excellent qualitative tool for investigations involving quenching. It is probably applicable to all fluids since similar behavior was observed in the preliminary experiments with water.

In most of the experiments, the heat-transfer coefficients in the film region did not exceed 100 Btu/hr-ft². The Biot number, defined as hR/K , is therefore of the order 0.1. This makes it possible to consider the sphere as cooling down in an isothermal fashion. The error involved is relatively small, as can be seen in Fig. 21. The assumption of a uniform temperature distribution in the sphere, just before the Leidenfrost point is

reached, gives an average sphere temperature of 2760°F. Since the calculated temperatures at the center and the surface were 2771 and 2738°F, respectively, the error introduced is <0.5% at the center and <1% at the surface. The error in the heat flux is expected to be near the average of these two values.

It is therefore important to identify the film-boiling region whenever a rapid and reasonably accurate estimate of the heat flux is desirable.

In the present investigation, film-boiling heat fluxes were obtained by the data-reduction method described in Ch. VI. The general effect of subcooling on film boiling was an increase in the heat flux, as can be seen in Fig. 19. This was due to the added contribution of the natural-convection heat transfer to subcooled sodium. Because it is a good heat conductor, sodium is expected to be influenced by subcooling to a larger extent than nonmetallic fluids.

The total heat-transfer coefficient in film boiling is given by

$$h_t = \frac{q}{A(T_w - T_{sat})} \quad (54)$$

The heat flux (q/A) is that determined from the heat lost by the sphere. The wall temperature T_w was calculated by the program SPHERE. This was done by volume-weighted linear extrapolation using the temperature at the first node next to the surface ($M-1$) and the temperature measured at the thermocouple position. The saturation temperature of sodium T_{sat} was taken to be 1618.6°F.⁷³

Consider the models shown in Figs. 2 and 3 for natural-convection film boiling from spheres to subcooled liquids. It is clear from the model that heat will be transferred faster across the thin vapor film ($\theta < \theta^*$) than across the thick dome ($\pi > \theta > \theta^*$). It is, therefore, expected that the heat flux across the dome area (single or multiple) will be less than that across the film area.

The data used to obtain heat fluxes in film boiling, as well as other boiling regimes, were derived from a thermocouple placed at the lower end of the sphere. The thermocouple cannot actually "see" the vapor dome on the other end of the sphere. Hence, the thermocouple measurements might not account for the much lower heat-transfer rates at the dome. The heat-flux data obtained using these measurements may thus reflect a higher overall cooling rate of the sphere than is the actual case considering the dome area.

In order to trace this possible inaccuracy, reference is made to Eq. 34. Although this equation was derived for saturated film boiling, it is

assumed valid in the context of the following argument. The last term in the right-hand side of the equation accounts for the heat transfer through the vapor dome. In the film-boiling region, calculations showed that the contribution of the dome area to the total heat transferred from the 1-in. sphere was 10-15%. The area of sphere covered with the dome was approximately one-fifth of the total sphere area. The average heat-transfer coefficient (assuming uniform overall heat transfer) would be 29% lower than that based on the heat-transfer rate at the lower end of the sphere only. When heat transfer by radiation and that by convection in the liquid were taken into account, this percentage became 15.5%.

Although the above argument is basically correct, it does not imply that the thermocouple measurement was 15.5% in error. The reason is that if the actual overall heat flux is lower than that predicted from the thermocouple measurements, the Biot number hr/K will become smaller, and the sphere behave in a more isothermal fashion than was demonstrated earlier (see Fig. 21). The calculation of the heat flux was based solely on the temperature distribution in the sphere. The accuracy of the final result improves as the temperature in the sphere becomes more uniform.

Actually, the sphere might cool down fastest at the lower end. This is because of the added effects of higher heat-transfer rates and because of the assumption that the vapor film around the sphere is uniform in thickness, which certainly is an approximation. In flowing up and around the sphere, the accumulation of vapor would cause the film to increase in thickness from a minimum at the lower end to a maximum at the vapor dome. This causes a faster rate of heat removal from the lower end of the sphere than at any other position. In order to correct for the expected inaccuracy, the experimental heat fluxes were reduced by 10%. This is considered a reasonable compromise between complete acceptance (heat flux should be 15.5% lower) and complete rejection (heat flux unchanged) of the previous considerations.

It must be noted that the preceding arguments are valid only in the film-boiling regime because of the fundamental geometrical differences between the lower and upper parts of the vapor film. Since gravity is not expected to play a comparable role in the other two boiling regimes, the sphere was assumed to behave more uniformly in transition and nucleate boiling than in film boiling.

The factor of 10% has been used only in this section and affects the data plotted in Figs. 27-31. Elsewhere in this work, the boiling curves represent the actual experimental data.

Fig. 27

Correlation of Film-boiling Data for 1-in. Spheres, $\Delta T_{sc} = 11.8^\circ\text{F}$. ANL. Neg. No. 900-1002

Fig. 28. Correlation of Film-boiling Data for 1-in. Spheres, $\Delta T_{sc} = 20.6^\circ\text{F}$. ANL. Neg. No. 900-1006.

Fig. 29. Correlation of Film-boiling Data for 1-in. Spheres, $\Delta T_{sc} = 29.6^\circ\text{F}$. ANL. Neg. No. 900-1008.

Fig. 30. Correlation of Film-boiling Data for 1-in. Spheres, $\Delta T_{sc} = 38.6^\circ\text{F}$. ANL. Neg. No. 900-1009.

Fig. 31. Correlation of Film-boiling Data for 1-in. Spheres, $\Delta T_{sc} = 56.6^\circ\text{F}$. ANL. Neg. No. 900-1007.

The reduced film-boiling data are correlated by

$$h_t = h_{fb} + 0.88h_r + Kh_c(\Delta T_{sc}/\Delta T_{sat}), \quad (55)$$

where

$$K = 17.9/(\Delta T_{sc})^{0.7} \quad \text{for 1-in. spheres,} \quad (56)$$

$$K = 27.5/(\Delta T_{sc})^{0.65} \quad \text{for 3/4- and 1/2-in. spheres,} \quad (57)$$

$$\Delta T_{sc} = T_{sat} - T_b(^{\circ}\text{F}), \quad (58)$$

and

$$\Delta T_{sat} = T_w - T_{sat}(^{\circ}\text{F}). \quad (59)$$

All the parameters in Eq. 55, except possibly $(\Delta T_{sc}/\Delta T_{sat})$, can be immediately seen as affecting the total heat-transfer coefficient. If Eq. 55 is rearranged, the reason for choosing the parameter $(\Delta T_{sc}/\Delta T_{sat})$ can be immediately seen; thus,

$$h_t \Delta T_{sat} = h_{fb} \Delta T_{sat} + 0.88h_r \Delta T_{sat} + Kh_r \Delta T_{sc} \quad (60)$$

or

$$Q_t = Q_{fb} + 0.88Q_r + KQ_c. \quad (61)$$

This in effect states that the total heat transfer from the surface can be accounted for by the weighted sum of the heat transferred by conduction, convection, and radiation across the film, as well as the heat transferred by natural convection to the subcooled liquid.

1. Saturated Film-boiling Heat-transfer Coefficient, h_{fb}

Equation 34 was used to calculate h_{fb} at each data point of the boiling curve in the film region. The functions $G(Bo)$ and θ^* were estimated from Figs. 6 and 8 in Ref. 22. This makes it possible to compare the calculated and experimental values at corresponding temperatures.

Since Eq. 34 contains several physical properties of sodium vapor which are functions of temperature, a correlation for each physical property as a function of temperature was used. These properties were the thermal conductivity, specific heat, viscosity, and density. The correlations and the references are given in Appendix E, together with the functions $G(Bo)$ and θ^* . Also given is a sample calculation of h_{fb} .

2. Radiative Heat-transfer Coefficient, h_r

This coefficient is calculated using Eq. 35 at each data point. The emissivity of tantalum, ϵ_1 , is a function of the wall temperature T_w . A correlation is obtained using data in Ref. 75 and is given in Appendix E.

The emissivity of liquid sodium, ϵ_2 , is somewhat difficult to obtain. The values given in Ref. 61 involve a basic uncertainty. It is reasonable to expect that these values do not represent solely the emissivity of sodium. The presence of a thin film of sodium on the surface of a hot metal makes it almost certain that the emissivity of the metal actually affects the measured value.

Because of the above uncertainties, a correlation that relates the emissivity of liquid metals to their thermal conductivities is used:⁶²

$$\epsilon = 1.0 \times 10^{-4} TK^{-1/2}, \quad (62)$$

where T is in $^{\circ}K$ and K in $W/cm^2 \text{ } ^{\circ}K$. A sample calculation of ϵ_2 and of h_r is given in Appendix E.

3. Natural-convection Heat-transfer Coefficient in the Liquid, h_c

For liquid metals at a horizontal tube, the Nusselt number $h_c D/K$ is given by²⁸

$$Nu = 0.53(Gr Pr^2)^{1/4}, \quad (63)$$

where $D > 0.01$ ft, and $(Gr Pr)^{1/3} < 5400D$ ft; for nonmetallic fluids with the same restrictions,

$$Nu = 0.53(Gr Pr)^{1/4}. \quad (64)$$

For nonmetals at a sphere surface, where $10^{4/3} < (Gr Pr)^{1/3} < 10^3$,

$$Nu = 0.75(Gr Pr)^{1/4}. \quad (65)$$

No correlation is reported in the literature for the case of a sphere in liquid metals. Since the sphere diameters used in the present investigation were such that $(Gr Pr)$ satisfies the validity ranges of Eqs. 63-65, it is assumed that a correlation of the form

$$Nu = 0.75(Gr Pr^2)^{1/4} \quad (66)$$

may be reasonable to apply for the case of a sphere in a liquid metal.

The physical properties of sodium are to be evaluated at the liquid film temperature $\frac{1}{2}(T_{\text{sat}} + T_b)$. In all the experiments involving film boiling, the lowest sodium temperature used was 1562°F, only 57°F below the saturation temperature. This is considered a small difference, and its effect upon the physical properties was neglected in the calculations. All properties of liquid sodium were therefore evaluated at the saturation temperature of 1619°F. A sample calculation of h_c is given in Appendix E.

The total heat transfer coefficient h_t is correlated by Eqs. 55-57, and plotted against the wall superheat in Figs. 27-31. The equation of Hamill and Baumeister, Eq. 23, is also shown for comparison. Equation 23 is a correlation based on the theory developed for film boiling from a horizontal surface.¹⁴ In spite of the fact that it is an approximation of the graphical solution of the problem over a selected range of parameters, the authors predicted that it may turn out to be a general solution for different geometries.

It can be seen from Figs. 27-31 that this is a reasonable prediction except for the value of the weighing factors. Indeed, had the authors¹⁴ chosen a different range of parameters, Eq. 23 would have contained different values of the constants.

