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HEAT TRANSFER FROM A SPHERE TO
LIQUID SODIUM DURING FORCED CONVECTION

by

Larry C. Witte

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by

Larry C. Witte

Chemical Engineering Division

Submitted to the Faculty of the Graduate College of
the Oklahoma State University
in partial fulfillment of the requirements
for the degree of
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January 1967

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NOMENCLATURE

A - area	α - thermal diffusivity
C - constant	δ - vapor film thickness
C_p - specific heat capacity	ϵ - emittance of sphere
D - diameter	η - transformation variable $\left(\int_0^\theta \sin^3 \theta d\theta\right)$
h - heat-transfer coefficient	θ - dimensionless time
ΔH - enthalpy change	θ° - azimuthal angle
k - thermal conductivity	λ - latent heat of vaporization
K - constant in Eq. 40	λ' - effective latent heat of vaporization
m - mass	μ - viscosity
M - $2/3 Ra/U_\infty$	ρ - density
M_n - Eigenvalues for analytical conduction Eq. 37	σ - Stefan-Boltzmann Constant $[0.173 \times 10^{-8} \text{ (Btu)/(ft}^2\text{(hr)(}^\circ\text{R}^4\text{)}]$
n - correlating exponent	τ - shear stress
Nu - Nusselt number	ϕ - angle measured relative to free stream velocity
p - pressure	Ψ - transformation variable
Pr - Prandtl number	<u>Subscripts</u>
q - quantity of heat transferred	avg - average
q/A - heat flux	B - bulk liquid
r - axisymmetric radial coordinate	c - cross section
r° - radial coordinate	co - conduction
R - radius of sphere	i - initial
Re - Reynolds number	j - number index for sphere surface elements
t - time	L - liquid
Δt - time change	Na - sodium
T - temperature	r - radiation
ΔT - temperature change	r° - radial
u - velocity in x-direction	sat - saturation
$\bar{u}$ - average velocity in x-direction	T - total
U - velocity	v - vapor
v - velocity in y-direction	vap - vaporization
w - mass flow rate	w - surface
x - space coordinate measured along surface of sphere	ϕ - angular
y - space coordinate measured normal to surface of sphere	∞ - free stream

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HEAT TRANSFER FROM A SPHERE TO LIQUID SODIUM DURING FORCED CONVECTION

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ABSTRACT

Heat-transfer rates from 1/2-in.-diam tantalum spheres to liquid sodium were measured as the sphere moved through a sodium pool. The experimental data were obtained by a transient technique. These experimentally obtained heat-transfer rates were compared to theoretically obtained heat-transfer rates. The range of conditions investigated was:

Sodium Temperature: 572 and 842°F

Pressure: Atmospheric

Sphere Velocity: 3.48 to 14.06 ft/sec

Sphere Surface Temperature: Up to 3600°F

This study showed that large amounts of subcooling in the liquid and motion of the sphere prevent the formation of vapor at the surface of high-temperature spheres. A theory was developed by assuming that no vapor was formed at the sphere surface and that superheating of liquid sodium near the surface could occur. This theory provides a simple technique for calculating heat-transfer rates from high-temperature spherical particles to highly subcooled liquids of high thermal conductivity.

The film-boiling theory, which was also developed in this study, represented only a first-order approximation of the actual heat-transfer process. Furthermore, the film-boiling theory predicts extremely thin vapor films, which are physically unrealistic. The thin films calculated from the film theory support the superheat theory.

The experimental data could be correlated by an expression of the form,

$$q/A = C(T_w - T_B)^n,$$

with C and n varying with sphere velocity and bulk sodium temperature. The values of C and n for three sets of experimental conditions were found:

U_∞ , ft/sec	T_B , °F	C	n
10.0	572	2.98×10^4	0.88
6.0	572	2.23×10^4	0.88
10.0	842	4.94×10^4	0.77

CHAPTER I

INTRODUCTION

Nuclear power reactor technology has progressed to the point where large reactors can be operated near metropolitan areas. An important part of the development of reactor technology has been the development of knowledge related to the consequences of severe accidents, both hypothetical and actual. The design of each reactor is accompanied by extensive analyses of many minor and some major accidents. The detailed analyses of major accidents make it possible to show that the probability of experiencing such an accident is extremely low and, moreover, that the consequences of such an occurrence would not constitute an undue public safety hazard. In effect, an accident analysis is meaningful only if the course of the hypothetical accidents and the individual accident steps have a sound scientific basis.

A destructive power excursion is one of the hypothetical accidents that must be considered in the safety analysis of a power reactor. This is particularly true for liquid-metal-cooled, fast reactors where, because of very short neutron lifetimes, there is little or no time for external control mechanisms to terminate an excursion. An important question in assessing the consequences of a destructive excursion concerns the rate of vapor generation that occurs when molten or fragmented core material contacts the liquid-metal coolant. Sudden coolant boiling may not only damage pressure pulses but may also affect the magnitude of the nuclear energy release through the agency of sodium-void reactivity feedback. Thus, the neutronic behavior as well as the mechanical behavior of the reactor core depends upon the rate of heat transfer and vapor generation occasioned by contact between overheated core fragments and the coolant.

Fast power reactors presently under development use liquid sodium as the coolant. Sodium is especially attractive as a heat-transfer medium because of its high thermal conductivity and its high boiling point (1620°F). The high boiling point makes operation under high pressure unnecessary.

This investigation is a fundamental study of the heat transfer from moving spherical particles to liquid sodium. The spherical particle simulates an extremely hot core fragment, which might be discharged into the coolant of a sodium-cooled fast reactor during a destructive nuclear excursion. The basic mechanism of heat transfer from the spherical particles to the liquid was of primary interest in this investigation.

Both theoretical and experimental studies of this problem were performed. Neither theoretical nor experimental data for heat-transfer rates from extremely hot particles moving through a liquid metal such as sodium were currently available. A theory was developed on the assumption that

film boiling was the basic mechanism for heat transfer from the spherical particles to liquid sodium. However, alternate mechanisms such as superheating of the sodium were considered and compared with experimental data in order to establish the actual heat-transfer mechanism.

Heat-transfer rates from a 1/2-in.-diam tantalum sphere to liquid sodium were measured as the sphere moved through a sodium pool. These data were obtained using a transient quenching technique similar to the method used by Clark and Merté¹ to obtain basic heat-transfer data from spherical bodies to liquid nitrogen. Sphere temperatures up to 3600°F were attained. Measurements were made in liquid sodium at 572 and 842°F (300 and 450°C). The velocity of the sphere was varied from 3.48 to 14.06 ft/sec. All measurements were made at atmospheric pressure. These experimental data were compared to heat-transfer rates computed from the theories developed in this study.

CHAPTER II

LITERATURE SURVEY

The literature survey was oriented toward boiling processes since high surface temperatures and large temperature differences between the surface and the bulk of the liquid were to be encountered in the experiments.

In recent years, the field of boiling heat transfer has received much attention. It was recognized that boiling was an efficient way to remove large quantities of energy from surfaces surrounded by liquids. In the nucleate-boiling regime, high heat fluxes can be achieved with relatively low surface temperatures. Higher surface temperatures are required to achieve high heat fluxes in the film-boiling regime.

Film boiling (which is characterized by the existence of a vapor film completely enclosing the heated surface) and nucleate boiling have been known for many years. In nucleate boiling, vapor is formed at discrete locations on the heated surface called nucleation sites. As early as 1756, the German scientist, Leidenfrost,² reported a phenomenon whereby a liquid and a heated surface were separated by a vapor film. This phenomenon is still referred to as the Leidenfrost phenomenon. Also, Boutigny in 1843 in work related to boiler explosions referred to this phenomenon and called it the "spheroidal" state of water (see Ref. 3).

Pilling and Lynch⁴ in 1920 reported the apparent differences in heat-transfer rates between nucleate and film boiling in experiments in which metal samples were quenched in water and other fluids. Mosciki and Broder⁵ in 1926 heated wires of several metals electrically. They observed that there was a definite temperature limit for each liquid in which the metal wires were submerged, above which the liquid would not wet the metal surface. This was the phenomenon of film boiling.

In 1934, Nukiyama⁶ performed boiling experiments with an electrically heated platinum wire in water. He obtained two distinct regimes of boiling behavior, nucleate and film boiling. Nukiyama also postulated that there was a transition region of boiling, which connected these two regimes, but was unable to attain it with the electrically heated wire.

The following pages summarize the more recent literature concerning nucleate and film boiling, with particular emphasis on their application to systems using liquid metals. Although not directly applicable to this investigation, these studies provide a basis for a better understanding of the problem.

A. Nucleate Boiling

The nucleate-boiling regime has resisted attempts to treat it in a theoretical manner. It is a complex physical phenomenon, and although it has been extensively investigated, there are still widely varying theories concerning the basic mechanism of nucleate boiling.

McAdams *et al.*⁷ studied surface boiling of water in forced convection in a stainless steel annulus. The effect of bulk velocity was small in the nucleate-boiling regime. Also, for a given pressure, the driving potential for heat transfer in the forced-convection boiling regime is the temperature difference, $\Delta T_{\text{sat}} = T_w - T_{\text{sat}}$, where T_w is wall temperature and T_{sat} is saturation temperature. If heat-flux data are plotted against ΔT_{sat} , velocity has no effect on the heat flux in the nucleate-boiling regime. The critical heat flux could be increased by increased subcooling in the fluid. Kreith and Summerfield⁸ confirmed this finding with similar data.

Vliet and Leppert^{9,10} investigated forced-convection nucleate boiling in both saturated and subcooled water flowing normal to a circular cylinder. They observed that, in nearly saturated water, nucleation first begins on the rear half of the cylinder and spreads to the forward portion of the cylinder as the heat flux is increased. The vapor bubbles that form on the forward portion of the cylinder grow and move around the cylinder. The bubbles separate from the cylinder somewhat downstream of the 90° position. As the heat flux is increased further, the two-phase mixture in the cylinder wake increases in vapor percentage until a vapor cavity forms behind the cylinder. Liquid, however, is still supplied to the downstream portion of the cylinder. It apparently moves into the region through breaks in the cavity wall and between individual bubbles.

Vliet and Leppert postulated two reasons for this type of flow behavior. The first was that, as heat flux increases, the amount of vapor in the cylinder wake is sufficient to allow the liquid flowing around the cylinder to continue in a tangential direction at the 90° position. The second concerns the relative densities of the liquid and vapor in the two-phase mixture moving around the surface of the cylinder. As the mixture reaches the 90° position on the cylinder and tries to follow the surface of the cylinder, an outward force arising from the inward centripetal acceleration acts upon the mixture. Since the liquid is more dense than the vapor, it experiences a greater force and tends to be separated from the vapor. Therefore, upon reaching the 90° position on the cylinder, the liquid tends to flow in a tangential direction and the vapor tends to move into the wake region.

At higher heat flux, the amount of liquid entering the vapor cavity decreases because of more closely packed bubbles. A condition is soon reached where the amount of liquid in the vapor cavity is insufficient to

cool the rear portion of the cylinder. The entire cylinder then rapidly overheats, and the critical heat flux is reached.

The flow mechanism in highly subcooled water was observed to be quite different, however. Because of the rapid condensation, there is insufficient vapor to form a vapor cavity. The liquid-vapor boundary layer continues around to practically the 180° position on the cylinder. Nucleation fails when irregular and unstable accumulations of vapor occur in this region. The mechanism of nucleation failure may be similar to that in nearly saturated water, but the region in which it occurs is much smaller and less well defined.

The combined effect of subcooling and velocity on the peak heat flux was observed to be very large. The peak heat flux from a 1/8-in.-OD tube at a fluid velocity of 11.0 ft/sec and a subcooling of 100°F was approximately seven times as great as the peak heat flux for a 1/8-in. tube at 0.65 ft/sec and subcooling of approximately 12°F.

B. Film Boiling

The existence of a film-boiling regime has been known for many years, e.g., the Leidenfrost phenomenon named after the German scientist who published his observations in 1756. This phenomenon is sometimes called the "spheroidal" state. It was so named by Boutigny in 1843. However, very little research was actually done in the area of film boiling until relatively recently. Work was begun by Bromley and his co-workers.¹¹⁻¹³ Colburn, in unpublished lecture material, referred to the apparent similarity between film boiling and film condensation (see Ref. 14). Bromley¹¹ investigated this similarity. The resulting equation for the heat-transfer coefficient for film boiling around a horizontal cylinder is identical to that given by Nusselt's theory for thin film laminar condensation with the gravity term replaced by the buoyancy term and with a differing constant (see Ref. 15). The case considered by Bromley was natural-convection film boiling, and the contribution of radiant heat transfer was neglected. Experiments performed by Bromley with six organic liquids indicated that this equation predicted the heat-transfer rates from tubes ranging in outside diameter from 0.188 to 0.468 in. Later experiments by Banchemo, Barker, and Boll¹⁶ and experiments by McAdams et al.¹⁷ indicated that this equation fails to hold for tube diameters less than 0.025 in. and greater than 0.750 in.

In a subsequent paper, Bromley et al.¹² included the effects of velocity upon film boiling. Again radiation was neglected, although corrections were made for radiation at the end of the derivation. In flows where forced convection predominates over free convection, the heat-transfer coefficient due to conduction alone varied as

$$h_{co} = C \left(\frac{k \rho \lambda' U_{\infty}}{D \Delta T} \right)^{1/2}, \quad (1)$$

where k is the thermal conductivity, ρ is the density, λ' is the latent heat of vaporization, U_{∞} is the bulk velocity of the fluid, D is the tube diameter, and ΔT is the wall temperature minus the saturation temperature. In theory, the constant C could be evaluated, but in practice, it was obtained from experimental data and is given as 2.70. For this case, Bromley gives as the total heat-transfer coefficient

$$h = h_{co} + \frac{7}{8} h_r, \quad (2)$$

where h_r is the coefficient due to radiation.

In both preceding analyses by Bromley, a linear temperature profile in the vapor film was assumed. This was shown to be valid if an "effective" latent heat of vaporization is used. This quantity is

$$\lambda' = \lambda \left(1 + \frac{0.4 C_p \Delta T}{\lambda} \right), \quad (3)$$

where C_p is the specific heat capacity of the vapor and was derived by Bromley.¹⁸

In the final paper in the series, Motte and Bromley¹³ included the effect of subcooled liquid in the flowing system. The resulting expression for the heat-transfer coefficient was quite complex, and the authors state that the equation as such is not exact and should be used for correlation purposes only. Data taken with various organic liquids point out markedly the increased heat-transfer coefficients that can be achieved with large degrees of subcooling.

In 1959, McFadden and Grosh¹⁹ performed an analysis for laminar film boiling in the free convection regime which included the effects of inertia and convection in the vapor film. This was done by using the boundary layer concept with the appropriate boundary conditions. The cases analyzed were the vertical flat plate and the horizontal cylinder. It was shown that through suitable assumptions, the boundary-layer equations were identical for these two cases.

Chang²⁰ and Hsu and Westwater²¹ analyzed film boiling on vertical plates. Hsu and Westwater included the effects of turbulence in their theoretical work.

Bradfield et al.²² performed experiments in film boiling from the surface of various hydrodynamic shapes, including a sphere, in water and

liquid nitrogen. The primary object of this program was to test the possibility of drag reduction by the formation of a vapor film around the body. Heat-transfer data were taken also, however. Some interesting photographic observations were made in conjunction with these measurements. The effect of motion was to stabilize the liquid-vapor interface (damp the laminar waves present in free-convection film boiling) up to a point where transition to turbulence resulted in "froth" flow in the boundary layer. Subcooling in the liquid also had this effect. This can be explained by a reduction in vapor film thickness due to the effects of motion and subcooling in the liquid. For very large degrees of subcooling, the liquid-vapor interface again became unstable and would collapse, and a transition to nucleate boiling would apparently occur. The surface temperature at which this happened had a definite dependence upon the degree of subcooling in the liquid. Recently, Bradfield²³ showed that there can be liquid-surface contact in the normally stable film boiling regime.

Sparrow and Cess,²⁴⁻²⁶ Tachibana and Fukui,²⁷ and Koh²⁸ analyzed the film-boiling problem. Sparrow and Cess used the boundary-layer theory in achieving solutions to the problem. The horizontal and vertical flat plate were the surfaces considered in these studies. Koh also used the boundary-layer treatment with success. Tachibana and Fukui performed an analysis on film boiling from a small wire. They used integral techniques to achieve a solution.

Frederking and Clark²⁹ analyzed the natural-convection film boiling from a sphere in a saturated liquid. The contribution of thermal radiation to the net heat transfer was neglected. The velocity and shear stress at the liquid-vapor interface were assumed to be zero. Kinetic-energy effects and energy-convection effects in the vapor film were neglected also. The film thickness was considered small compared to the radius of the sphere. By writing an energy balance at the liquid-vapor interface, an expression for the film thickness as a function of angular position could be written and solved exactly. This immediately leads to heat-transfer results, since heat is assumed to be transferred across the vapor film by conduction only, and the neglect of the energy convection terms gives a linear temperature profile in the vapor film. A comparison of this theory to data taken in liquid nitrogen shows some disagreement; however, very little data have been taken from a sphere, and further work should be undertaken to test this theory.

Clark and Merté made measurements in saturated liquid nitrogen from a 1-in.-diam copper sphere. Heat-transfer data were taken during a quenching process; i.e., the sphere was heated to a desired initial temperature and then inserted into the liquid-nitrogen pool. The thermal diffusivity of copper is such that the sphere can be assumed to be at a uniform temperature. This was verified by thermocouples installed at various radial locations in the sphere. A comparison of these data, obtained

during a transient process, to steady-state data shows good agreement. This points out the usefulness of this technique to obtain boiling data efficiently and rapidly in all regimes. A comparison of these data to the theory of Bromley shows considerable disagreement. However, these data were taken under turbulent conditions, and Bromley's theory is known to apply only to laminar-flow cases.

Kobayashi³⁰ developed a theory for the prediction of forced-convection film boiling around a sphere in a saturated liquid. An approach similar to that used by Bromley for forced-convection film boiling around a cylinder was used. Unfortunately, the pressure distribution corresponding to that around a cylinder instead of a sphere was used in the formulation of the problem. This introduces some error into the results of the theory. The manner in which Kobayashi treats the effect of thermal radiation is also in error. He states that the total heat-transfer coefficient in film boiling from a sphere is the direct sum of the coefficient calculated by assuming that thermal radiation is negligible and the coefficient calculated from radiation considerations. These two heat-transfer coefficients are not independent of each other, and cannot be added directly to obtain the total coefficient.

C. Liquid-metal Boiling

Farmer³¹ was among the first investigators to study extensively the boiling of liquid metals. Farmer boiled pure mercury from copper and chromium surfaces. The copper surface gave twice as large a film coefficient as the chromium surface. This apparently could be attributed to differences in wetting characteristics of copper and chromium by mercury.

Lyon, Foust, and Katz³² measured boiling heat-transfer coefficients for pure mercury, sodium, and cadmium and also for mercury with 0.1% sodium, mercury with 0.02% magnesium and a trace of titanium, and sodium potassium alloy. These fluids were boiled at temperatures from 670 to 1600°F and at atmospheric pressure. These liquid metals were boiled on the outside of a 3/4-in.-diam single, horizontal, stainless steel tube. A comparison of heat flux measured to mercury and mercury with 0.1% sodium at a particular temperature difference showed the effects of additives. Mercury alone did not wet stainless steel, and the heat-transfer rates for pure mercury were much lower than those for mercury with traces of sodium. Pure mercury apparently went from the convection heat-transfer regime directly into the film-boiling regime, whereas the mercury plus 0.1% sodium wet the surface and experienced nucleate boiling.

Madsen and Bonilla³³ boiled sodium-potassium alloy (44 w/o K) on a horizontal nickel surface. The boiling surface was heated electrically. Rapid fluctuations in temperature in both heating surface and boiling liquid were observed. These temperature fluctuations were accompanied by

distinct "bumps," which were immediately followed by a rapid temperature decrease. The experimental heat-transfer rates could be best correlated with the equation

$$q/A = 134p^{0.25}(\Delta T)^{1.24}, \quad (4)$$

where q/A is the heat flux, p is the pressure in the vapor space above the molten metal pool in absolute millimeters of mercury, and ΔT is the difference between the temperature of the heating surface and the equilibrium temperature for the liquid at the prevailing vapor space pressure in degrees Fahrenheit.

Korneev³⁴ reported experiments with mercury and magnesium amalgam boiling on horizontal and vertical steel tubes from 0.5 to 0.87 in. in diameter submerged in a pool. Korneev's nucleate boiling data, which were compared to the data of Lyon and Bonilla, show poor agreement. Korneev's data show ΔT_{sat} one order of magnitude higher for the same heat flux than do the data of Lyon and Bonilla. Korneev also performed experiments with pure mercury in the film-boiling regime. His data agree closely with those of Lyon. Korneev also observed no appreciable difference in temperature between the top and the bottom of the horizontal tube during film boiling in mercury.

Noyes³⁵ studied boiling from the surface of a horizontal 3/8-in.-diam cylinder immersed in a saturated sodium pool. Heat fluxes up to 800,000 Btu/hr-ft² were measured, and the surface temperature of the cylindrical test section varied from 1200 to 1500°F. Burnout data were taken at pressures between 0.5 and 1.5 psia. In the nucleate-boiling regime, the following empirical equation provided a good fit for the data:

$$q/A = 180(\Delta T_{\text{sat}})^{2.35}, \quad (5)$$

where ΔT_{sat} represents the difference in temperature between the heater surface and the boiling temperature of the sodium at the existing pressure. No variation in ΔT_{sat} at a given heat flux due to pressure variation was noted. Later work by Noyes³⁶ with 1/4-in.-diam cylinders indicated that Eq. 5 also holds for 1/4-in. test sections. However, in this experiment audible thumps and bumps were heard and were accompanied by large fluctuations in the temperature of the test-section surface.