In the range of subcooling investigated, for which film boiling of sodium was possible, the coefficient K of the subcooling parameter, $[h_c(\Delta T_{\text{sc}}/\Delta T_{\text{sat}})]$, varied between 1.01 and 5.6. This is much larger than the 0.12 used in Eq. 23. This is due to the fact that K , calculated by Eqs. 56 and 57, correlates the data over the complete range of the film-boiling region rather than a particular part of the range. The effect of subcooling on a liquid having a high thermal conductivity like sodium can be expected to be stronger than its effect on nonmetallic fluids.

Since K is a weighting factor for the subcooling parameter, it emphasizes the relative importance of natural convection in the liquid sodium in comparison with other modes of heat transfer through the vapor film. The experimental points in Figs. 27-31 fall above the line representing Eq. 23. This is because of the larger weighting factor of the subcooling parameter.

As the degree of subcooling increases from 11.6°F in Fig. 27 to 29.6°F in Fig. 30, the correlation represents the data points more closely (the average deviation being $\pm 10\%$ as compared to $\pm 18\%$ for $\Delta T_{\text{sc}} = 11.6^\circ\text{F}$). This is caused by the decreased percentage of uncertainty in ΔT_{sc} as the degree of subcooling increases. It can be seen from Fig. 31 that the data points in run 30 are not well represented by the correlation (55). The temperature of sodium in this experiment was 1562°F. At lower temperatures, film boiling in sodium was not possible. This indicates that the temperature of sodium in run 30 was near the range of unstable film boiling. As a

matter of fact, in both run 30 and run 43 (1/2-in. sphere), which were conducted in 1562°F sodium, the vapor film broke down more than once, giving place to transition boiling. However, it returned back after a short period of time to film boiling. This may explain the peaking of h_c (see Fig. 28), since transition boiling is associated with higher rates of heat transfer than film boiling. For spheres smaller than 1-in. diameter, a separate correlation (Eq. 57) is given for K . The small number of experiments conducted using 3/4- and 1/2-in. spheres does not permit any further conclusions as to the dependence of K on the sphere diameter.

C. The Leidenfrost Point (Point B in Fig. 1)

The Leidenfrost point is the point at which film boiling becomes unstable. The vapor film starts to collapse and intermittent liquid-solid contact takes place. Point B in Fig. 1 marks the onset of the regime of transition boiling. The rate of temperature drop at the surface of the sphere increases and relatively large pressure pulses due to the process of rapid vapor formation and collapse begin to appear.

Fig. 32. Estimated Time to Reach the Leidenfrost Point from Pressure Trace and Final Boiling Curve

The latter property (characteristic, isolated pressure pulses) is considered a reliable measure of the end of the film and start of the transition regime. Two methods are used to determine the point of minimum heat flux. One of them is by recording the first point on the pressure trace where relatively large pulses start to appear, as in Fig. 17. The second method uses the final plot of heat flux versus wall superheat, similar to that in Fig. 18. The estimates made by the two methods are plotted for comparison in Fig. 32. The similarity of the two independent estimates is rather clear. It emphasizes the accuracy

of determining the minimum temperature difference and, hence, minimum heat flux from the recorded temperature-time trace in a transient-boiling experiment.

In the present investigation, the changes of the minimum heat flux $(q/A)_{\min}$ and minimum superheat $\Delta T_{\min}$ with the degree of subcooling ΔT_{sc} were correlated by the following expressions:

$$(q/A)_{\min} = 6.3 \times 10^4 + 1.9 \times 10^3 \Delta T_{sc} \text{ (Btu/hr-ft}^2\text{)} \quad (67)$$

and

$$\Delta T_{\min} = 7.9 \times 10^2 + 12.2 \Delta T_{sc} (^{\circ}\text{F}) \quad (68)$$

The average deviation in $(q/A)_{\min}$ was $\pm 18\%$ and in $\Delta T_{\min}$ $\pm 15\%$.

According to the above expressions, the values at saturation of $(q/A)_{\min}$ is 6.3×10^4 ($\pm 18\%$) Btu/hr-ft², whereas that for $\Delta T_{\min}$ is 790 ($\pm 15\%$) $^{\circ}\text{F}$, both at atmospheric pressure. Several expressions reported in the literature to calculate $(q/A)_{\min}$ were developed from hydrodynamic considerations only. As an example, Lienhard and Sun³⁵ modified the expressions of Zuber⁷⁵ and Berenson³⁶ to arrive at an expression for the minimum heat flux at horizontal cylinders:

$$(q/A)_{\min} = 2.72 \left[\frac{\rho \lambda}{D} \left(2g \frac{\rho_L - \rho}{\rho_L + \rho} + \frac{4\sigma}{(\rho_L + \rho)D^2} \right)^{1/2} \left(g \frac{\rho_L - \rho}{\sigma} + \frac{2}{D_2} \right)^{3/4} \right] \quad (69)$$

This equation gives a value for $(q/A)_{\min}$ for sodium of 1.92×10^3 Btu/hr-ft². Padilla and Balzhiser³¹ measured minimum heat fluxes for potassium of 0.2×10^4 and 0.9×10^4 Btu/hr-ft² at pressures of 2 and 50 mm Hg, respectively, and estimated 1.5×10^4 for the minimum heat flux at 1 atm. They pointed out that both Zuber's and Berenson's equations underestimate the actual values as measured experimentally. Padilla and Balzhiser did not explain how they estimated $(q/A)_{\min}$ at 1 atm. Their two reported values indicate that the rate of increase of $(q/A)_{\min}$ with pressure has an exponent 2.14. This gives a $(q/A)_{\min}$ of 3.2×10^4 Btu/hr-ft², approximately twice their estimated value. In either case, their values are lower than that for sodium as predicted by Eq. 57, but nevertheless are within the same order of magnitude.

The equation developed by Berenson³⁶ for the minimum superheat at a horizontal surface is

$$\Delta T_{\min} = 0.127 \frac{\rho \lambda}{K} \left[\frac{g(\rho_L - \rho)}{\rho_L + \rho} \right]^{2/3} \left[\frac{g_c \sigma}{g(\rho_L - \rho)} \right]^{1/2} \left[\frac{\mu}{g_c(\rho_L - \rho)} \right]^{1/3} \quad (70)$$

This gives a value of 51°F for sodium, which is much lower than the value of 790°F calculated from Eq. 68. The experimental values reported for potassium³¹ at pressures of 2 and 50 mm Hg are 400 and 550°F , respectively. An extrapolation of these two values to 1 atm gives $\Delta T_{\min}$ of 736°F , which is 54°F (7%) lower than the value calculated for sodium by Eq. 68. Equation 70 is also reported to underestimate the value of $\Delta T_{\min}$ for potassium appreciably.⁷⁷

The data used to obtain correlations 67 and 68 were those for the 1-in. sphere. For the 3/4-in. sphere, the correlations predict $(q/A)_{\min}$

and $\Delta T_{\min}$ with an average deviation from the correlations of $\pm 10\%$ and $\pm 4\%$, respectively. In the same order, the predictions for the 1/2-in. sphere are $\pm 20\%$ for $(q/A)_{\min}$ and $\pm 20\%$ for $\Delta T_{\min}$.

It can be concluded that the correlations 67 and 68 predict the minimum heat flux and superheat in sodium with a reasonable deviation for both the 1- and 3/4-in. spheres. The deviation in the case of 1/2-in. spheres, although acceptable for this type of experiment, is somewhat high. No data are available in the published literature for direct comparison. The closest data, those of Padilla and Balzhiser³¹ for potassium on a horizontal plate, are in the same broad range, although all the compared values are extrapolations of the experimental results obtained at pressures much lower than atmospheric.

D. The Transition Region (Line BC in Fig. 1)

The transition boiling region is characterized by its chaotic nature.⁷⁷ In sodium, this is reflected by the relative violence associated with it. The measure of violence in this investigation is taken as the size of the pressure pulses generated during this period as recorded by the pool pressure transducer. To be more specific, the size of the maximum recorded pressure pulse is taken as a measure of violence in the transition region.

A marked effect of the degree of subcooling on the violence of transition boiling was observed. A plot of the degree of subcooling versus the size of the maximum pressure pulse recorded is shown in Fig. 33. The level of violence changed considerably as the degree of subcooling varied between 50 and 300°F. Thus, the most violent range for transition boiling in sodium occurred when sodium temperature ranged from 1570 to 1320°F. The largest interaction occurred when the sodium temperature was 1382°F.

Fig. 33

Effect of the Degree of Subcooling on the Violent Interaction in the Transition Region

The maximum pressure pulse measured was 83.7 psig (5.7 atm) at the transducer position (12 in. below the sphere position).

Appreciable expulsion of sodium from the pot was noticed in the above range of subcooling (50-300°F), usually accompanied by loud noise. The event resembled, in appearance, explosions encountered in experimental dropping of molten UO_2 in subcooled sodium.^{65,78} The condition of the sodium pot following an experiment with 1472°F sodium is shown in Fig. 34.

Fig. 34

Condition of the Sodium Pot following a Violent Interaction, $T_{\text{Na}} = 1472^\circ\text{F}$. ANL Neg. No. 116-1115 No. 6.

The transition region is one of the least explored regions of the boiling curve. More interest has been shown lately in this region because of its possible connection with the safety of nuclear reactors.

One hypothesis was proposed to explain violent fuel-coolant interaction. Witte *et al.*⁷⁹ assume that the vapor explosion observed in connection with molten-material/subcooled-fluid interaction is due to the rapid heat transfer associated with fragmentation. This fragmentation may be caused by the pressure pulses associated with the transition boiling, which overcomes the surface-tension forces holding the molten material intact.

Calculations made using data in Fig. 33 indicated that the force generated at the peak pressure pulse was several times that required to break up molten UO_2 .⁸⁰ This shows that fragmentation would have occurred had the sphere been composed of molten UO_2 .

The pressure-pulse history of molten UO_2 -sodium experiments in which violent interaction took place was examined.⁷⁸ In all the experiments extensive fragmentation was observed. The record of the pressure pulses, however, did not show a primary pressure pulse preceding the first large

pressure event (the primary pressure pulse is that which may occur as a result of violent transition boiling and is supposed to trigger the fragmentation process; this leads eventually to very large pressure pulses). The resolution of the system (the ability to distinguish two consecutive pressure pulses) was estimated to be of the order of 0.2 msec⁷⁸ if a primary pressure pulse actually took place, the subsequent processes of fragmentation, mixing, and pressure buildup must be in the order of 0.2 msec.

Cho and Wright⁸¹ analyzed the mechanism of molten fuel-coolant interaction. They presented two models, both of which assume the absence of any coolant vapor on the surface of the hot molten particles. The calculated rise time for the resulting pressure pulse is of the order of 0.1-0.4 msec.

When noncondensable gases were assumed present, the corresponding times were increased to 0.2-1.8 msec. The presence of any coolant vapor on the surface would further increase these times.

According to the model accepted for the process of fragmentation, mixing, and pressure generation, two possibilities can be assumed. If the time required to accomplish the preceding sequence of events is $\ll 0.2$ msec, it is possible that a triggering pressure pulse occurs which is responsible for the break up of the molten material. This will lead to a pressure rise fast enough to mask the initial pressure event (which will not be detected).