Subbotin et al.³⁷ reported sodium boiling data taken with 1.5-in.-diam horizontal stainless steel, nickel, and copper surfaces heated by electron bombardment. These surfaces were used in the bottom of a 6.3-in.-diam stainless steel vessel with a sodium depth of 6 to 7.9 in. The data of Subbotin et al. show considerable scatter, which can probably be attributed to the use of three different types of boiling surfaces.

Figure 1 shows the data of Lyon *et al.*,³² Noyes,³⁶ and Subbotin *et al.*³⁷ for pool boiling of sodium. The data of Lyon and Noyes compare well only in the low heat-flux region. Subbotin's data generally fall somewhat lower than the data of Noyes for a given heat flux. Both Noyes' and Subbotin's data indicate that critical heat fluxes for sodium are near 1,000,000 Btu/hr-ft².

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Fig. 1. Pool-boiling Data for Sodium

bubble nucleation from heated surfaces exposed to liquid metals. Krakoviak, by modeling the behavior of the liquid metals upon fluids such as water, estimated the amount of superheat in the liquid necessary for bubble nucleation. For sodium at atmospheric pressure exposed to a surface to one on which it takes 30°F of superheat to sustain a bubble in water, Krakoviak calculated that 258°F of superheat would be necessary to initiate a bubble. If compared to distilled, degassed water, the superheat required for bubble nucleation in sodium becomes 774°F. This explains the bumping observed by several investigators when liquid metals were boiled. Since the liquid can become highly superheated, the energy of superheat in the surrounding fluid when a bubble does nucleate, can be rapidly released and a small "explosion" occurs. Holtz also used a similarity modeling between the liquid metals and water but in a different manner. Holtz's results were in fair agreement with those of Krakoviak.

In addition to the references cited previously, much data exist for boiling heat transfer in pipe and channel flow. Much of this data concerns the problem of two-phase flow in channels caused by the boiling phenomenon. Tang's⁴³ review of this literature contains more information concerning this problem.

Marto and Rohsenow³⁸ studied the effects of surface conditions on nucleate pool boiling of sodium. They found that surface roughness and past history can affect the nucleate-boiling process significantly. Marto and Rohsenow³⁹ also studied the nucleate-boiling instabilities of sodium.

Adams⁴⁰ used a quenching technique to obtain heat-transfer data from a 3-in.-diam tungsten sphere to magnesium.

Krakoviak⁴¹ and Holtz⁴² studied the conditions necessary for

CHAPTER III

ANALYTICAL STUDIES

A. Introduction

The problem of film boiling from a sphere to a liquid, such as sodium, flowing past the sphere, is quite complicated and has, at present, no satisfactory analytical solutions. Even the relatively simple problem of the velocity distribution in a fluid moving past a sphere must be solved by using a "Blasius" type solution (see Ref. 44). In this solution, the velocity of the potential flow around the sphere is assumed to have the form of a power series in x , where x is measured along the surface of the sphere from the stagnation point. The velocity distribution in the boundary layer is also represented by a power series in x with the coefficients assumed to be functions of y , the direction normal to the sphere surface. From these power series approximations, a solution for the velocity distribution around a sphere can be found.

Film boiling is normally characterized by the existence of a vapor film completely surrounding the heated surface. At some distance from the solid surface, an interface exists between the vapor film and the surrounding liquid. Since the liquid is forced

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Fig. 2. Model for Forced-convection Film Boiling around a Sphere

past the sphere, relative motion between the sphere, the vapor, and the liquid will occur. The model used for this phenomenon is shown in Fig. 2. The problem may be thought of as one of two boundary-layer systems occurring simultaneously. The first, the vapor film, moves past the solid sphere surface and is influenced in its motion by the presence of the sphere surface and also by the presence of the liquid at the liquid-vapor interface. The

second, the liquid layer overriding the vapor film, actually feels the influence of the vapor layer only. It cannot actually "see" the sphere surface. Mass crosses the liquid-vapor interface from one boundary-layer system to the other as the liquid flows past the sphere.

The problem is further complicated by the fact that the bulk liquid may be at a temperature lower than the boiling temperature of the liquid at the existing pressure. This condition of the liquid (referred to as sub-cooled) allows heat to be transferred away from the liquid-vapor interface into the bulk of the liquid. This effect will be most severe for liquids such as sodium, which have extremely high thermal conductivity. The thickness of the vapor film is intimately affected by the heat transferred into the bulk of the liquid as will be shown in Section D of this chapter.

The problem has now been formulated and consists of two phases of a fluid flowing past a sphere with simultaneous heat transfer occurring in both phase regions. The boundary-layer treatment will be applied to both the vapor boundary layer and the liquid boundary layer to obtain a solution for this problem.

B. Liquid Boundary Layer

The liquid will be assumed to move around the sphere in laminar flow. As stated previously, it shall be assumed that the liquid feels only the influence of the vapor film. If it is assumed that the liquid-vapor interface is smooth and in dynamic equilibrium, then the shearing stress acting upon the liquid-vapor interface in the liquid must be equal in magnitude to the shear stress acting upon the liquid-vapor interface in the vapor. The following relation can be written:

$$\tau_l = \tau_v, \quad (6)$$

where l and v refer to conditions in the liquid and vapor, respectively, and τ stands for shearing stress. The shearing stress may be written in an alternate form as

$$\mu_l \left(\frac{\partial u}{\partial y} \right)_l = \mu_v \left(\frac{\partial u}{\partial y} \right)_v. \quad (7)$$

Equation 7 relates the product of viscosity, μ , and velocity gradient, $\partial u / \partial y$, in the liquid at the liquid-vapor interface to the corresponding product in the vapor. The velocity gradient in the liquid may be written as

$$\left(\frac{\partial u}{\partial y} \right)_l = \frac{\mu_v}{\mu_l} \left(\frac{\partial u}{\partial y} \right)_v. \quad (8)$$

For conditions sufficiently far from the critical point, the ratio of viscosities, μ_v / μ_l , is quite small. For sodium at 1 atm and at its boiling point, the ratio is 0.071. Further, for this analysis, the velocity gradient, $\left(\frac{\partial u}{\partial y} \right)_v$, will be assumed to be sufficiently small in magnitude so that $\left(\frac{\partial u}{\partial y} \right)_l$ remains small. This assumption permits the use of potential flow theory for the calculation of the velocity field in the liquid sodium. This greatly simplifies the problem, and a solution can be obtained which, without this assumption, may have been impossible.

By using spherical coordinates, assuming axial symmetry, and neglecting conduction in the ϕ -direction, as compared to convection in that direction, the differential equation describing the temperature distribution around the spherical vapor film becomes

$$u_{r^*} \frac{\partial T}{\partial r^*} + u_{\phi} \frac{1}{r^*} \frac{\partial T}{\partial \phi} = \alpha \frac{\partial^2 T}{\partial r^{*2}} + \frac{2}{r^*} \frac{\partial T}{\partial r^*}, \quad (9)$$

where T is temperature and α is the thermal diffusivity of the liquid. The velocity components, u_{r^*} and u_{ϕ} , in accordance with the assumption of frictionless flow, can be written as

$$u_{r^*} \approx -3U_{\infty} \frac{r^* - R}{R} \cos \phi, \quad u_{\phi} \approx \frac{3}{2} U_{\infty} \sin \phi. \quad (10)$$

These are obtained directly by differentiating the stream and potential functions for frictionless flow around a sphere (see Ref. 45).

Sideman⁴⁶ showed that if the assumption is made that the heat transfer takes place in a thin layer near the interface, the term $(2/r^*)(\partial T/\partial r^*)$ may be neglected in comparison to the term $\partial^2 T/\partial r^{*2}$ in Eq. 9. This further implies that the product of the Reynolds number and the Prandtl number, called the Peclet number, is much larger than unity. The Reynolds number and the Prandtl number are defined by the equations

$$Re = \frac{\rho U D}{\mu}, \quad Pr = \frac{C_p \mu}{k}, \quad (11)$$

where ρ is density, U is velocity, D is the diameter of sphere, μ is viscosity, C_p is specific heat capacity, and k is the thermal conductivity of the liquid. Equation 9 can be simplified to

$$u_{r^*} \frac{\partial T}{\partial r^*} + u_{\phi} \frac{1}{r^*} \frac{\partial T}{\partial \phi} = \alpha \frac{\partial^2 T}{\partial r^{*2}}. \quad (12)$$

The boundary conditions are

$$\begin{aligned} T &= T_B, & r^* &= \infty, & \phi &\geq 0; \\ T &= T_{sat}, & r^* &= R, & \phi &> 0; \\ T &= T_B, & \infty &\geq r^* \geq R, & \phi &= 0. \end{aligned} \quad (13)$$

The assumption has also been made that the penetration of heat into the liquid layer is limited to small distances. Equations 12 and 13 are solved in Appendix A. The resulting expression for the temperature distribution in the liquid around the sphere is

$$\frac{\Delta T}{\Delta T_B} = \operatorname{erfc}\left(\frac{\psi}{2\sqrt{M\eta}}\right), \quad (14)$$

where

$$M = \frac{2}{3} \frac{R\alpha}{U_\infty},$$

$$\eta = \int_0^\phi \sin^3 \phi \, d\phi,$$

$$\psi = y \sin^2 \phi,$$

$$y = r - R,$$

$$\Delta T = T - T_B,$$

$$\Delta T_B = T_{\text{sat}} - T_B,$$

R is the radius of the sphere, U_∞ is the velocity of the bulk liquid, α is the thermal diffusivity of the liquid, and r is the radial coordinate. Differentiation of Eq. 14 will give the heat flux, as also shown in Appendix A. The resulting expression for heat transfer from the liquid-vapor interface into the bulk liquid is

$$\frac{q}{A} = k \frac{\Delta T_B}{\sqrt{\pi}} \frac{\sin^2 \phi}{\sqrt{M\eta}}. \quad (15)$$

C. Vapor Boundary Layer

The solution to the heat-transfer problem in the liquid boundary layer which is subject to the prescribed restrictions allows the case of the vapor boundary layer to be considered. The boundary-layer equations for axisymmetric flow of a fluid past a sphere are as follows:⁴⁴

$$\begin{aligned} \frac{\partial(ur)}{\partial x} + \frac{\partial(vr)}{\partial y} &= 0, \\ u \frac{\partial u}{\partial x} + v \frac{\partial u}{\partial y} &= -\frac{1}{\rho} \frac{\partial p}{\partial x} + \frac{\mu}{\rho} \frac{\partial^2 u}{\partial y^2}, \\ u \frac{\partial T}{\partial x} + v \frac{\partial T}{\partial y} &= \alpha \frac{\partial^2 T}{\partial y^2}. \end{aligned} \quad (16)$$

These are the equations of conservation of mass, momentum, and energy, respectively. The coordinate system is given in Fig. 2, with x measured along the surface of the sphere and y normal to the surface of the sphere. In addition, u is the velocity of the x -direction, v is the velocity in the y -direction, and p is the pressure. Implicit in Eq. 16 is the assumption that the vapor film thickness is much less than the radius of the sphere.

The following additional assumptions concerning the problem were made:

1. The vapor is pure and incompressible.
2. The sphere surface is isothermal.
3. The physical properties of the vapor are constant with respect to temperature.
4. The liquid-vapor interface is smooth.
5. Inertia effects and energy-convection effects in the vapor film can be neglected.
6. The vapor is transparent to thermal radiation; i.e., the vapor does not participate in the thermal-radiation heat-transfer process.
7. The velocity of the liquid-vapor interface can be approximated by the velocity calculated from potential theory.
8. The liquid-vapor interface is at saturation temperature.

Assumption 7 is consistent with the preceding development for the liquid layer.

Reportedly, sodium vapor is highly absorptive to thermal radiation at some wavelengths (see Ref. 47). Sparrow⁴⁸ investigated the effect of a participating vapor upon film boiling and reported that for fluids such as water under normal pressure conditions, the effect is negligible. Sparrow also pointed out that very little is known about the radiative properties of vapors, particularly vapor emissivities. In addition, it will be shown in Section D of this chapter that radiative contribution to the net heat transfer is unimportant. Therefore, assumption 6 can be used for this case.

Assumption 4 has been verified in part by Bradfield *et al.*²² They observed that during film boiling from variously shaped hydrodynamic bodies, including a sphere, ripples formed at the liquid-vapor interface if the surrounding liquid was at rest with respect to the body. If, however, the liquid was set in motion, the effect was to damp out these ripples and the liquid-vapor interface became much smoother. As the velocity was increased further, turbulence developed and the liquid-vapor interface became unstable and broke down. Bradfield described this type of flow as "froth" flow.

Assumption 2 cannot be totally justified. During film-boiling experiments from horizontal cylinders, Bromley *et al.* observed that the heat-transfer rates were higher on the forward side of the cylinders than on the backward side, and consequently the forward portion was somewhat cooler

than the rear portion. This is due to the phenomenon of flow separation that occurs in film boiling, just as in single-phase flow past a sphere or any bluff body. However, Kutateladze *et al.*⁴⁹ measured heat-transfer rates from a circular cylinder during cross flow of a liquid metal and observed that variation in heat-transfer rates around the periphery of the cylinder are far less severe than corresponding variations in fluids such as air (see Fig. 3). This is due in part to the extremely high thermal

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Fig. 3. Comparison of Heat-transfer Coefficients for Flow of Air and Liquid Metal around a Cylinder

conductivity of the liquid metals, which causes the flow patterns of the liquid around the body to take on a relatively small role in determining the heat-transfer rates from the body to the liquid. For subcooling in liquid sodium, the amount of heat transferred from the sphere is proportional to the heat transferred from the liquid-vapor interface into the bulk of the liquid, as will be shown in succeeding paragraphs. Potential flow of the liquid sodium above the vapor film has been assumed, but Kutateladze's data were taken with the liquid in contact with the solid surface. Kutateladze showed, however, that it makes little difference whether potential flow or

viscous flow is assumed; the measured heat-transfer rates compare well with either theory. Therefore, it would appear that for a liquid such as sodium, and for relatively large degrees of subcooling, assumption 2 will be valid. However, this assumption will not hold for liquids such as water, and will be increasingly in error as the liquid sodium approaches its saturation temperature.

The solution to the momentum and energy conservation Eqs. 16 may easily be written in accordance with assumption 5. These solutions are shown in Appendix A. The expression for velocity at any point in the vapor film is

$$u = \frac{3}{2} U_{\infty} \sin \phi \left[\frac{3}{4} \frac{\rho U_{\infty}}{\mu R} (\cos \phi)(y\delta - y^2) + \frac{y}{\delta} \right]. \quad (17)$$

The temperature distribution, which turns out to be linear, is

$$\frac{T - T_{sat}}{T_w - T_{sat}} = 1 - \frac{y}{\delta}, \quad (18)$$

where T_{sat} is the boiling or saturation temperature of the liquid, T_w is the sphere surface temperature, and δ is the vapor film thickness.

D. Energy Balance on Differential Film Element

Writing an energy balance on a differential element of the vapor film shown in Fig. 4, one obtains

$$dq_c + dq_r = dq_{\text{vap}} + dq_B. \quad (19)$$

Equation 19 states that the energy leaving the sphere surface, composed of conductive and radiative parts, arrives at the liquid-vapor interface, is used partially to form vapor, and is partially transferred to the bulk liquid. If the bulk liquid is at saturation temperature, then all the energy arriving at the liquid-vapor interface goes into forming vapor, and the last term, dq_B , in Eq. 19 is not necessary. The energy used to vaporize liquid at the liquid-vapor interface can be written as the product of the effective heat of vaporization and the increase in the mass flowrate in the differential film element,

$$dq_{\text{vap}} = \lambda' dw. \quad (20)$$

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Fig. 4. Differential Element
for Film Boiling
around a Sphere

The increase in mass flowrate in the film element must come from vapor which is formed at the liquid-vapor interface. Bromley¹⁸ showed for laminar film condensation that the assumption of a linear temperature profile is valid as long as an "effective" latent heat of vaporization is used in an energy balance on an element of the film. Bromley also used this concept in film-boiling analyses with good success. The expression for the effective latent heat of vaporization, λ' , as derived by Bromley is

$$\lambda' = \lambda \left(1 + \frac{0.4 C_p \Delta T}{\lambda} \right), \quad (21)$$

where $\Delta T = T_w - T_{\text{sat}}$. This expression actually accounts for the heat capacity of the vapor.

The increase in mass flowrate, dw , can be written as

$$dw = d(\rho A_c \bar{u}), \quad (22)$$

where A_c is the flow cross section of the film and $\bar{u}$ is the average velocity of the vapor flowing through A_c . A_c can be written in terms of the film thickness and angular position as

$$A_c = 2\pi R \delta \sin \phi. \quad (23)$$

The average velocity, $\bar{u}$, can be found from

$$\bar{u} \delta = \int_0^\delta u \, dy. \quad (24)$$

Replacing u by its value as given by Eq. 17, one obtains

$$\bar{u} = \frac{1}{\delta} \int_0^\delta \frac{3}{2} U_\infty \sin \phi \left[\frac{3}{4} \frac{\rho U_\infty}{\mu R} (\cos \phi) (y\delta - y^2) + \frac{y}{\delta} \right] dy. \quad (25)$$

Evaluation of this integral gives

$$\bar{u} = \frac{1}{\delta} \left\{ \frac{3}{2} U_\infty \sin \phi \left[\frac{3}{4} \frac{\rho U_\infty}{\mu R} (\cos \phi) \frac{\delta^3}{6} + \frac{\delta}{2} \right] \right\}. \quad (26)$$

Equation 22 may now be written in terms of the film thickness and angular position as

$$dw = d \left\{ \rho 2\pi R (\sin \phi) \frac{3}{2} U_\infty \sin \phi \left[\frac{3}{4} \frac{\rho U_\infty}{\mu R} (\cos \phi) \frac{\delta^3}{6} + \frac{\delta}{2} \right] \right\}. \quad (27)$$

Now, Eq. 19 becomes

$$\begin{aligned} \frac{k \Delta T}{\delta} dA + \frac{q_r}{A} dA = \lambda' 2\pi R \rho \frac{d}{d\phi} \left\{ \frac{3}{2} U_\infty \sin \phi \left[\frac{3}{4} \frac{\rho U_\infty}{\mu R} (\cos \phi) \frac{\delta^3}{6} + \frac{\delta}{2} \right] \right\} d\phi \\ + \frac{k_\ell}{\sqrt{\pi}} \frac{\sin^2 \phi}{\sqrt{M\eta}} \Delta T_B dA, \end{aligned} \quad (28)$$

where dA refers to the area element on the sphere surface adjacent to the film element. The expression for the heat transferred into the bulk of the liquid has been derived in Section B of this chapter. The radiative heat-transfer rate, q_r/A , is calculated by assuming that the surface of the sphere and the liquid-vapor interface are approximated by parallel infinite plates. Liquid sodium is highly absorptive to thermal radiation; therefore the absorptivity of the liquid-vapor interface is assumed to be unity. The heat-transfer rate due to radiation becomes

$$\frac{q_r}{A} = \sigma \epsilon (T_w^4 - T_{\text{sat}}^4), \quad (29)$$

where

σ = Stefan-Boltzmann constant,

and

ϵ = emittance of the sphere surface.

The differential area element, dA , can be written as

$$dA = 2\pi R^2 \sin \phi d\phi. \quad (30)$$

Substituting this into Eq. 28, one obtains

$$\begin{aligned} k \frac{\Delta T}{\delta} 2\pi R^2 \sin \phi d\phi + \sigma \epsilon (T_w^4 - T_{sat}^4) 2\pi R^2 \sin \phi d\phi = \lambda' 2\pi R \rho \frac{d}{d\phi} \\ \left\{ \frac{3}{2} U_\infty \sin^2 \phi \left[\frac{3}{4} \frac{\rho U_\infty}{\mu R} (\cos \phi) \frac{\delta^3}{6} + \frac{\delta}{2} \right] \right\} d\phi \\ + \frac{k_\ell}{\sqrt{\pi}} \frac{\sin^2 \phi}{\sqrt{M\eta}} 2\pi R^2 \Delta T_B \sin \phi d\phi. \end{aligned} \quad (31)$$

Simplification of Eq. 31 gives

$$\begin{aligned} k \frac{\Delta T}{\delta} + \sigma \epsilon (T_w^4 - T_{sat}^4) = \frac{3}{2} \frac{\rho U_\infty \lambda'}{R \sin \phi} \frac{d}{d\phi} \left\{ \sin^2 \phi \left[\frac{3}{4} \frac{\rho U_\infty}{\mu R} (\cos \phi) \frac{\delta^3}{6} + \frac{\delta}{2} \right] \right\} \\ + \frac{k_\ell}{\sqrt{\pi}} \frac{\sin^2 \phi}{\sqrt{M\eta}} \Delta T_B. \end{aligned} \quad (32)$$

Equation 32 describes the film-thickness variation around the periphery of the sphere. Both the effect of subcooling in the bulk liquid and thermal radiation on the film thickness are included. Solution of Eq. 32 for film thickness would allow the calculation of the heat transfer from the sphere surface during film boiling to a surrounding fluid. The total heat-transfer rate would become the direct sum of the heat transferred due to conduction and that due to thermal radiation.