But if it is assumed that the sequence of events leading to the pressurization event requires a time $\gg 0.2$ msec, the initial pressure pulse should have been detected. A definite choice between the above two possibilities would ascertain or deemphasize the role which transition boiling actually plays in the process of initiating fragmentation.

It is implied in the above discussion that fragmentation is the mechanism which leads to explosive or very violent vapor formation. This appears to be fairly well accepted among many investigators working in this area. Therefore, no discussion will be made of the validity of that postulate.

The results in Fig. 33 point toward the fact that there exists an optimum degree of subcooling at which the hot sphere-sodium interaction is maximum. This may be accounted for by the following argument. The characteristic pressure pulses in the transition region are assumed to be due to the combined effects of rapid vapor formation at the surface (fast buildup of pressure) followed by rapid vapor condensation (fast depressurization). At the lowest end of subcooling (namely, at saturation), vapor formation is fastest. Practically all the heat transferred to the liquid at saturation goes into vapor formation. Vapor collapse will be relatively nonexistent, and there would not be a measurable pressure pulse due to this event.

Of the total heat required to raise the liquid enthalpy to that of the vapor at the upper limit of subcooling (the melting point), only 20% is used to heat up the liquid to the saturation temperature. If this is what actually happens, the rate of vapor formation would be 80% of that at saturation.

This is hardly the case, due to the competing effect of heat transfer to the highly subcooled liquid. This heat transfer probably takes place by either a quenching mechanism³⁷ or by the combined effects of quenching and nucleate boiling.³⁸ In either case, since sodium has a very high thermal conductivity, a large proportion of the heat available is transferred rapidly into the liquid.

Consequently, one would expect only a small proportion of the available heat to go into vapor formation. This vapor will condense rapidly because of the high degree of subcooling, but since the vapor buildup is small to start with, no appreciable pressurization or depressurization pulses are expected to occur.

It can be reasonably assumed that neither "very cold" nor "very hot" sodium will be favorable for violent interaction with hot materials. The degree of subcooling which maximizes the sum of the rates of vapor formation and collapse will probably be the optimum for a violent interaction to take place. This is found to be between 50 and 300°F, with a possible maximum at 237°F subcooled sodium.

The discussion in this section indicates that transition boiling in sodium is often violent. The degree of violence is a function of the subcooling. Both low and high degrees of subcooling do not give rise to appreciable pressure pulses. The maximum degree of violence occurs in sodium between temperatures of 1320 and 1570°F.

Violent interaction was repeatedly observed in this range, and pressure pulses up to 5.7 atm were recorded. Although the recorded pressures could well account for the breaking up of molten UO₂ droplets,⁸⁰ there is no evidence that such pressures occurred prior to violent interaction between molten UO₂ and subcooled sodium.⁷⁸

E. The Critical Heat Flux q_{crit}'' (Point C in Fig. 1)

The critical heat flux has been shown to increase with the degree of subcooling (see Fig. 19). The data are correlated by the expression

$$q_{crit}''(sc) = 4.1 \times 10^6 [1 + 7.8 \times 10^{-3} \Delta T_{sc}] (\text{Btu/hr-ft}^2). \quad (71)$$

The average deviation between the correlation and the measurement is 20%, and the maximum 150%. The expression correlates more closely

the data at $\Delta T_{sc} > 100^\circ\text{F}$, with an average deviation of 10% and a maximum of 19%. At saturation ($\Delta T_{sc} = 0$), Eq. 71 gives

$$q''_{\text{crit}}(\text{sat}) = 4.1 \times 10^6. \quad (72)$$

Thus,

$$q''_{\text{crit}}(sc) = q''_{\text{crit}}(\text{sat})(1 + 7.8 \times 10^{-3} \Delta T_{sc}). \quad (73)$$

Most theories predict a linear relationship between the critical heat flux and the degree of subcooling, of the form

$$q''_{\text{crit}}(sc) = q''_{\text{crit}}(\text{sat})(1 + K \Delta T_{sc}). \quad (74)$$

Katateladze and Shneiderman⁸² derived an expression in which

$$K = \left(\frac{\rho_L}{\rho} \right)^{0.923} \left(\frac{C_{pL}}{25\lambda} \right). \quad (75)$$

Ivey and Morris⁴⁴ derived another expression for K:

$$K = 0.1 \left(\frac{\rho_L}{\rho} \right)^{0.75} \left(\frac{C_{pL}}{\lambda} \right). \quad (76)$$

For liquid sodium, Eq. 75 gives a value for $K = 11 \times 10^{-3}$; Eq. 76 gives $K = 6.9 \times 10^{-3}$. The average value of K from Eqs. 75 and 76 is 8.95, which is 15% higher than the value of Eq. 73. This is in a good agreement, since the average deviation in Eq. 73 is 20%.

Several correlations have been reported for the critical heat flux of saturated sodium. The Caswell and Balzhiser⁴¹ correlation is given by

$$q''_{\text{crit}} = 5.05 \times 10^5 (P)^{0.18} (\text{Btu/hr-ft}^2). \quad (77)$$

Subbotin *et al.*⁴² suggest the correlation

$$q''_{\text{crit}} = (5.52 + 0.326P) \times 10^5 (\text{Btu/hr-ft}^2). \quad (78)$$

Noyes and Lurie⁴³ correlated their data by

$$q''_{\text{crit}} = (4.0 + 1.75P^{0.467}) \times 10^5 (\text{Btu/hr-ft}^2). \quad (79)$$

All these correlations are in good agreement. At atmospheric pressure, they predict 8.2×10^5 , 1×10^6 , and 1×10^6 Btu/hr-ft², respectively. Equation 79 together with the empirical correlation proposed by Caswell and Balzhiser⁴¹ for liquid metals:

$$q_{\text{crit}}'' = 1.18 \times 10^{-8} \left(\frac{\rho_\ell}{\rho} \right)^{0.71} \left(\frac{A^2 \rho K_\ell}{C_{p\ell} \sigma} \right) \quad (80)$$

are recommended in Ref. 40. The last equation is shown to be in good agreement with Eq. 79.

The value of $q_{\text{crit}}''(\text{sat})$ given by Eq. 72 is about four times higher than that given by any of the investigators. This may be acceptable according to the findings of Schrock et al., who investigated transient boiling extensively. The results of their work indicate that the apparent critical heat flux under transient conditions are 5 to 10 times greater than the corresponding critical fluxes for steady-state, subcooled pool boiling.

The findings of this section indicate that the effect of subcooling on the critical heat flux for sodium as given by Eq. 73 is in fairly good agreement with theoretical predictions. The extrapolated value of q_{crit}'' to the saturation temperature is higher than the steady-state values predicted in the literature. All the correlations previously used were based on data collected at pressures lower than atmospheric.

VIII. GENERAL DISCUSSION, CONCLUSIONS, AND RECOMMENDATIONS

It is useful to analyze the experimental results obtained in this investigation from two distinct points of view. The first is the extent to which the present study represents the case of steady-state boiling from a sphere. The second is the manner in which these results could improve the understanding of the fuel-coolant interaction problem and the safety aspects connected with it.

Although it is difficult to achieve steady-state-boiling heat transfer from a sphere, one has to consider the applicability of the results of the transient-boiling method used in this study to the steady-state case. The relationship of quench data to steady-state pool data has been investigated by Bergles and Thompson⁵⁹ for the case of a cylinder. Their results for both water and Freon-113 indicate that in quenching there is a general tendency toward higher wall superheats and lower values of the surface heat flux, both in the transition and nucleate regions. There are, however, several uncertainties connected with the surface condition of their test specimens which may account for these tendencies.

Because of the lack of data to compare with, it is impossible to make any firm conclusions about this point, particularly in the case of sodium.

It is reasonable to assume that the data for the film region are comparable to those which may be obtained in steady-state experiments. This is because of the "slow" nature of the cooling process in that region. One can hardly expect the same for transition or nucleate boiling, where rapid cooling of the surface takes place.

The preliminary results with water and stainless steel spheres indicated that wall superheats were much higher than the steady-state values reported.⁵⁹ But the values of the critical heat flux for sodium, as well as water, were in reasonable agreement with the steady-state value.

From a reactor-safety point of view, it has been demonstrated that subcooled sodium is capable of violent interaction with hot, solid particles. The measured pressures during such interactions are appreciable, considering that this was caused by single particles.

One important finding is the presence of an "optimum" range of subcooling at which the interaction has a maximum degree of violence. This might be interesting from a reactor-design point of view. If the LMFBR is designed such that under no circumstances would it be operated in the "optimum range," it might be possible to avoid serious damage that would otherwise result from a core-meltdown accident. Operating under higher

pressures might be one solution. This would increase the degree of subcooling beyond the "optimum range" without sacrificing thermodynamic efficiency.

Several conclusions can be drawn from the experimental results and the discussion. These conclusions are assumed to apply strictly to the transient-boiling case, although it is believed that, in a qualitative sense, at least, these conclusions also apply to the steady-state case.

The experimental results in this study were fairly reproducible in the film-boiling region. In both the transition- and nucleate-boiling regimes, a strong dependence on the surface finish existed. This led to an appreciable scatter in the experimental data. The reproducibility of data in the last two regions was noticeably better than that reported by several investigators.

In the film-boiling region, the initial wall temperature had a negligible influence on the cooling behavior of the hot sphere, which cooled in an almost isothermal fashion. However, similar hot spheres, having different initial temperatures, behaved differently when submerged in liquid sodium when the initial regime was either transition or nucleate boiling.

Subcooling is the one variable studied that had the most noticeable effect on the boiling curve. As subcooling increased, the boiling curve shifted toward higher heat fluxes and higher wall superheats.

For a given initial sphere temperature, a degree of subcooling was reached which was high enough to suppress film boiling completely. This was achieved at an initial sphere superheat of 2530°F, when the degree of subcooling was between 57 and 75°F.

A small variation in the depth at which the sphere was submerged did not affect its cooling behavior appreciably. Greater depth would affect the different boiling regimes, probably in the same manner as does an increase in the external pressure.

Changing the sphere diameter from 1 to 3/4 in. did not have a detectable effect on their boiling curves, whereas for the 1/2-in. sphere the heat-transfer rates increased in the film boiling region. The increased rate was roughly proportional to $(1/D)^{1/4}$, provided that the contribution of radiative heat transfer is small.

A model for the natural-convection, subcooled-film-boiling, heat-transfer coefficient was suggested. This closely followed the one developed by Hendricks and Baumeister²² for the saturated case. The general solution was suggested to have the form of Eq. 23 and was given by the expression

$$h_t = h_{fb} + 0.88h_r + Kh_c(\Delta T_{sc}/\Delta T_{sat}). \quad (55)$$

where K is a function of the subcooling given by Eqs. 56 and 57. The correlation represents the data reasonably well, except in the range of subcooling when film boiling is suppressed.