For highly subcooled liquids, such as sodium at less than 950°F and at atmospheric pressure, the contribution of the vapor film to the net energy exchange between the sphere and the surrounding liquid can be neglected. For a tantalum sphere which has relatively low emittance, the effect of radiation heat transfer on the vapor film thickness can also be neglected. Even if the emittance of the sphere surface were equal to unity, the contribution of radiative heat transfer compared to the heat transferred into the bulk of the subcooled liquid sodium amounts to less than 10% of the total heat-transfer

rate. Consequently, the second term on the left and the first term on the right of the Eq. 32 can be neglected. Equation 32 can then be written as

$$\frac{k\Delta T}{\delta} = \frac{k_\ell}{\sqrt{\pi}} \frac{\sin^2 \phi}{\sqrt{M\eta}} \Delta T_B. \quad (33)$$

The solution of Eq. 33 for the film thickness gives

$$\delta = \frac{k\Delta T}{k_\ell \Delta T_B} \frac{\sqrt{\pi M \eta}}{\sin^2 \phi}. \quad (34)$$

A simpler result of the neglect of these two terms in Eq. 32 is due to the realization that the vapor film thickness must adjust its thickness to accommodate the heat transferred through it and subsequently into the subcooled liquid. The heat-transfer rate from the sphere can now be found by calculating the amount of heat transferred from the liquid-vapor interface into the bulk of the liquid. The following expression (which was derived in Appendix A) is obtained:

$$\frac{q}{A} = \frac{k_\ell}{\sqrt{\pi M}} \frac{\sin^2 \phi}{\sqrt{\eta}} \Delta T_B. \quad (35)$$

The basic theory developed for this study is presented in this chapter and Appendix A.

CHAPTER IV

EXPERIMENTAL APPARATUS

The experimental apparatus used in this investigation is shown in Fig. 5. The basic parts of the apparatus were a tank to contain liquid sodium, a tantalum sphere with a thermocouple imbedded in it supported by a tantalum support tube, and a motor to propel the heated sphere through the liquid-sodium pool. The basic apparatus in its initial stage of construction is shown in Fig. 6. The tank and support structure was constructed of Type 304 stainless steel. The tank was semicircular, with a radius of 10 in., and 4 in. wide. The tank was designed in the semicircular shape to minimize the amount of sodium required for the experiment.

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Fig. 5. Sodium Heat-transfer Apparatus, Ready for Operation

Fig. 6

Sodium Heat-transfer Apparatus,
Early Stage of Construction

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The motor used in this experiment to move the sphere through the sodium pool was a Photocircuits Corp. Model 488 printed-circuit d.c. motor. One requirement for this experiment was a constant linear sphere velocity as the sphere moved through the sodium pool. This printed-circuit motor has its armature windings "printed" upon a thin ceramic disc. Consequently the motor has very low internal inertia and relatively high angular acceleration. For the range of velocities in this experiment, the motor reached a constant angular velocity in 90° of rotation or less. At extremely low velocities, the velocity of the sphere varied slightly in its motion through the sodium pool. This variation never exceeded 10%. Whenever a variation occurred, an average value for velocity was used.

The motor was connected to a stainless steel shaft, which extended over the sodium tank and was supported by bearings contained in pillow blocks. These pillow blocks were mounted on Teflon shims to protect the bearings from overheating. During data runs, the motor was connected to the shaft by a simple sleeve connector. This provided a rigid connection and prevented the motion of the sphere from lagging behind the motion of the motor.

The motor was driven by an EICO Model 1064 d.c. power supply. The velocity of the motor is a function of the applied voltage and torque. The torque was applied by a friction brake, which is discussed in Section C of this chapter. For a particular voltage setting and applied torque, the velocity of the motor was reproducible to within 10%.

The sodium was heated from below by two Hevi-Duty 50 KTS flat electrical resistance heating units limited to heater temperatures up to 2190°F . These heaters were contained in an insulating box, which fit closely over the sodium tank, as shown in Fig. 7. The heaters were connected in parallel to a General Radio Type W20V d.c. (Variac) power supply.

108-9409

Fig. 7. Insulating Box for Sodium Tank Containing Electrical Heaters

A. Sphere-Thermocouple Assembly

A 1/2-in.-diam tantalum sphere was used as the test body in this experiment. Tantalum was chosen because of its high melting temperature (5425°F) and its relative ease in machining. The thermal properties⁵⁰ of tantalum are practically constant from room temperature to near its melting point.

Installed in this sphere was a tungsten-5% rhenium/tungsten-26% rhenium thermocouple. The wires of this thermocouple were 30 gage and were contained in a 0.062-in.-OD tantalum sheath. Surrounding the wires was vitrified beryllium oxide powder, which served as an insulator. The thermocouple bead was welded to the bottom of the sheath.

The method of fabrication was as follows: A tantalum sphere was machined on a special Hardinge lathe, and a billet of excess tantalum was left on the forward portion of the sphere. A 0.066-in.-diam hole was drilled to within 0.031 in. of the tangent line of the forward portion of the sphere. A 1/8-in.-OD tantalum tube was fitted into a countersunk hole in the rear of the sphere and electron-beam welded. The thermocouple was then inserted into the hole and was Heli-arc welded to the bottom of the hole by applying the arc to the outside of the sphere. The assembly, before welding, is shown in Fig. 8. After the thermocouple was welded, the sphere was again put into the lathe and the excess material was removed from the front portion of the sphere. The sphere-thermocouple assembly directly after welding is shown in Fig. 9. The finished product was practically mirror-smooth and is shown in Fig. 10 after being bent into the desired

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Fig. 8. Tantalum Sphere and Support Tube, before Welding

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Fig. 9. Sphere-Thermocouple Assembly, after Welding

108-9412

Fig. 10. Sphere-Thermocouple Assembly, Ready for Use

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Fig. 11. Cross Section of Sphere-Thermocouple Assembly

shape. A cross section of the sphere drawn to scale is shown schematically in Fig. 11. Attaching the thermocouple in this manner provided the best possible thermal contact between the sphere and the thermocouple.

The sphere-thermocouple assembly was attached to the shaft by a simple clamping device. The sphere support tube was clamped to a holder, which was, in turn, clamped to the

motor shaft. Figure 5 shows the completed apparatus ready for operation. The distance from the shaft axis to the sphere could be varied with this attaching device.

X-ray photographs were taken of the sphere in an effort to locate the thermocouple bead very precisely. However, because of the scattering effect of the tantalum at the curved forward portion of the sphere, the X-ray photographs were of poor resolution. Therefore, the location of the thermocouple had to be estimated from mechanical measurements.

B. Temperature-recording System

The output of the tungsten-rhenium alloy thermocouple inside the sphere was recorded with a high-frequency oscillograph (a Honeywell-Heiland Model 906A Visicorder). This instrument uses galvanometers to achieve accurate response to rapidly changing input signals. A mirror

suspended upon the galvanometer wires rotates as the input signal varies. The galvanometer is a current-sensitive device (as compared to potentiometric recorders, which are voltage-sensitive). The galvanometer used to record the sphere temperature was an M-1000 Heiland galvanometer with a natural frequency of 1000 cps. However, to damp this galvanometer adequately, it was placed in a resistance circuit, which reduced the frequency response to approximately 640 cps.

Because of the relatively low output of the tungsten-rhenium thermocouple, and the low sensitivity of the M-1000 galvanometer, the input thermocouple signal had to be amplified. An EICO Model A12 d.c. amplifier was used to drive the galvanometer. The amplifier also eliminated a problem that would have been present if the amplifier were not used. Since galvanometers are basically current-sensitive devices, the change of resistance with temperature of the thermocouple leads must normally be considered. However, the input impedance of the EICO amplifier is 10,000 ohms, so that small changes in source resistance are negligible in comparison to this large impedance.

A bias circuit was integrated into the temperature-recording system as shown schematically in Fig. 12. This bias circuit allowed the galvanometer trace to be deflected so that when the sphere was heated, a 1000°F temperature range was displayed on the 6-in. recording paper used in the Visicorder. This was done to minimize the effect of light-beam trace width on the magnitude of the recorded temperature. The light beam could be positioned anywhere on the paper with this circuit. The bias circuit was calibrated periodically during data runs to give an output of 50 mV at 0.5 mA and proved to be stable.

108-9665

Fig. 12. Schematic Diagram of Temperature-recording Circuit

Although the operating specifications for the electrical components indicated that the frequency response of the system would be adequate, a simple test for the response of the system was performed. An electrical

input of sufficient magnitude to give a galvanometer deflection of 5 in. was applied to the system in the form of a step-function input. The signal was applied with a simple push-type switch. The signal actually overshoot the correct value of the step input slightly and then assumed the correct value. The total time required by the amplifier-galvanometer system for the response to the complete step input was less than 5 msec. This was felt to be more than adequate for accurately recording the signals anticipated in this experiment.

A calibration circuit identical to the bias circuit was also included in the system so that spot checks of the system calibration could be made. Later it was found that the bias-circuit output was stable and the calibration circuit was used very little. To protect the M-1000 galvanometer from high current, two Zener diodes were placed back to back in parallel with the galvanometer.

C. Measurement and Control Instrumentation

The angular velocity of the motor was measured by a timing device located at the rear of the d.c. motor. All the measurement and control instrumentation for the motor was located at the rear of the motor to protect it from the high temperatures of the sodium pool. This instrumentation is shown in Fig. 13. The timing device consisted of a light source and a photoconductive cell separated by a Plexiglas wheel. The Plexiglas wheel was

108-9405

Fig. 13. Measurement and Control Instrumentation

perforated with drill holes at 15° intervals on a base circle 2.5 in. in radius. As the motor shaft turned, the timing wheel also turned. The face of the wheel was darkened, and as holes alternately moved between the light source and the photocell, the resistance of the cell decreased sharply.

108-9664

Fig. 14. Velocity-recording Circuit

The cell was contained in a circuit that also contained the recording device (see Fig. 14). The decrease in cell resistance caused a recording galvanometer to deflect. This galvanometer was contained in the Honeywell-Heiland Model 906A Visicorder described previously. Timing lines are placed on the record by special galvanometers. This device recorded the deflections that occurred as the motor moved the sphere through the sodium. The angular velocity was calculated by measuring the time between several galvanometer

deflections as the sphere moved through the sodium and dividing it into the total angle traversed in this time increment. The linear velocity of the sphere was then calculated by multiplying this calculated angular velocity by the measured distance from the motor-shaft axis to the center of the sphere.

Figure 13 also shows three microswitches mounted around the periphery of the motor. These microswitches were part of a relay circuit which provided for remote control of the motor and for the following modes of operation:

1. Forward motion from the sphere heating device, through the sodium pool, and to rest above the sodium pool. (This was the data-gathering mode of operation.)
2. Backward motion through the sodium pool to the sphere heater.
3. Backward motion through the sodium pool to a position above the sodium pool but outside the sphere heater.
4. Backward motion from the position described in mode 3 to the sphere heater.

The control panel for this circuit is shown in Fig. 15. This photograph also shows the power supply for the d.c. motor, the Honeywell-Heiland Visicorder and timing unit, and the amplifier used in the temperature-measuring circuit.

The torque for the d.c. motor was provided by a friction brake shown in Fig. 13. This brake consisted of a spring-loaded felt-faced disc pressed against the Plexiglas timing wheel. The pressure exerted by the disc upon the wheel could be varied by turning a single nut mounted on the centerline

of the brake. As stated previously, by varying the torque with this friction brake, and varying the voltage to the motor, the motion of the sphere could be closely controlled.

108-9598

Fig. 15. Control Panel, Power Supply, Amplifier, Timing Unit, and Visicorder for Sodium Heat-transfer Apparatus

D. Sphere-heating Apparatus

The tantalum spheres used in this experiment were heated inductively in a copper coil connected to a Thermonic Model 1500 induction generator. Induction heating operates on the principle that a conductor in the electrical field of another current-carrying body has a current induced in it. The Thermonic generator used in this experiment operated at 250 kc. At this frequency, the induced current penetrates only a small portion of the outer skin of the sphere. The interior of the sphere must then be heated by conduction.

Three coil designs were used during this investigation. In general, the coils were elliptical in shape and built so that the sphere could freely move in and out. The first coil, shown on the right in Fig. 16, consisted of a single turn of copper plate. Copper tubing carrying cooling fluid was soldered to the outside of the coil. The sphere was very tightly coupled to this coil; i.e., the distance between the sphere and the sides of the coil was

very small. It was impossible to achieve high sphere temperatures with this coil. Arcing between the coil and the sphere surface occurred at relatively low generator power settings. A three-turn coil, which was not so tightly coupled, is also shown in Fig. 16. The third coil design, shown in Fig. 17, consisted of three turns of copper tubing and was more tightly coupled than the other three-turn coil. Both three-turn coils were capable of heating 1/2-in. tantalum spheres to well over 3600°F. Both coils were used during this experimental program. The loosely coupled three-turn coil greatly diminished the arcing problem.

Fig. 16
Induction Heating Coils Used
to Heat Tantalum Sphere

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Fig. 17. Heat-transfer Apparatus Showing Three-turn Induction Heating Coil

E. Glovebox and Auxiliary Equipment

Since sodium is highly reactive with both oxygen and water vapor, this experiment had to be performed in a dry, inert atmosphere. The apparatus described in the preceding sections of this chapter was contained in a two-module, argon-atmosphere glovebox shown in Fig. 18. Argon gas is continually circulated from the glovebox through a purification system. The gas in the system was continually sampled, and the oxygen and water-vapor content was determined. The purification system maintained both the oxygen and water-vapor content of the argon gas below 5 ppm.

108-9597

Fig. 18. Argon-atmosphere Glovebox

The purification system consists of a palladium catalyst bed and a molecular sieve. The molecular sieve removes water vapor from the circulating argon gas. If the atmosphere becomes high in oxygen content, hydrogen gas is added to the flowing gas before it passes through the palladium bed. The palladium bed serves as a catalyst to react hydrogen with oxygen to form water vapor. The gas then flows through the molecular sieve, which removes the water vapor. The purified gas then returns to the glovebox.

A coaxial cable was provided from the Thermonic induction generator into the glovebox. Other auxiliary equipment included a silicone oil recirculating system for cooling the induction heating coil, an air lock for

introducing materials into the box without contaminating the atmosphere, grounded electrical outlets inside the box, and a bubbler to provide a safety release for any sudden pressure increases inside the box.

Modifications to the system, made specifically for this experiment, include the installation of electrical lead-throughs for the motor control circuits, lead-throughs for an additional silicone oil recirculating system, which was used to cool the outside of the copper box that contained the electrical heaters for the sodium tank, and the introduction of a set of W-26% Re/W-5% Re thermocouple wire lead-throughs. Figure 19 shows the manner in which these auxiliary systems were introduced into the system.

108-9596

Fig. 19. Argon-atmosphere Glovebox Modifications

The sodium temperature was measured with a Chromel-Alumel thermocouple contained in a stainless steel thermocouple well inserted into the sodium pool. Another Chromel-Alumel thermocouple was attached to the outside of the sodium-tank copper heat-exchanger box. Both thermocouples were connected to Chromel-Alumel thermocouple lead-throughs provided in the glovebox. These thermocouples were connected to a Brown Elektronik 17 recorder.

The operating procedure for conducting the experimental studies with this equipment is given in the following chapter.

CHAPTER V

EXPERIMENTAL PROCEDURES

The experimental data collected in this investigation consisted of temperature-time traces of a thermocouple at a known location in a tantalum sphere as the sphere passed through a pool of molten sodium. The apparatus for obtaining these data was discussed in Chapter IV. Briefly, the data were obtained in the following manner: The sphere was heated to the desired initial temperature in the induction heater. The motor was then actuated and the heated sphere moved in a circular arc from the induction heating coil, through the sodium pool, and to rest above the sodium pool. The output of the thermocouple was continuously recorded during this sequence of events. The velocity of the sphere in its motion from the heating coil through the sodium pool was recorded simultaneously with the temperature of the sphere.

In addition to the experimental procedures associated with the acquisition and accuracy of the experimental data, procedures associated with safety requirements were followed.

A. Safety Procedures

Rigid safety standards were required for this experiment because of the potential danger of molten sodium coming in contact with oxygen or water. As stated in Chapter IV, all cooling systems used silicone oil as the primary coolant. Secondary heat exchangers were located outside the box, and water was used as the secondary coolant. The oil in these systems was checked periodically to insure that water had not leaked into the oil. Standard safety-glass panels were used for the glovebox windows.

In addition to the bubbler on the glovebox, a solenoid valve was set to release gas into an exhaust system if the pressure difference between the box and the surroundings exceeded 0.90 in. of water. The sodium tank and heaters were enclosed in a copper heat exchanger (described in Chapter IV) to prevent heating of the argon atmosphere by natural convection from the sodium tank and electrical heaters. Overheating of the argon gas could result in a pressure increase in the glovebox.

Another potential hazard in this experiment was the use of beryllium oxide as the insulating material in the W-26% Re/W-5% Re thermocouple. Beryllium oxide is very toxic, and very low concentrations in air are extremely dangerous. However, the BeO in the thermocouples was vitrified and was contained in a tantalum sheath. Whenever there was any danger of exposing BeO to the air, all work was done with rubber gloves inside a hood.

B. Thermocouple Calibration

The accuracy of the sphere temperature measurements was assured by calibration of the output of the tungsten-rhenium alloy thermocouples that were used in the tantalum spheres. This was done after they had been installed in the spheres. Because these thermocouples were to be used to measure relatively high temperatures, an optical pyrometer was used as the calibrating device.

A Leeds and Northrup Model 8622-C optical pyrometer was calibrated against a tungsten filament lamp certified by the National Physical Laboratory of England. Actually, three pyrometers were compared to the tungsten filament lamp, and the one that compared best was used as the secondary calibration device. Figure 20 shows the results of this comparison. All pyrometers compared favorably up to approximately 2200°C, where one of the pyrometers began to show erroneous readings. However, the other two pyrometers agreed very well with the certified lamp over the entire range of temperatures, and did not vary from the filament temperature by more than 10°C at any temperature.

108-9663

Fig. 20. Leeds and Northrup Pyrometer Calibration Curve.

A blackbody enclosure system for calibrating the thermocouples was designed and tested. This method was designed to eliminate the problem of

having to use tantalum emittance properties to compute the true sphere temperature. The apparatus used in the "blackbody" calibration is shown schematically in Fig. 21. The sphere was heated inside a cylindrical blackbody enclosure. The 0.75-in.-diam tantalum cylinder, which served as the blackbody enclosure, was heated by a circular induction heating coil and, in turn, heated the sphere by radiation. The electrical output of the thermocouple and the reading of an optical pyrometer were simultaneously recorded whenever equilibrium was reached. The cylinder was enclosed at the bottom, and the sphere was inserted into it from the top. A tight-fitting cover cut to conform to the shape of the sphere support tube formed the top of the blackbody enclosure. A 0.060-in.-diam hole was drilled in the bottom of the enclosure, and a right-angle prism was used to sight the optical pyrometer upon the hole. An additional hole was drilled in the side of the enclosure so that the opposing wall of the enclosure could be compared to the sphere surface. This ensured that the enclosure was at equilibrium before any calibration points were recorded. The calibration had to be done in an inert atmosphere because tantalum oxidizes readily when heated in air. The calibration apparatus was contained in the argon-atmosphere glovebox described in Chapter IV.

108-9662

Fig. 21. Schematic Diagram of Blackbody Calibration Apparatus

The glovebox window and the prism absorbed a small amount of the radiant energy passing through them. These effects were taken into account by actually measuring the difference in temperature by observing a tungsten filament lamp directly and then observing the lamp through the window glass and the prism. At high temperatures, these corrections become appreciable, as can be seen in Figs. 22 and 23.

The blackbody calibration data shown in Fig. 24 indicated sufficient discrepancy to warrant refinement in the calibration technique. Calibration points for one of the thermocouples measured by the blackbody method are shown in Fig. 24. Also shown in Fig. 24 are points taken by looking directly at the sphere surface and correcting the observed temperature for emittance effects. If a true blackbody existed, these two sets of points should agree rather closely. However, as can be seen, quite a discrepancy exists between the data sets. The emittance correction was also applied to the points presumably taken under blackbody conditions, and these points are

also plotted in Fig. 24. Actually, these points agree fairly well with the emittance-corrected points. The circular induction coil provided a uniform temperature distribution on the portion of the tantalum enclosure inside the coil. However, near the ends, where the cylinder extended outside the coil for support purposes, the temperature varied considerably. The enclosure apparently did not approximate blackbody conditions closely enough for this method to be used. Consequently, the emittance-corrected temperature technique was used in the calibration of sphere thermocouples.

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Fig. 22. Prism Temperature-correction Curve

108-9667

Fig. 23. Glovebox-window Temperature-correction Curve

Fig. 24
Comparison of Calibration Techniques

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The thermocouples used in this experiment were furnished by the Pyro Electric Co. and the Thermo-Couple Products Co. In general, the measured output of the thermocouples did not vary greatly from the manufacturer's data. A typical calibration curve for one thermocouple is shown in Fig. 25. This curve shows the calibration only up to approximately 1700°C. This thermocouple-sphere assembly was used for relatively low temperature data, and consequently the calibration was not carried to higher temperatures. Other thermocouples, however, were calibrated up to approximately 2000°C.

108-9668

Fig. 25. Typical Thermocouple Calibration Curve (Thermocouple No. 92)

C. Sphere-heating Difficulties

The major difficulty encountered in the acquisition of the experimental data was in heating the tantalum sphere to relatively high temperatures. In the first set of data runs in which a single-turn induction coil was used to heat the tantalum sphere, $\sim 2750^{\circ}\text{F}$ was the upper limit for attainable sphere temperature. Usually, before 2750°F was reached, arcing between the induction coil and the sphere occurred. Some of these arcs were quite violent, and the surface of the sphere was slightly pitted by them. When the arcing occurred, the induction generator had to be shut down. The reason for the arcing appeared to be ionization of argon gas between the coil and the sphere. Later it was found to be caused by the ionization of sodium vapor rising from the sodium pool surface upward between the coil and the sphere assembly. The arcing problem was never completely solved, but was minimized greatly by building a more loosely coupled induction coil.