To the best knowledge of the author, no analytical or empirical solution has been reported in the literature to account for the problem of subcooled, natural-convection, film boiling from spheres. Nor are there any reported data for subcooled film boiling in sodium for any geometry at atmospheric pressure.

Both the heat flux and wall superheat at the Leidenfrost point depended on the degree of subcooling. These were correlated by

$$(q/A)_{\min} = 6.3 \times 10^4 + 1.9 \times 10^3 \Delta T_{sc} (\text{Btu/hr-ft}^2) \quad (67)$$

and

$$\Delta T_{\min} = 7.9 \times 10^2 + 12.2 \Delta T_{sc} (^{\circ}\text{F}). \quad (68)$$

Again, no data are reported in the literature for sodium to allow comparison. The above correlations represent the data fairly well, but give higher values than theoretical predictions based on hydrodynamic stability. The onset of transition boiling could be determined with reasonable accuracy from pressure measurements in the sodium pool.

The transition-boiling region was characterized by violent interaction between the hot spheres and the subcooled sodium. The degree of violence increased with subcooling, reached a maximum, and then decreased again as subcooling becomes small. The region of maximum violence occurred for sodium temperatures between 1320 to 1570°F at atmospheric pressure.

The critical heat flux increased with the degree of subcooling. This was correlated by the expression

$$q_{\text{crit}(sc)}'' = 4.1 \times 10^6 [1 + 7.8 \times 10^{-3} \Delta T_{sc}] (\text{Btu/hr-ft}^2). \quad (71)$$

The coefficient of T_{sc} is in reasonable agreement with theoretical predictions. The predicted value of $q_{\text{crit}(sat)}'' (\Delta T_{sc} = 0)$ is higher than the value predicted by several other correlations, most of which were based on experiments conducted under pressures lower than atmospheric.

After reviewing the major conclusions of the present study and using the background information collected by similar experiments in water, it can be concluded that there is a marked similarity between sodium and water in their transient-cooling behavior. Almost all the findings reported for sodium apply in a qualitative sense to water.

Some recommendations are made for future work:

1. An analytical solution to the heat-transfer problem in the film region should be explored.
2. A substantial amount of needed information can be obtained by investigating the saturated-film region. The experimental equipment used in the present investigation could handle such a task, although some modifications may be advantageous. These include external cooling of the pool pressure transducer, rather than cooling the cold trap. Also, some form of a condenser on the top section of the pool can eliminate the problems caused by vapor escaping from the top. A better interpretation of the pressure data may be achieved if the pressure transducer is moved to the same level as the sphere.
3. The range of violent interaction is of particular interest from a reactor-safety point of view. This range should probably be more thoroughly investigated, especially in experiments using molten UO_2 and sodium.
4. Spheres of $1/4$ -in. diameter can probably be investigated, particularly at low degrees of subcooling. This is because film boiling is likely to occur and the heat-transfer rate is low enough to allow for a reasonable cooling time. This can be followed by investigating multiple particles, made of $1/4$ -in. spheres and shaped in the form of a 1-in. sphere (as close a geometry as possible). In this manner, the obtained results may be compared to the results of the solid, 1-, and $1/4$ -in. spheres.
5. The problem of uniformity of the cooling pattern in the sphere was not completely resolved in this investigation. A series of runs using three identical spheres with thermocouples at 0, 90, and 180° from the top can establish a better averaging procedure than that suggested in the present work.
6. Although this may prove to be difficult experimentally, it would be interesting to investigate the case of a spherical shell filled with a molten metal, having thermocouples attached at various radial positions. For this purpose it would be easier as well as more instructive to use transparent, low-boiling point liquids, such as Freon-113.

APPENDIX A

Sodium-boiling Curves

The total number of experiments conducted in the present investigation, with sodium, were 43. Only 23 were considered completely successful, since they produced a satisfactory temperature-time trace. This made possible proceeding with the numerical analysis (see Ch. VI) using these temperature-time data. The variables used in the 23 "successful" runs are given in Table A.1. The final results of the numerical analysis of these data are the sodium-boiling curves appearing in Appendix A of the doctoral dissertation *Transient Boiling Heat Transfer from Spheres to Sodium* (August 1971), by M. M. K. Farahat, which may be obtained from Northwestern University, Evanston, Illinois.

TABLE A.1. Summary of the Experimental Variables Which Correspond to the Boiling Curves

Run No.	Sphere			Sodium			Duration ^a of Boiling, sec	Duration of Film Boiling, sec	Duration of Transition Boiling, sec
	No.	Diameter, in.	Thermocouple Distance, in.	Initial Temp, °F	Temp, °F	Depth, in.			
5	2	1.00	0.020	3442	392	4.5	0.7	-	-
8	4	0.99	0.025	4039	752	4.5	1.3	-	0.08
9	4	0.99	0.025	3820	752	4.5	0.7	-	0.04
10	5	1.00	0.011	4132	932	4.5	1.1	-	0.04
11	6	1.00	0.025	4132	1112	4.5	1.3	-	0.08
12	7	1.01	0.016	4012	1292	4.5	2.0	-	0.19
24	10	1.00	0.037	4195	1607	4.5	61.3	43.00	38.03
25	10	1.00	0.037	3894	1607	4.5	53.0	36.25	16.35
26	10	1.00	0.037	3607	1607	4.5	43.6	25.05	38.08
27	10	1.00	0.037	3463	1607	4.5	40.4	21.60	38.60
28	10	1.00	0.037	2785	1607	4.5	26.8	1.50	34.90
29	10	1.00	0.037	4195	1607	4.5	57.4	40.15	16.65
30	10	1.00	0.037	4234	1562	4.5	21.5	38.00	2.60
32	11	1.00	0.044	3993	1598	4.5	54.6	37.00	17.50
33	11	1.00	0.044	4030	1580	4.5	44.9	27.70	16.33
35	12	1.00	0.045	3980	1589	4.5	47.6	32.00	15.10
36	12	1.00	0.045	3995	1589	3.0	53.3	36.90	16.12
37	12	1.00	0.045	4148	1544	4.5	4.0	-	3.14
38	13	0.75	0.032	4088	1607	4.5	41.2	27.60	13.50
39	13	0.75	0.032	4180	1598	4.5	36.4	24.25	12.03
41	14	0.50	0.024	4080	1607	4.5	22.8	15.60	7.15
42	14	0.50	0.024	4297	1589	4.5	17.9	13.10	4.78
43	14	0.50	0.024	4281	1562	4.5	11.8	7.90	3.77

^aThe duration of nucleate boiling is the difference between the total boiling time and the sum of film and transition periods. This is not the total duration of nucleate boiling, but rather the period of time through which nucleate boiling was taken into consideration.

APPENDIX B

Experimental Techniques

The typical step-by-step procedure followed in the sodium experiments is given below:

1. Check the dry-box atmosphere for the oxygen and water contents, which should not exceed 30 and 5 ppm, respectively.
2. Turn on all electrical equipment and allow a 30-min period to warm up.
3. Adjust the sodium in the pot (at 392°F) to the desired level, and check the shiny appearance of the sodium surface. This is an important double check on the oxygen and water analyzers.
4. Adjust the oscillograph, amplifiers, and the bias voltage to the proper setting.
5. Turn on the sodium-heater power.
6. Turn on the water for the oil heat exchanger, the oil pump, and the transformer-cooling water.
7. Check the pressures of oil and transformer-cooling water. The induction generator will not operate unless both pressures are at the correct level.
8. Continuously monitor the sodium temperature measured at the cold leg. This should not exceed 400°F. When it gets higher, start circulating helium gas through the cooling jacket. The temperature should always remain below 400°F, but exceed 250°F.
9. Check the pressures of the argon gas used in the transport and gate pneumatic cylinders.
10. Use the outside manual controls to raise the sphere up.
11. Allow the transport-pneumatic cylinder to exhaust and close the argon-feed solenoid valve.
12. Adjust the sphere in the center of the induction coil (this will remain in place so long as the argon-feed valve is closed).
13. Turn on the generator power, and start heating the sphere. This step should be timed, so that the sphere reaches the required temperature, approximately 5 min after the pool attains its steady-state temperature.
14. Continuously monitor the sphere-thermocouple output, using the DVM (the oscillograph cannot be used for this purpose because of the effect of the induction generator on the galvanometers).
15. When the sphere reaches the required temperature, turn off the generator and energize the oscillograph pilot switch.

16. After collecting the experimental data, the exhaust solenoid valve is opened and the sphere raised up manually (the hot sphere should be handled carefully). The gate is closed, using the outside switch.

17. Allow 5 min for the sodium temperature to reach steady state. The sodium temperature is then measured using an extra (C-A) thermocouple placed inside the dry box. This measurement is the one used for calculation purposes.

18. Reduce the power of the sodium heater to 50%, and reset the pool-temperature controller to 392°F.

19. Allow the sodium to cool down to 392°F.

20. Remove the sphere assembly from the dry box.

21. Submerge the sphere in mercury to dissolve the sodium that usually remains on the sphere surface. Clean the sphere surface with ethyl alcohol, then with water, and inspect the surface.