To achieve more efficient sphere heating, a three-turn coil was designed. It was elliptical, and the sphere assembly was free to move in and out of the coil. This coil was somewhat more efficient than the single-turn coil, and a sphere could be heated to 3700°F with ease if sodium vapor were not present. However, when the sodium was molten and forming vapor, arcing was again prevalent.

A third coil (also elliptical, but much larger than the previous two coils) was designed and tested. A tantalum susceptor was placed inside this coil with the sphere inside the susceptor. It was hoped that the coil would heat the susceptor and the susceptor would, in turn, heat the sphere by radiation. This was done because, apparently, the sphere was absorbing sodium from the pool, and as it was reheated, it outgassed this sodium as great quantities of vapor, thereby enhancing the arcing problem. Sphere temperatures of approximately 2900°F were achieved by using this technique. Higher sphere temperatures necessary for this experiment would have required higher susceptor temperatures than were attainable. The low view factor between the susceptor and the sphere and the low emittance of the tantalum susceptor surface prevented efficient heating of the sphere.

After the failure of the susceptor heating method, the operation of the induction generator was checked. The generator was found to be operating at only two-thirds of its capacity because of two faulty fuses. After these fuses were replaced, the third coil design was efficient enough to heat the sphere to well over 3700°F without the use of the susceptor. Since the coil was so loosely coupled to the sphere, the problem of arcing was held to a minimum.

The problem of sphere contamination from exposure to liquid sodium was eventually solved by applying rigid cleanliness requirements to the sphere. If a sphere was used for no more than four data runs and then removed from the apparatus and cleaned with alcohol and abrasive cloth, it could be kept relatively clean. After the superficial deposition was removed from the sphere with abrasive cloth, the sphere was returned to the apparatus and reheated to approximately 3450°F. This removed any discoloration remaining on the sphere surface, and the sphere returned to a bright, shiny condition identical to its original as-fabricated condition.

A sphere that had been exposed to liquid sodium for several runs was sectioned, polished, and photomicrographed. For comparison, a sphere that had not been exposed was also photomicrographed. The results are shown in Fig. 26. The surface of the exposed sphere is relatively rough with apparent corrosion pitting of the surface, and with cracks extending down into the sphere to a depth of approximately 0.020 in. These cracks were probably caused by thermal stresses experienced by the sphere as it passed through the liquid sodium.

Tantalum is very susceptible to hydrogen absorption. Since opportunities were present for the tantalum sphere to absorb hydrogen during the experiments, this may have occurred and was contributing to the heating problem. The addition of hydrogen in the purification system provided the opportunity for hydrogen to be present in the system. Consequently, one of the spheres that had been used in experimental runs was subjected to a vacuum diffusion test for hydrogen. The hydrogen content proved to be rather high, approximately 100 ppm. As a check, a piece of tantalum from the rod stock that was used to make the sphere was also subjected to the same test. This sample actually showed higher hydrogen content (200 ppm) than the sphere. Therefore, the idea that hydrogen contamination of the tantalum sphere was contributing to the heating problems was discarded.

During the heating of the sphere in the induction heating device, the high frequency of the induced current seriously affected the sphere temperature recordings. The galvanometer deflected randomly and rapidly. The light beam appeared to be approximately 1/2 in. wide to an observer's eye, and was altered somewhat from the actual value of the thermocouple signal. As soon as the induction generator was shut down, the signal assumed its proper value in only 20 to 30 msec. This, therefore, did not seriously affect the operation of the equipment since the sphere moved from the induction coil before striking the sodium pool. The generator was also shut down before the sphere left the coil. However, this phenomenon did prevent duplicate runs from being made, since the initial temperature of the sphere could never be predicted with any degree of accuracy.

(a)

(b)

Fig. 26. Photomicrograph of Tantalum Sphere (90X).
(a) As-fabricated and (b) after several data runs.

D. Sodium Cleanliness

On one occasion, when the sodium was heated with relatively high oxygen content in the argon gas, a strange formation appeared upon the surface of the sodium pool. This formation was quite porous and in no way resembled the oxide film that normally formed on the sodium pool surface. The experiment was immediately terminated, and a sample of the formation was submitted for chemical analysis. It proved to be very high in silicon content. Apparently, silicone oil from the induction-heating-coil cooling system had been spilled inadvertently into the sodium tank while an induction coil was being changed. Precautions were taken in subsequent experiments to prevent spillage of silicone oil into the sodium, and no further problems of this kind were experienced.

Upon being removed from the tank, the sodium used for the first 25 data runs was found to be quite dirty. A sample of a coarse "sludge" near the bottom of the tank was submitted for chemical analysis. It was highly pyrophoric and extremely difficult to handle. Nothing extraordinary was reported from the analysis. With this in mind, the sodium in the tank was changed frequently, and a very clean sodium pool was maintained.

The sodium used in the experiment was furnished by the Baker and Adamson Co., and was reagent grade. Table I gives the limits of impurities that could have been present in the sodium as specified by the manufacturer. The percentage of other impurities such as SO_4 and nitrogen was less than that specified by the ACS Code 2188.

TABLE I. Limits of Sodium Impurities

Substance	Chlorine	PO_4	Iron	Heavy Metals
Limit of Impurity, %	0.0015	0.0005	0.001	0.0005

E. Experimental Technique

The operating procedure given in Appendix B was followed for the series of experimental runs with the sodium heat-transfer apparatus. This sequence of events usually took 5 to 8 hr. The actual data-taking occupied only a small portion of this time. The most time-consuming parts of the sequence were heating and cooling the sodium and achieving steady-state sodium conditions. Steady-state conditions were achieved by adjusting the power input to the sodium tank electrical heaters until the heat losses and heat input were balanced at the desired sodium temperature. The temperature of the sodium tended to drift, and constant vigilance was required to maintain a steady sodium temperature. This did not present a serious problem.

The recorded data obtained in these experiments consisted of thermocouple emf-time traces. The emf-time data were converted into temperature-time data in the following way: During data runs, the bias circuit described in Chapter IV was used to position the galvanometer mirror trace at the desired spot on the recorder paper. After the data run, a plot was made of emf versus scale deflection for the bias circuit setting used in the run. The values for this plot were obtained by replacing the thermocouple with a potentiometer and recording the emf values required to deflect the galvanometer mirror trace over the range of the recorder paper. One of these plots is shown in Fig. 27. By using a plot of this kind, data were converted into emf-time data. The thermocouple calibration curves such as the one shown in Fig. 25 were then used to convert the emf-time data into temperature-time data. The data-reduction methods are described in Chapter VI.

Fig. 27

Typical Emf-deflection Conversion Chart

108-9864

CHAPTER VI

EXPERIMENTAL RESULTS

A total of 87 experimental data runs were performed during this investigation. Measurements were made in 572 and 842°F (300 and 450°C) sodium. Runs 1 through 27 should be termed exploratory since uncertainty was present concerning the cleanliness of the sodium and the surface conditions of the tantalum sphere. Runs 28 through 31 were not useful since the thermocouple used in these runs exhibited poor response characteristics. The thermocouples used in Runs 32 through 57 also had poor response. However, enough information was extracted from these runs to enable their use to show reproducibility of data. Run 58 was also exploratory. Runs 59 through 87 were used to compute the heat flux versus surface temperature data presented in this chapter.

The experimental data are presented in Appendix C. Since Runs 1 through 27 were exploratory and not used in the calculation of heat fluxes, they are not presented. Runs 28 through 57 are presented in tabular form giving the initial temperature, the exit equilibrium temperature, and the time that the sphere was exposed to sodium. Runs 59 through 87 are presented as temperature-time data as reduced from the data records. The time of entry into the sodium pool is denoted as time = 0.

A. Methods of Data Reduction

The data were reduced by a technique developed by Stolz.⁵¹ This technique enables the calculation of the surface heat flux and surface temperature versus time for a sphere for which a temperature history at a known radial point is known. Initially the sphere must be considered isothermal.

The accuracy of this data-reduction method was determined by calculating surface heat flux and temperature when the heat flux and temperature histories were accurately known. The case considered was a 1/2-in.-diam sphere undergoing convective cooling described by the relation

$$q/A = h(T_w - T_B), \quad (36)$$

where T_w refers to the surface temperature of the sphere and T_B to the bulk temperature of the surrounding medium. For convenience, the heat-transfer coefficient, h , was considered constant. The analytical solution to this problem is

$$\frac{T - T_B}{T_i - T_B} = 2 \frac{R}{r^*} \sum_{n=1}^{\infty} \frac{\sin M_n - M_n \cos M_n}{M_n(M_n - \sin M_n \cos M_n)} e^{-M_n^2 \theta} \sin \left(M_n \frac{r^*}{R} \right) \quad (37)$$

where $\theta = \alpha t / R^2$, t is time, and M_n is defined by

$$1 - M_n \cot M_n = Nu.$$

The Nusselt number, Nu , is defined by

$$Nu = \frac{hR}{k}.$$

Figure 28 shows the surface temperature and the temperature at a radial point of $0.85R$ for the heat flux, which is also shown. The temperature history for the radial point $r = 0.85R$ was used as input for the data-reduction technique of Stolz. The results of the calculation are shown in Fig. 28 as discrete points. Both the heat flux and surface temperature computed by the Stolz method compare well with the analytical solution. The first calculated heat-flux point in time is somewhat low, but the method quickly converges to the correct values.

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Fig. 28. Comparison of Heat Flux and Surface Temperature Calculated by Stolz's Method to Analytical Solution

data was lost if longer time increments were used. Therefore it was felt inappropriate to use these longer increments.

The data-reduction difficulty was resolved by using two methods of data reductions and "piecing" the two methods together. The "constant h " analytical solution given in Eq. 37 was used as follows: The measured temperature history at the known radial location for a data run was plotted in dimensionless form as shown in Fig. 29. Run 71 has been used as an example. An h value was selected in an effort to "match" the measured temperature with the analytical solution. The measured temperature history

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Fig. 29. Matching Technique for Data Reduction in Initial Portion of Data Run

was matched for only a short time at the beginning of the data run, usually about 0.01 sec. Once the correct value for h was found, the heat flux and surface temperatures were calculated. The data-reduction method of Stolz was also applied to the data run. A time interval was selected for which the solution would show convergence. Figure 30 shows these calculated points for Run 71. The points calculated near the initial part of the data run are omitted. Figure 30 also shows the results of the "h matching" for Run 71. The solutions are "pieced" together by fairing curves through the heat-flux and surface-temperature points. The points calculated further in time by the Stolz method are generally considered more accurate than those calculated near the initial portion of the run. Consequently, these points and the high-temperature points calculated by the analytical solution are used primarily to define the heat-flux and surface-temperature histories for any data run for which the "piecemeal" solution was necessary. This piecemeal type of data reduction allowed the data to be reduced without great loss of detail. Both high- and low-temperature points could be obtained with this method. Computer programs using these methods are shown in Appendix D.