22. If the surface is not damaged, check the thermocouple for continuity, then polish the sphere surface.

23. Wash the sphere with water and alcohol, and reinstall in the dry box.

APPENDIX C

FORTRAN Computer Programs

1. SPHERE (data reduction)

```

PROGRAM SPHERE INPUT, OUTPUT, PLOT, PUNCH, TAPE33=PUNCH
S, TAPE60=INPUT, TAPE61=OUTPUT, TAPE99=PLOT
COMMON R(15), RRR(15), DW(15), TOLD(15), INEW(15)
STM (4110), SUMO(4110), Q(4110), SLP(4110)
S, PFLUX(1000), PTSAT(1000), PTSC(1000), PCOE(1000)
WR=3.1428*DENS*R**3*4/3
CP(1)=(3.764E-02)-(2.203E-06)*1+(1.080E-09)*T
THE PURPOSE OF THIS PROGRAM IS TO NUMERICALLY SOLVE THE TIME DEPENDENT
HEAT EQUATION FOR THE SPHERE USING DIFFERENCE EQUATIONS
THE FINAL OUTPUT IS PRINTED IN STATEMENT 600 AS TEMPERATURE DISTRIBUTION
HEAT FLUX, AND HEAT TRANSFER COEFFICIENT
READ IN THE PHYSICAL DIMENSIONS AND PROPERTIES OF THE SPHERE
DENS=DENSITY, LBS/CU FT., CH=SPECIFIC HEAT, COND=THERMAL CONDUCTIVITY,
BTU/HR, FT, DEG F., M=NUMBER OF NODES, IRUN=RUN NUMBER,
DIAM=SPHERE DIAMETER, IN., TCOFF=DISTANCE OF THERMOCOUPLE END FROM SPHERE
SURFACE, IN., TBLK=SODIUM TEMP, TO=INITIAL SPHERE TEMP DEG F
111 READ 100, DENS, CH, COND, M, IRUN, DIAM, TCOFF, TBLK, TO
IF (IN.EQ.0) GO TO 11
RO=DIAM/24.
OR=DIAM/(2*(M-1))
TOFF=(OR-TCOFF)/12.
DRHO=2*OR/DIAM
THETA=0.
ALFA=COND/(DENS*CH)
PI=3.1428
CALCULATE RADIUS AND VOLUME AT EACH SPHERE SUBDIVISION
K=M-1
DO 1 I=1, M
R(1)=(I-1)/M-1.*RO
RRR(1)=R(1)**3
TOLD(1)=TO
1 CONTINUE
CALCULATE RADIUS AND VOLUME AT THERMOCOUPLE POSITION
R(M)=R(1)
VK=R(K)**3
VM=R(M)**3
SUMO(1)=W(R(M))*CP(TQ)*TO
ROFF=TOFF+R(K)
VOFF=ROFF**3
VRATIO=(VK-VM)/(VK-VOFF)
PRINT 200, TBLK, TO, DIAM, OR, TCOFF, IRUN
L=0
NM=0
READ IN TIME INTERVAL, TEMPERATURE READING INTERVAL, TEMPERATURE
CALCULATING INTERVAL AND THE NUMBER OF SUB-SUBDIVISIONS
2 READ 400, DTIM, JK, JJ, N
TN(1)=TOLD(1)
JO=JK+1
IF (JJ.EQ.0) GO TO 10
NM=(N-1)/JJ
NM=NM+NM
READ SURFACE TEMPERATURE AT EACH DTIM INTERVAL
READ 505, (TW(J), J=JO, N, JK)
CALCULATE WEIGHT OF EACH OF (M-1) SPHERE DIVISIONS
DO 3 I=1, M
DW(I)=W(R(1+I))-W(R(1))
3 CONTINUE
JK=JK-1
NJK=N-JK
DOT=DTIM/JK
DTIME=DOT/3600
DTAU=ALFA*DTIME/(RO*RO)
RATIO=DTAU/IDRHO*DRHO
PRINT 700, RATIO, JJ, DTIM
CALCULATE SUBDIVIDED SURFACE TEMPERATURE
DO 5 J=1, NJK, JK
SLP(J)=(TN(1)-TW(J+JK))/DTIM
DO 4 KK=1, JKK
TW(1+KK)=TW(1)-SLP(J)*DDT*KK
4 CONTINUE
5 CONTINUE
JA=-1

```



```

      TH=TWINI
      DO 8 J=1,N
      JA=JA+1
      TW(J)=TOLD(KI)-(TOLD(KI)-TW(J))*VRATIO
12  TOLD(N)=TW(J)
C   CALCULATE TEMPERATURE FOR NEXT TIME INTERVAL AT THE CENTER
      TNEW(1)=(6*RATIO)+(TOLD(2)-TOLD(1))+TOLD(1)
C   CALCULATE TEMPERATURE AT THE NEXT TIME INTERVAL ELSEWHERE
      DO 6 J=2,K
      TNEW(J)=TOLD(J)+RATIO*(1/(1-1)*TOLD(J+1)-2*TOLD(J)+(1-2)/(1-1))*
      $TOLD(J-1))
      6 CONTINUE
C   IF(JA.NE.JJ)GO TO 81
C   CALCULATE TOTAL HEAT FLUX,TOTAL HEAT TRANSFER COEFFICIENT,
C   AND CORRECTED AVERAGE SURFACE TEMPERATURE
      SUMQ(J)=0
      DO 7 I=1,K
      RAY=(1/RRR(I)+RRR(I+1))/2.)**(1/3.)
      TAV=TOLD(1)+(TOLD(I+1)-TOLD(1))/IRR(I+1)-R(1)) *(RAY-R(1))
      DO=DW(I)*CP(TAV)*TAV
      7 SUMQ(J)=SUMQ(J)+DO
      TAV=SUMQ(J)/(CH*W(RIM))
      Q(J)=SUMQ(J)-JA-SUMQ(J)
      FLUX=Q(J)/(DTIME*4*PI*RO*RO*JA)
      STAV=(TW(J)-JA)+TW(J)+.5
      TSC=STAV-TBULK
      TSAT=STAV-1618.6
      IF(FLUX.LE.0.0) GO TO 34
      GO TO 35
34  PRINT 344,FLUX
      GO TO 30
35  IF(TSC.LE.0.0) GO TO 36
      GO TO 37
36  PRINT 366,TSC
      GO TO 30
37  IF(TSAT.LE.0.0) GO TO 38
      GO TO 39
38  PRINT 381,TSAT
39  DO14 I=1,100
      READ 335,SPACE
      IF(SPACE.EQ.0.0) GO TO 10
14  CONTINUE
39  L=L+1
      COEFF=FLUX/TSAT
      PCOEFF=ALOG10(COEFF)
      TIME=DDT*(J-1)+THETA
      PFLUX(L)=ALOG10(FLUX)
      PTSAT(L)=ALOG10(TSAT)
      JA=0
81  DO 82 I=1,K
      TOLD(I)=TNEW(I)
82  CONTINUE
      8 CONTINUE
      THETA=TIME
      SUMQ(1)=SUMQ(N)
      TOLD(M)=TH
      GO TO 2
10  PRINT 1000
      PRINT 990,TBULK,TO
      PRINT 331,NM,L
      PRINT 334,TCOFF,IRUN
      PRINT 1000
      PUNCH 332,IRUN
      PUNCH 332,L
      PUNCH 332,1(PTSAT(1),PFLUX(1),L-1,L)
      GO TO 111
11  STOP
100 FORMAT (F6.1,X,F6.4,X,F5.2/2(12,X),2(F5.3,X),2(F7.2,X),1
200 FORMAT (1M1, *COOLANT TEMPERATURE*,BX,*,*,F6.1),* F*/X,*INITIAL SPH
      $ERE TEMPERATURE*,X,F6.1,* F*/X,*SPHERE DIAMETER*,12X,*,*,F5.3,*
      $ INCH*/X,*RADIAL INCREMENT*,11X,*,*,F5.3,* INCH*/
      $X,*THERMOCOUPLE DISTANCE *,F5.3,* INCH*/X,*RUN NUMBER *,13////)
300 FORMAT (X,*TIME*,10X,*T E M P E R A T U R E*,10X,*O I L T R I B U T
      $ I O N*,BX,*TAV*,BX,*FLUX*,4X,*PFLUX*,2X,*TSAT*,X,*PTSAT*,4X,*COEF
      $F*,4X,*PCOEFF*)
301 FORMAT (X,*TSAT = *,F9.2)
331 FORMAT (X,*NUMBER OF POINTS *,15/X,*L= *,15)
332 FORMAT (15)
333 FORMAT (4E20,13)
334 FORMAT (1X,*THERMOCOUPLE DISTANCE *,F5.3,* INCH*/
      $X,*RUN NUMBER *,12)
335 FORMAT (2X,F6.1)

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```

344 FORMAT(X,4FLUX = *,E10.2)
366 FORMAT(X,7TSC = *,F9.2)
400 FORMAT(F5.3,2I4,16)
500 FORMAT (E15.6)
505 FORMAT (10F8.2)
400 FORMAT(X,F6.3,2X,11(X,F5.0),2X,F5.0,2X,E9.2,2X,F5.3,2X,F5.3,2X,
      F5.3,2X,E9.2,2X,F5.3)
700 FORMAT(X///,X,*RATIO = *,E15.6/X,*NUMBER OF DIVISIONS BETWEEN TWO
      TEMPERATURE READINGS = *,14/X,*TIME INTERVAL BETWEEN TWO TEMPERAT
      URE READINGS = *,F7.3,* SECS*//)
900 FORMAT(X///,63X,
      $ LOG TSAT DEG F*//X,*TBULK = *,F7.2,* F*/X,*INITIAL
      TEMPERATURE = *,F7.2,* F*)
910 FORMAT (15)
1000 FORMAT(1H1)
2000 FORMAT (X,2E20,13)
END

```

2. WRETC (voltage input/degrees F output)

```

PROGRAM WRETC(INPUT,OUTPUT,PLOT,PUNCH,TAPE33=PUNCH
      1, TAPE60=INPUT,TAPE61=OUTPUT,TAPE99=PLOT)
C THIS PROGRAM ACCEPTS THE DATA POINTS IN VOLTS, AS READ BY THE DATA SCALAR
C AND TRANSFORMS THEM INTO TEMPERATURE OUTPUT IN DEG F
      DIMENSION T(150),TMV(150),TV(1000),TFEN(1000),TCEN(1000)
C THE MILLIVOLTS/DEGREES F - THERMOCOUPLE CALIBRATION DATA ARE FIRST READ
C THE RANGE OF TEMPERATURE IS 100-4400 F
      READ 1000,(T(I),TMV(I)),I=1,401
C THE PROGRAM ACCEPTS THE FOLLOWING AS INPUT DATA
C BIAS=THE BIAS VOLTAGE IN MILLIVOLTS, AMP=POWER AMPLIFIER GAIN, N=NUMBER
C OF DATA POINTS, TV(I)=OUTPUT OF THE DATA SCALAR IN VOLTS
      11 READ 2000,IRUN,TB,TO,RT,RTMV,BIAS,AMP,N,(TV(I)),J=1,N)
      IF(IRUN.EQ.0)GO TO 12
      PRINT 2001,IRUN,RT,TB,TO
      PRINT 2000
      K=0
      DO 10 J=1,N
      C THE PROGRAM CONDUCTS A SEARCH USING LINEAR INTERPOLATION IN STEPS OF 100 F
      TV(J) = TV(J)*1000./AMP + BIAS + RTMV
      DO 2 J=1,45
      IF (TV(J).LE.TMV(1).AND.TV(J).GT.TMV(1+1)) 1,2
      1 TFEN(J)=T(1+1) + (T(1)-T(1+1))/(TMV(1)-TMV(1+1))*(TV(J)-TMV(1+1))
      TCEN(J)=TFEN(J)-32./1.8
      K=K+1
      PRINT 4000,TV(J),TFEN(J),TCEN(J)
      GO TO 10
      2 CONTINUE
      10 CONTINUE
      PRINT 7000
      KK=N-1
C CALCULATE THE DIFFERENCE BETWEEN EVERY TWO CONSECUTIVE TEMPERATURES
      L=0
      DO 3 K=1,KK
      DELTF = TFEN(K)-TFEN(K+1)
      L=L+1
      PRINT 8000,L,TFEN(K),DELTF
      3 CONTINUE
      PUNCH 110,IRUN
      K=2
      99 READ 100,L
      IF(L.LT.0)GO TO 13
      N=L-K+1
      PUNCH 100,N
      PUNCH 6000,(TFEN(J)),J=K,L)
      K=L+1
      GO TO 99
      6000 FORMAT (10F8.2)
      13 GO TO 11
      12 STOP
      100 FORMAT(13)
      110 FORMAT(X,*RUN NUMBER =*,12)
      1000 FORMAT (F9.3)
      2000 FORMAT (12,X,F6.1,X,F6.1,X,F4.1,X,F5.3,X,F6.3,X,F6.1,X,15/
      $10F8.3)
      2001 FORMAT(1H1,X,*RUN NUMBER =*,12/X,*ROOM TEMP =*,F4.1/
      $X,*SODIUM TEMP=*,F6.1/X,*INITIAL SPHERE TEMP =*,F6.1/)//)
      3000 FORMAT (10X,*MILLIVOLTS*,10X,*TEMP DEG F*,10X,*TEMP DEG C*//)
      4000 FORMAT (10X,F6.3,14X,F7.2,13X,F7.2,10X,13)
      5000 FORMAT (15)

```

```

7000 FORMAT (1M1,TEMP DEG F,10X,DIFFERENCE DEG F)
8000 FORMAT (X,(3,X,F7.2,13X,F7.2)

```

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END

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* END OF RECORD

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```

4400. 37.570
4350. 37.490
4300. 37.380
4250. 37.230
4200. 37.045
4100. 36.619
4000. 36.133
3900. 35.522
3800. 34.830
3700. 34.120
3600. 33.401
3500. 32.661
3400. 31.903
3300. 31.136
3200. 30.368
3100. 29.591
3000. 28.796
2900. 27.990
2800. 27.152
2700. 26.313
2600. 25.453
2500. 24.574
2400. 23.679
2300. 22.757
2200. 21.802
2100. 20.822
2000. 19.823
1900. 18.813
1800. 17.793
1700. 16.764
1600. 15.723
1500. 14.671
1400. 13.608
1300. 12.536
1200. 11.456
1100. 10.379
1000. 9.304
900. 8.230
800. 7.163
700. 6.106
600. 5.058
500. 4.036
400. 3.063
300. 2.145
200. 1.280
100. 0.475

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DEGREES F/MILLIVOLTS, INPUT DATA

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FOR SPHERE THERMOCOUPLE

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APPENDIX D

Preliminary Experiments with Water1. Experimental Apparatus

The experimental apparatus used in the water experiments is shown in Fig. D.1. In general, the arrangement used for the water experiments differs from that used with sodium in the details of the sphere-thermocouple assembly and the sphere-transport system.

Fig. D.1. Experimental Apparatus Used in Water Experiments.
ANL Neg. No. 116-1114 No. 17.

a. The Sphere-thermocouple Assembly

In the preliminary experiments with water, a 1-in.-dia Type 304 stainless steel sphere was used. A sphere was connected to the dropping system through a 3/16-in.-dia supporting tube, 9 in. long, and a 1-in.-dia flange, as shown in Fig. D.5. The hot grounded junction of a Chromel-Alumel (C-A) thermocouple was welded to the surface of the sphere. The sheaths of all thermocouples were stainless steel of 1/8-in. OD with alumina insulation. The hot junctions were of the grounded type. The sphere-thermocouple assembly used is shown in Fig. D.5.

b. The Sphere-dropping System

The sphere-dropping system for the water experiments is shown in Fig. D.8. This is a 3-in. stainless steel pipe, 18 in. long, with a circular flange on the top and a square flange on the bottom end of the pipe. A solenoid that has a rectangular latch plunger is installed on the upper flange. The sphere assembly is supported by the solenoid plunger. When the

solenoid is activated, the plunger is pulled back and the upper circular disc on the sphere assembly is free to fall downwards. The sphere stops at a predetermined height when the lower disc on the assembly reaches a buffer ring. The depth of the sphere in the

pot is determined by the vertical position of the buffer ring. This position can be adjusted such that the sphere is inserted 2 to 8 in. deep in water in 1-in. steps.

Fig. D.2. Sodium Pot, Scale 1:2

Fig. D.3. Pool Sliding Gate, Scale 1:1.
ANL Neg. No. 900-1013.

c. Other Parts of Experimental Apparatus

Some components of the experimental apparatus common to both the water and sodium experiments are shown in this appendix. The sodium/water pot, the pool gate, and the cold leg are shown in detail in Figs. D.2-D.4. The power transformer and induction coil are shown in Figs. D.6 and D.7.

2. Experimental Procedures

An important factor to be considered in designing equipment to be

Fig. D.4. Pool Cold Trap, Scale 1:1.
ANL Neg. No. 900-1018.

TANTALUM SPHERE

304-S.S. SPHERE

Fig. D.5. Sphere-Thermocouple Assembly. ANL Neg. No. 900-1004.

Fig. D.6

Transformer and Power Leads.
ANL Neg. No. 116-1114 No. 9.

Fig. D.7. Induction Coil. ANL Neg. No. 116-1114 No. 23.

Fig. D.8. Dropping System for Water Experiments

installed in a glovebox is the requirement of a minimum amount of maintenance or repair. This factor becomes more significant if the work is carried out inside the box. The primary considerations in designing the experimental apparatus for the present investigation were, therefore, simplicity and ease of replacement and repair of the various components. A detailed program of testing these components was carried out before closing the glovebox.

Preliminary experiments with water were conducted while the box was open. A considerable time was spent to improve and modify various parts of the apparatus.

a. Variables Sought

A total of 24 experiments were conducted. The water temperature was varied between 74 and 194°F. The initial temperature of the sphere was varied between 842 and 1472°F. In all the experiments 1-in. stainless steel spheres were used. The spheres were dropped to a depth of 4 in. in the water. Boiled-out distilled water was used.

b. Experimental Procedure

The procedure for the water experiments was designed to explore the possibility of reproducing these experiments accurately. This was essential to gain confidence in the experimental apparatus and instrumentation.

Water was first heated to the desired temperature. The electrical power to the heater was then reduced to a level sufficient to compensate for heat losses. In this manner, a steady state was maintained without overheating the pot or the insulation.

The stainless steel sphere was polished and positioned at the center of the induction coil. In order to heat the sphere, the generator power was turned on and slowly increased. During heating, the sphere surface temperature was monitored using the sphere thermocouple, reading through a digital voltmeter. When the sphere temperature reached 100°F above the desired temperature, the generator was turned off. The sphere-thermocouple signal was switched to the oscillograph. A pencil mark on the recording paper indicated the required initial sphere temperature. As soon as the light beam reflected by the galvanometer reached the marked point, the sphere was dropped. The oscillograph transport mechanism was automatically turned off after a preset length of paper was ejected. The sphere assembly was pulled up, using the chain, until it rested on the solenoid plunger. The recording paper was exposed to light to obtain a visible trace and sprayed with a stabilizing lacquer.* It was left to dry, then stored for future data read-off.

c. Troubleshooting and Remedies

Several problems were encountered during the preliminary experiments with water. These led to major changes in the equipment design prior to the sodium experiments.

One of the main problems faced was the effect on the sonic and pressure transducers of the impact of the sphere-assembly buffer. The impact was strong enough to mask any useful signal generated during the first 600-700 msec of the experiment. The mechanical shock, moreover, was detrimental to the sphere-thermocouple junction, which broke after a few experiments. Minor changes, such as placing rubber, cork, or Teflon rings of various thicknesses as buffers, were not effective.

A shock absorber was tried in an effort to eliminate this problem. A horizontal arm was attached to the falling buffer disc. At the end of the arm, a small disc was attached facing downwards. A pneumatic cylinder (Bimba Model 044-DP) was mounted vertically on top of the heater box. As the sphere dropped, the small disc fell on the air cylinder plunger rod, and they traveled down until the buffer disc stopped. The extent of shock absorption was controlled by varying the rate of gas leakage from the cylinder. The transport speed of the sphere was reduced in order to get a reasonable shock reduction. In this manner, a longer part of the cooling period occurred while the sphere moved through the liquid. This was not desirable since the present investigation was concerned mainly with the study of heat-transfer rates from the sphere at rest.

---

*Kodak Linagraph Stabilizing Lacquer.

Although the above modification reduced the effect of the impact, two difficulties remained in connection with the transport system:

(1) The entire length of the sphere assembly (20 in.) should be guided to fall coaxially with the dropping pipe.

(2) The sphere assembly was difficult to disconnect from the dropping system.

The first difficulty was reduced by attaching guides to the sphere assembly to slide along the sides of the dropping pipe. Furthermore, a 20-in.-long, 1/4-in. guide tube was mounted on top of the dropping assembly. The tube slid through a 1-in.-thick Teflon block.

The second problem was solved by shortening the sphere supporting tube. An extension tube with flanges on both ends was bolted to the supporting tube on one end and to the dropping disc on the other. This made it possible to disconnect the sphere and short supporting tube from the dropping system.

The changes made on the experimental apparatus were satisfactory. But the apparatus was not as simple as desirable for handling in a glovebox. Basic modifications were sought to reduce the complexity of the experimental setup.

d. Safeguards

(1) One of the major concerns in designing the electric circuit for the transport system was to avoid accidental dropping of the sphere, particularly if the gate was closed. The sphere, thermocouple, supporting tube, and the gate could be damaged. A microswitch was installed in series with the dropping circuit. The switch was normally open and mounted such that it closed only when the gate was completely open. In this manner, the sphere was not released if the gate was closed. Another external switch, mounted in series with the same circuit, allowed the gate to be opened without actuating the transport mechanism.

(2) The experimental procedure was simplified by using only one pilot switch to run an experiment. Once this was actuated, both the recording paper transport system and a time-delay relay were energized. Enough time (~0.6 sec) was allowed for the paper to reach a constant speed. The delay relay triggered the gate-transport solenoid valve. Once the gate was open, the dropping solenoid was actuated and the sphere assembly released.

(3) When no experiments were conducted, the sphere assembly was raised slightly above the solenoid plunger. The chain was locked by two

hooks installed for this purpose. This removed the weight of the sphere assembly from the plunger and acted as an extra safeguard against accidental dropping of the sphere.

(4) Two safety pressure switches were added in series with the induction generator power circuit to ensure safe operation. The switches monitored the exit pressures of the silicon oil coolant of the induction coil and the water coolant of the transformer, respectively. In this manner, the induction generator could not be operated unless the required pressures of both coolants were maintained in their respective circuits.

3. Experimental Results

The experimental results with water are treated in a similar manner to those obtained with sodium. The data-reduction method used in Ch. VI is used together with the program "SPHERE" (see Appendix C).

The Chromel-Alumel thermocouple readings were transformed to temperature data by direct reading from the standard thermocouple tables supplied by the manufacturers.

No effort was made for detailed analysis of the results. In general, as mentioned before, both water and sodium behave in a similar manner with respect to the effect of subcooling on the boiling curve. This is shown in Figs. D.9-D.11.

The initial sphere temperature is shown to have little effect at the same degree of subcooling (Fig. D.12).

Fig. D.9. Effect of Subcooling on the Boiling Curve of Water; $T_{int} \sim 1418^{\circ}F$

Fig. D.10. Effect of Subcooling on the Boiling Curve of Water; $T_{int} \sim 1148^{\circ}F$

Fig. D.11. Effect of Subcooling on the Boiling Curve of Water; $T_{init} \sim 839^{\circ}\text{F}$

Fig. D.12. Effect of Initial Water Temperature on the Boiling Curve; $\Delta T_{sc} = 18^{\circ}\text{F}$

APPENDIX E

Sample Calculation

To illustrate the heat-transfer relations obtained (Eq. 55), a numerical example of a natural-convection, subcooled-film-boiling, heat-transfer coefficient is considered. The example considers a 1-in. tantalum sphere at a uniform temperature of 3760°F submerged in sodium at 1589°F (run 35). The film temperature, assuming the vapor-liquid interface to be at the saturation temperature, will be

$$T_f = \frac{3760 + 1618.6}{2} = 2689.3^\circ\text{F},$$

and

$$\Delta T_{\text{sat}} = 2140^\circ\text{F}; \Delta T_{\text{sc}} = 29.6^\circ\text{F}.$$

1. Physical Properties of Vapor

The physical properties of sodium vapor are given by Golden and Tokar.⁷⁴ Therein the specific heat and density are correlated using the data given, and correlations for the thermal conductivity and viscosity are reported:

$$K(T) = 0.1639 \times 10^{-2} + 0.3977 \times 10^{-4}T - 0.9697 \times 10^{-8}T^2 \text{ (Btu/hr-ft}^\circ\text{F)}; \quad (\text{E.1})$$

$$\mu(T) = 0.03427 + 8.176 \times 10^{-6}T \text{ (lb}_m\text{/hr-ft)}; \quad (\text{E.2})$$

$$C_p(T) = 0.840 - 1.27 \times 10^{-4}T \text{ (Btu/lb}_m\text{ }^\circ\text{F)}; \quad (\text{E.3})$$

$$\rho(T) = 1.81 \times 10^{-2} - 2.8 \times 10^{-6}T \text{ (lb}_m\text{/ft}^3\text{)}. \quad (\text{E.4})$$

All temperatures are in degrees F, and the validity range of Eqs. E.3 and E.4 is 2000-4000°F.

2. Physical Properties of Liquid

The physical properties of liquid are to be evaluated at

$$T_{fl} = \frac{1618.6 + 1589}{2} = 1603.8^\circ\text{F}.$$

Since this is close to the saturation temperature, the physical properties are evaluated at the saturation temperature.⁷⁴

$$\begin{aligned}
 \lambda &= 1666.79 \text{ Btu/lbm} \\
 \sigma &= 0.008123 \text{ lb/ft} \\
 \rho_L &= 46.2 \text{ lbm/ft}^3 \\
 K_L &= 29.38 \text{ Btu/hr-ft}^\circ\text{F} \\
 C_{pL} &= 0.3068 \text{ Btu/lbm}^\circ\text{F} \\
 \beta_L &= 1.82 \times 10^{-4} \text{ 1/}^\circ\text{F}
 \end{aligned}$$

3. Saturated Film-boiling Coefficient, h_{fb}

This is determined from Eq. 34 as follows: The Bond number Bo is determined, using Eq. 26 to be

$$Bo = 9.88.$$

This is used to determine $G(Bo)$ and θ^* from Figs. 6a and 6b in Ref. 22:

$$G(Bo) = -40.67; \quad \theta^* = 130^\circ.$$

The modified Raleigh number Ra^* is determined from Eq. 30:

$$Ra^* = 5.59 \times 10^7.$$

Thus, h_{fb} can be determined:

$$h_{fb} = 27.9 \text{ Btu/hr-ft}^2\text{ }^\circ\text{F}.$$

It is interesting to note that when Frederking and Clark's¹⁹ Eq. 6 is used, $h_{fb} = 24.7 \text{ Btu/hr-ft}^2\text{ }^\circ\text{F}$, which is in reasonable agreement with Eq. 34. It can be noted that Eq. 6 is much easier to use than Eq. 34.

4. Radiation Coefficient, h_r

This can be determined using Eq. 35. The emissivity ϵ_1 of tantalum was correlated using data of Ref. 75:

$$\epsilon_1(T) = 7.5 \times 10^{-5}T - 0.037, \quad 2000 < T < 4000^\circ\text{F}. \quad (\text{E.5})$$

The emissivity ϵ_2 of sodium was determined using Eq. 62:

$$\epsilon_2(1618.6) = 0.162.$$

The value of h_r was determined to be

$$h_r = 25.8 \text{ Btu/hr-ft}^2\text{ }^\circ\text{F}.$$

5. Natural-convection Coefficient, h_c

This was determined, using Eq. 66, as

$$\begin{aligned}
 h_c &= 0.75 \left(\frac{D^3 \rho_L^2 g \beta_L \Delta T_{sc}}{\mu_L^2} \times \frac{C_{pL}^2 \mu_L^2}{K_L^2} \right)^{1/4} \frac{K_L}{D} \\
 &= 0.75 [(K_L \rho_L C_{pL})^2 g \beta_L]^{1/4} (\Delta T_{sc}/D)^{1/4} \\
 &= 1070 \text{ Btu/hr-ft}^2 \text{ } ^\circ\text{F.}
 \end{aligned}
 \tag{E.6}$$

6. Total Film Coefficient, h_t

This was determined using correlations 55 and 56:

$$h_t = 74.9 \text{ Btu/hr-ft}^2 \text{ } ^\circ\text{F.}$$

APPENDIX F

Further Discussion on the Effect of Subcooling on the Boiling Curve

If one tries to predict the shape of the boiling curve of sodium as the degree of subcooling becomes extremely high, the available (limited) data suggest some definite trends. As the subcooling increases, the wall superheat required for nucleation increases. This means that the region of nucleate boiling is being replaced, gradually, by single-phase, convective heat transfer. On the other extreme of the boiling curve, as the wall temperature is increased, the contribution of radiative heat transfer to film boiling increases. Eventually the total heat flux in that regime will be roughly proportional to T_w^4 . This is true for all degrees of subcooling where film boiling occurs, as can be shown in Fig. 22.

A prediction of the effect of high degrees of subcooling and/or wall temperatures on the boiling curve is shown in Fig. F.1. It can be observed that the boiling curve "shrinks" on the low-temperature end due to the increased subcooling. On the high-temperature end, it "stretches," but at a slower rate because of the large slope of the line AB.

Fig. F.1. Predicted Effect of Increasing the Degree of Subcooling to Very Large Values on the Boiling Curve of Sodium. ANL Neg. No. 900-997.

The net result is that both the Leidenfrost point and the point of onset of nucleate boiling approach each other and will eventually coincide at $(\Delta T_w, \Delta T_{sc\infty})$. The degree of subcooling, $\Delta T_{sc\infty}$, can be assumed very high (at least higher than the maximum ΔT_{sc} in the present work), while

the wall temperature corresponding to ΔT_{∞} may be assumed higher than the critical temperature of sodium (since the critical temperature is the highest temperature at which a liquid can be in thermodynamic equilibrium with the vapor).

A hot particle, having a wall superheat $\Delta T_+ > \Delta T_{\infty}$, when quenched in subcooled sodium at $\Delta T_{sc+} > \Delta T_{sc\infty}$, will undergo film boiling until its temperature falls below ΔT_{∞} . There, only single-phase, convective heat transfer will take place.

The degree of subcooling, $\Delta T_{sc\infty}$, might not be obtained unless the pressure is much higher than atmospheric (possibly at the critical pressure of sodium or higher). At these high pressures (subcooling), Fig. F.1 shows that if the initial hot particle superheat is lower than ΔT_{∞} , the vapor phase will be completely absent from the particle surface.

For their rate-limited model of molten fuel-coolant interaction, Cho and Wright⁸¹ calculated the high pressures developed during the heating of the coolant by the hot fuel particles resulting from instantaneous fragmentation and mixing.

At these high pressures, based on thermodynamic considerations, no vapor formation can occur around the fragments. It is interesting to note that in the discussion in Appendix E, it was postulated that both high degrees of subcooling and high pressures will suppress vapor formation around hot particles. This is in agreement with the Cho-Wright model. However, it must be noted, that the above model⁸¹ considers the very early stages of hot fragments-coolant heat exchange (before the inertial effects play any role), a case which might not be applicable to the present investigation.

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REFERENCES

1. L. A. Bromely, *Heat Transfer in Film Boiling*, Chem. Eng. Prog. 46, 221 (1950).
2. J. C-Y. Koh, *Analysis of Film Boiling on Vertical Surfaces*, J. Heat Transfer 84, 55 (1962).
3. E. M. Sparrow and R. D. Cess, *The Effect of Subcooled Liquid on Laminar Film Boiling*, J. Heat Transfer 84, 149 (1962).
4. F. Tachibana and S. Fukui, "Heat Transfer in Film Boiling to Subcooled Liquid," *Proc. Second Internatl. Heat Transfer Conf. Part II*, 219-223 (1960).
5. K. Nishikawa and T. Ito, *Two-Phase Boundary Layer Treatment of Free Convection Film Boiling*, Int. J. Heat Mass Transfer 9, 103 (1966).
6. P. W. McFadden and R. J. Grosh, *High-flux Heat Transfer Studies: An Analytical Investigation of Laminar Film Boiling*, ANL-6060 (Oct 1959).
7. T. H. K. Frederking and J. Hoppenfeld, *Laminar Two-Phase Boundary Layers in Natural Convection Film Boiling of Subcooled Liquids*, Z. Angew. Math. Phys. 15, 388 (1964).
8. Y. Y. Hsu and J. W. Westwater, *Film Boiling from Vertical Tubes*, AIChE J. 4, 58 (1958).
9. Y. Y. Hsu and J. W. Westwater, *Approximate Theory for Film Boiling on Vertical Surfaces*, Chem. Eng. Prog., Symp. Ser. 30, 56, 15 (1960).
10. R. S. Dougal and W. M. Rohsenow, *Film Boiling on the Inside of Vertical Tubes with Upward Flow of the Fluid at Low Qualities*, MIT 9079-26 (1963).
11. P. J. Berenson, *Film Boiling Heat Transfer from a Horizontal Surface*, J. Heat Transfer 83C, 351 (1961).
12. T. D. Hamill and K. J. Baumeister, "Film Boiling Heat Transfer from a Horizontal Surface as an Optimal Boundary Value Process," *Proc. Third Internatl. Heat Transfer Conf.* 4, 59 (1966).
13. E. Ruckenstein, *Film Boiling on a Horizontal Surface*, Int. J. Heat Mass Transfer 10, 911 (1967).
14. T. D. Hamill and K. J. Baumeister, *Effect of Subcooling and Radiation on Film-boiling Heat Transfer from a Flat Plate*, NASA TN D-3925 (1967).
15. B. P. Breen and J. W. Westwater, *Effect of Diameter of Horizontal Tubes in Film Boiling Heat Transfer*, Chem. Eng. Prog. 58, 67 (1962).
16. K. J. Baumeister and T. D. Hamill, *Laminar Flow Analysis of Film Boiling from a Horizontal Wire*, NASA TN D-4035 (1967).
17. J. B. Siviour and A. J. Ede, "Heat Transfer in Subcooled Film Boiling," *Proc. Fourth Internatl. Heat Transfer Conf.*, B3.7 (1970).
18. J. E. Merte and J. A. Clark, *Study of Pool Boiling in an Accelerating System*, J. Heat Transfer, 83C, 233 (1961).
19. T. H. K. Frederking and J. A. Clark, *Natural Convection Film Boiling on a Sphere*, Advan. Cryog. Eng. 8, 501 (1963).

20. T. H. K. Frederking, R. C. Chapman, and S. Wang, *Heat Transfer and Fluid Motion During Cooldown of Single Bodies to Low Temperatures*, *Advan. Cryog. Eng.* 10, 353 (1965).
21. W. S. Bradfield, *Liquid-Solid Contact in Stable Film Boiling*, *Ind. Eng. Chem., Fundamentals* 5(2), 200 (1966).
22. R. C. Hendricks and K. J. Baumeister, *Film Boiling from Submerged Spheres*, NASA TN D-5124 (1969).
23. R. C. Hendricks and K. J. Baumeister, "Similarity and Curvature Effects in Pool Film Boiling," *Proc. Fourth Internatl. Heat Transfer Conf.*, B3.7 (1970).
24. W. S. Bradfield, *On the Effect of Subcooling on Wall Superheat in Pool Boiling*, *J. Heat Transfer* 89C, 269 (1967).
25. F. J. Walford, *Transient Heat Transfer from a Hot Nickel Sphere Moving Through Water*, *Int. J. Heat Mass Transfer* 12, 1621 (1969).
26. R. N. Jacobson and F. H. Shair, *Film Boiling from a Sphere During Forced Convection of Subcooled Water*, *Ind. Eng. Chem. Fundamentals* 9, 184 (1970).
27. R. E. Lyon, A. S. Foust, and D. L. Katz, *Boiling Heat Transfer with Liquid Metals*, *Chem. Eng. Prog. Symp. Series* 51(17), 41 (1955).
28. C. F. Bonilla, *Reactor Heat Transfer Conf. of 1956*, TID-7529, Reported by A. Padilla and R. E. Balzhiser (Ref. 31).
29. C. Lin et al., JPRS-3512 (1959), Reported by A. Padilla and R. E. Balzhiser (Ref. 31).
30. A. F. Poppendiek et al., GLR-42, AEC Contract No. A(04-3)-404, Geoscience Limited (Jan 1966), Reported by A. Padilla and R. E. Balzhiser (Ref. 31).
31. A. Padilla and R. E. Balzhiser, *Film Boiling of Potassium on a Horizontal Plate*, *Chem. Eng. Prog., Symp. Ser.* 82, 84, 81 (1968).
32. L. C. Witte, *Film Boiling from a Sphere*, *Ind. Eng. Chem. Fundamentals* 7, 517 (1968).
33. L. C. Witte, L. Baker, Jr., and D. R. Hawarth, *Heat Transfer from a Sphere into Subcooled Sodium During Forced Convection*, *J. Heat Transfer* 90C, 394 (1968).
34. N. Zuber and M. Tribus, *Further Remarks on the Stability of Boiling Heat Transfer*, UCLA58-5, AECU-3631 (1958).
35. J. H. Linehard and K. H. Sun, *Effect of Gravity and Size upon Film Boiling from Horizontal Cylinders*, *J. Heat Transfer* 92C, 292 (1970).
36. P. J. Berenson, *Experiments on Pool-Boiling Heat Transfer*, *Internatl. J. Heat Mass Transfer* 5, 985 (1962).
37. S. G. Bankoff and V. S. Mehra, *A Quenching Theory for Transition Boiling*, *Ind. Eng. Chem., Fundamentals* 1, 38 (1962).
38. J. Aoki and R. Welty, *Energy Transfer Mechanisms in Transition Pool Boiling*, *Int. J. Heat Mass Transfer* 13, 1237 (1970).

39. S. S. Kutaleladze, *On the Transition to Film Boiling under Natural Convection*, *Kotloturbostroenie* 3, 10 (1948), reported by J. G. Collier and P. G. Kosky (Ref. 40).
40. J. G. Collier and P. G. Kosky, *Natural Convective Boiling of the Alkali Metals - a Critical Review*, AERE-R5436 (1967).
41. B. F. Caswell and R. E. Balzhiser, *The Critical Heat Flux for Boiling Liquid Metal Systems*, *Chem. Eng. Prog. Symp. Ser.* 62(64), 41 (1966).
42. V. I. Subbotin, A. A. Ivashkevitch, and A. P. Kudryavzev, *Heat Removal from Reactor Fuel Elements Cooled by Liquid Metals*, Third Internatl. Conf. on Peaceful Uses of Atomic Energy, Geneva, 1964, A/CONF/28/P/328.
43. R. C. Noyes and H. Lurie, *Boiling Sodium Heat Transfer*, Third Internatl. Heat Transfer Conf., Chicago (1966) reported by Collier and Kosky (Ref. 40).
44. H. J. Ivey and D. J. Morris, *Critical Heat Flux of Saturation and Subcooled Pool Boiling in Water at Atmospheric Pressure*, Third Internatl. Heat Transfer Conf., Chicago (1966).
45. H. J. Ivey and D. J. Morris, *The Effect of Test Section Parameters on Saturation Pool Boiling Burnout at Atmospheric Pressure*, AIChE Symp. on Boiling Phenomena in Nuclear Heat Transfer, Chicago, Paper No. 160 (1962).
46. W. G. Camack and H. K. Forster, *Test of a Heat Transfer Correlation for Boiling Liquid Metals*, *Jet Propulsion*, 104 (Oct 1957).
47. H. K. Forster and N. Zuber, *Bubble Dynamics and Boiling Heat Transfer*, *AIChE* 1, 531 (1955).
48. R. C. Noyes, *An Experimental Study of Sodium Pool Boiling Heat Transfer*, *J. Heat Transfer* 85, 125 (1963).
49. R. C. Noyes, *Boiling Studies for Sodium Reactor Safety, Pt. I*, NAA-SR-7909 (1963).
50. H. Lurie and R. C. Noyes, *Boiling Studies for Sodium Reactor Safety, Pt. II*, NAA-SR-9477 (1964).
51. P. J. Marto and W. M. Rhosenow, *Effects of Surface Conditions on Nucleate Pool Boiling of Sodium*, *J. Heat Transfer* 88, 196 (1966).
52. B. S. Petukhof, S. A. Kovalev, and V. H. Zhukov, *Study of Sodium Boiling Heat Transfer*, Third Internatl. Heat Transfer Conf., Chicago (1966).
53. V. M. Borishauskii et al., *Heat Transfer in Boiling Alkali Metals*, *Sov. J. Atomic Energy* 19(2), 191 (1965).
54. N. Madsen and C. F. Bonilla, *Heat Transfer to Sodium-Potassium Alloy in Pool Boiling*, *Chem. Eng. Prog. Symp. Ser.* 66(30), 251 (1960).
55. R. E. Holtz, *The Effect of the Pressure-Temperature History upon Incipient-boiling Superheats in Liquid Metals*, ANL-7184 (June 1966).
56. A. I. Krakoviak, *Notes on the Liquid-Metal Boiling Phenomena*, ORNL-TM-618 (1963).

57. R. M. Singer and R. E. Holtz, *On the Role of Inert Gas in Incipient Boiling Liquid-Metal Experiments*, Internatl. J. Heat Mass Transfer 12, 1045 (1969).
58. R. E. Holtz, Ph.D. thesis, Northwestern University (1971).
59. A. E. Bergles and W. G. Thompson, Jr., *The Relationship of Quench Data to Steady-state Pool Boiling Data*, Int. J. Heat Mass Transfer 13, 55 (1970).
60. J. R. Howell and R. Siegel, *Thermal Radiation Heat Transfer, II*, NASA SP-164, 114 (1969).
61. B. Morren and J. Mareselle, *Emissivity Coefficient of 304-Stainless Steel Wetted by Sodium*, EURFNR-439.
62. H. L. Slettin, *Heat Transfer Analysis of 19-rod Fuel Element in Handling Cask*, NAA-SR-MEMO-1745.
63. S. Atallah, *A Relationship Between Emissivity and Thermal Conductivity of Metals*, Brit. J. Appl. Phys. 17, 573 (1966).
64. D. Swift and J. Pavlik, "Fuel-Coolant and Cladding-Coolant Interaction Studies," *Chemical Engineering Division Semiannual Report: July-December 1965*, ANL-7125, p. 187 (May 1966).
65. D. Armstrong, ANL, private communication.
66. C. E. Mendrihall, *College Physics*, D. C. Heath & Co., Boston, 4th Edition.
67. C. N. Kelber et al., *Safety Problems of Liquid-metal-cooled Fast Breeder Reactors*, ANL-7657, p. 58 (Feb 1970).
68. C. N. Kelber et al., *Ibid*, p. 62.
69. C. N. Kelber et al., *Ibid*, p. 60.
70. D. R. Armstrong, F. J. Testa, and D. Raridon, Jr., *Molten UO₂-Sodium Dropping Experiments*, Trans. Am. Nucl. Soc. 13(2) 660 (1970).
71. J. Crank, *The Mathematics of Diffusion*, Oxford University Press, Oxford, 1956.
72. C. A. Hampel, Editor, *Rare Metals Handbook*, Reinhold Publishing Corp., London, Second Edition (1961).
73. D. P. Jordan, *Film and Transition Boiling*, Adv. in Heat Transfer 5, 104 (1968).
74. G. H. Golden and J. V. Tokar, *Thermophysical Properties of Sodium*, ANL-7323 (Aug 1967).
75. W. D. Wood, *High-Temperature Materials, Thermal Radiative Properties*, Plenum Press, New York (1964).
76. N. Zuber, *Hydrodynamic Aspects of Boiling Heat Transfer*, ABCU-4439, Reported by D. P. Jordan (Ref. 73).
77. R. Cole, *A Photographic Study of Pool Boiling in the Region of the Critical Heat Flux*, Fourth Nat. Heat Transfer Conf., Buffalo (1960).
78. D. R. Armstrong, ANL, private communication.

79. L. C. Witte, J. E. Cox, and J. E. Bouvier, *The Vapor Explosion*, J. Metals, 39 (Feb 1970).
80. A. Cronenberg, Ph.D. thesis, Northwestern University (1971).
81. D. H. Cho and R. W. Wright, *A Rate Limited Model of Molten Fuel-Coolant Interaction*, Trans. Am. Nucl. Soc. 13(2), 659 (1970).
82. S. S. Kutateladze and L. L. Shneiderman, *Experimental Study of the Influence of the Temperature of a Liquid on a Change in the Rate of Boiling*, AEC-TR-3405, 95 (1953). Reported by Ivey and Morris (Ref. 44).
83. V. E. Schrock et al., *Transient Boiling Phenomena*, SAN-1013 (1966).