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Fig. 30. Data-reduction Method Using both Analytical Method and Stolz's Method

A third method of data reduction was based upon equilibrium consideration of the sphere temperature before and after encountering the sodium pool. This method was used primarily for reduction of data taken with thermocouples that exhibited poor response characteristics. Average heat flux at some average sphere temperature could be obtained for each data run by using this technique. The average heat flux from the sphere to the liquid sodium pool due to the sphere's encounter with the pool can be written as

$$(q/A)_{\text{avg}} = \frac{m\Delta H}{A\Delta t},$$

where m is the mass of the sphere, ΔH is the change in enthalpy, A is the surface of the sphere, and Δt is the time that the sphere was exposed to sodium. If the change in temperature of the sphere due to its encounter with the sodium pool is known, the change in enthalpy can be estimated accurately.

A technique was developed for obtaining the equilibrium temperature of the sphere after its exit from the sodium pool. The equilibrium temperature of the sphere before it struck the sodium pool was simply the temperature recorded by the imbedded thermocouple. This is a close estimate since cooling of the sphere in argon gas is slow and the sphere can be assumed to be at one temperature. The technique for obtaining the exit-equilibrium sphere temperature is shown in Fig. 31. It consists of plotting the measured temperature-time history for the data run to the point in time where the sphere has once again reached its equilibrium cooling rate. This was done by superimposing a cooling curve for the sphere in argon gas upon the measured temperature trace. The equilibrium point was taken as the point where

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Fig. 31. Technique for Obtaining Exit-equilibrium Sphere Temperature

the slopes of the curves first coincided. To correct for the amount of heat transferred from the sphere to the argon during the time between exit and the attainment of equilibrium, the argon cooling curve was plotted back to the time of exit. This temperature point was used as the sphere exit-equilibrium temperature.

The average heat flux obtained in this manner was referenced to an average temperature found by arithmetically averaging the entrance and exit-equilibrium temperatures.

B. Reproducibility of Results

Runs 32 through 57 were performed to show reproducibility of heat fluxes obtained from different sphere-thermocouple assemblies and two different batches of sodium. The thermocouples used in these runs exhibited poor response. One of these data records is shown in Fig. 32. These data (presented in Fig. 33) were reduced according to the equilibrium

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Fig. 32. Oscillograph Record; Poor Thermocouple Response

Fig. 33

Average Heat Flux vs. Average
Sphere Temperature, $U_{\infty} =$
10.0 ft/sec, $T_{Na} = 572^{\circ}\text{F}$

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technique described in Section A of this chapter. Although some experimental scatter is present, good agreement exists between data taken with different sphere-thermocouple assemblies. No effect of changing sodium was evident. All these data were taken in 572°F (300°C) sodium at a sphere velocity of approximately 10 ft/sec.

C. Presentation of Data

Data obtained in Runs 59-87 were used to calculate the heat flux versus sphere surface temperature data presented in this section. These data were taken with thermocouples that exhibited excellent response characteristics. An example of one of these data runs (Run 59) is shown in Fig. 34. Some of the data runs were duplications of previous runs. Consequently, only one run of a duplicated run was reduced.

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Fig. 34. Oscillograph Record; Good Thermocouple Response

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Fig. 35. Heat Flux vs. Surface Temperature;
 $U_{\infty} = 10.0$ ft/sec, $T_{Na} = 572^{\circ}\text{F}$;
 Bent Support Tube

Figures 35-38 present the data taken in 572°F (300°C) sodium. Figure 35 shows data obtained at a sphere velocity of approximately 10.0 ft/sec. The data in Fig. 35 were obtained with the thermocouple located near the stagnation point of the sphere. This was accomplished by bending the sphere support tube through a 90° angle. Consequently, the sphere was pushed through sodium with the support tube in the wake region. Figure 36 shows data also taken at approximately 10.0 ft/sec but with the support tube straight.

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Fig. 36. Heat Flux vs. Surface Temperature:
 $U_{\infty} = 10.0$ ft/sec, $T_{Na} = 572^{\circ}\text{F}$;
 Straight Support Tube

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Fig. 37. Heat Flux vs. Surface Temperature:
 $U_{\infty} = 6.0$ ft/sec, $T_{Na} = 572^{\circ}\text{F}$

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Fig. 38. Heat Flux vs. Surface Temperature:
 Various Velocities, $T_{Na} = 572^{\circ}\text{F}$

Consequently, the thermocouple location was near the 90° point in the sodium flow field. The data taken with the bent support tube are considered more representative of the actual heat-transfer process than those taken with a straight support tube. The straight support tube, located at the 90° position, could have affected the heat-transfer process occurring on the sphere surface. The bent support tube is located at the 180° position in the wake, where its disturbance of the heat-transfer process is not as appreciable as the disturbance by the straight support tube.

Figure 37 shows data taken at approximately 6.0 ft/sec. Runs 81-83 were taken with a bent support tube. Runs 64, and 66-68 were taken with a straight support tube. Higher initial sphere temperatures could be achieved by using the sphere-thermocouple assemblies with straight support tubes. This was because a much more efficient induction coil could be used to heat a sphere with a straight support tube. Essentially, this is the reason for taking data with spheres with straight support tubes.

Figure 38 shows data taken in 572°F (300°C) sodium at various sphere velocities. These data were taken to determine the effect of sphere velocity on the heat-transfer rate from the sphere to the liquid sodium.

Figure 39 shows data obtained in 842°F (450°C) sodium. These data are rather limited because they were extremely difficult to obtain. At 842°F , large quantities of sodium vapor were evolved from the surface of the sodium pool. The glovebox system was incapable of removing these quantities of

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Fig. 39. Heat Flux vs. Surface Temperature: $T_{Na} = 842^{\circ}F$

the faired heat-flux and surface-temperature curves for each data run. Heat-flux values at various surface temperatures for these data runs were selected and plotted on the graphs of heat flux versus surface temperature.

D. The Effect of Wetting

No noticeable effect of nonwetting was observed in the experiments performed in this study. A newly fabricated sphere was not wetted by the liquid sodium. As the sphere was heated and quenched in the sodium, wetting was observed.

E. Accuracy of Experimental Data

Several factors were involved in the overall accuracy of the heat-flux data acquired in this investigation. Since the data were calculated from temperature measurements, the accuracy of the data is directly dependent upon the accuracy of the measured temperatures. Associated with the temperature measurements were thermocouple-calibration errors, electronic-recording-system errors, and reading errors. The accuracy of the data is also affected by the accuracy of the data-reduction method.

Each thermocouple used in these experiments was calibrated after installation into the tantalum sphere. This calibration is estimated to be accurate to $\pm 1.5\%$. Possible errors arising from the electronic circuitry in the recording system were quite low and are estimated at not more than $\pm 1.0\%$. Reading errors could have contributed not more than $\pm 1.0\%$ error to the measured temperatures.

The accuracy of the analytical solution, Eq. 37, used in most of the data runs at the initial part of the run could be controlled by the number of

sodium vapor. Condensation of the sodium vapor in the argon atmosphere released energy to the gas, which was manifested as a temperature rise with accompanying expansion effects. Consequently the sodium pool had to be tightly covered until a data run was made. Even with this precaution, large quantities of vapor were accumulated. The scope of this portion of the experimental program was somewhat limited because of this difficulty.

These data points in Figs. 35-39 were obtained from

terms used in the infinite series. Five terms were used. This number of terms at the higher fluxes could have resulted in an error in the calculated heat fluxes of approximately $\pm 10\%$.

The accuracy of the Stolz data-reduction method differs from the accuracy of the analytical solution. The Stolz method involves a numerical error associated with an approximating equation. The analytical technique contains a series truncation error. The Stolz data-reduction method, generally used for low heat-flux values, is highly accurate at lower flux levels. The error associated with these flux levels should not exceed $\pm 3\%$.

The accumulation of these errors could result in overall maximum errors of approximately 20% at the highest heat-flux levels and 10% at the lower heat-flux levels. The accuracy of the data depends upon the value of the heat flux. These maximum error estimates are made by assuming that all the factors that contribute toward the overall error are combined in an adverse way. In all probability this would not occur, and the most probable error would fall between these maximum estimated errors and zero error. For example, if we take the most probable errors as one-half the maximum error, the experimental data can be considered to be 95 to 90% accurate, depending upon the flux level.

The limits of experimental scatter for the data presented in Figs. 35, 36, 37, and 39 does not exceed $\pm 27\%$ at any value of heat flux. The scatter limits actually tend to stay approximately constant over the entire heat-flux range. More data have been collected at the lower heat fluxes than at higher heat fluxes. Therefore the limits of scatter at the higher heat fluxes are based on a much smaller sample. In all likelihood the limits would increase if based on the same amount of data as the limits of scatter at the lower heat flux.

Experimental scatter can depend upon variations in the physical process being investigated as well as experimental errors. Thus, experimental scatter and experimental errors should not be confused. Experimental scatter can indicate the magnitude of the experimental error, however. The comparison of $\pm 27\%$ maximum experimental scatter to $\pm 20\%$ maximum estimated experimental error validates the estimated accuracy of the experimental data.

It is concluded that the experimental data obtained in this study are probably 85 to 90% accurate.

CHAPTER VII

DISCUSSION, CONCLUSIONS, AND RECOMMENDATIONS

The experimental results of this investigation, presented in Chapter VI, were compared to the results of the theory developed for film-boiling heat transfer from a sphere to a flowing liquid. This theory was developed in Chapter III. A theory in which it is assumed that vapor is prevented from forming at the sphere surface is developed in this chapter. The conclusions drawn from the comparison of these two theories to the experimental data are presented. Recommendations for further research are presented.

A. Comparison of Experimental Data and Film-boiling Theory

The theory developed in Chapter III used the assumption that film boiling would occur when an extremely hot spherical body moved through a pool of liquid sodium. An expression for the heat-transfer rate from the sphere surface into the bulk of the liquid was derived by using boundary-layer theory and by writing an energy balance on an element of the vapor film adjacent to the sphere surface. The resulting equation, Eq. 32, was simplified by neglecting the radiative contribution and the vapor-generation contribution to the net heat-transfer process. Even at sphere surface temperatures near 3600°F, the calculated radiative portion of the heat transfer from the sphere surface is two orders of magnitude less than the measured heat fluxes. The contribution of the generation of vapor at the liquid-vapor interface to the overall heat-transfer rate is also small. A comparison of this term in Eq. 32 to the term representing the heat flux from the liquid-vapor interface into the bulk of the liquid sodium shows that it is only about 1% of the latter term and can be neglected. The preceding statement applies to a sphere at 3472°F moving at 10 ft/sec through 572°F sodium. Thus, Eq. 32 is greatly simplified and becomes

$$q/A = \frac{k_l \sin^2 \phi}{\sqrt{\pi M \eta}} \Delta T_B, \quad (38)$$

where η is a function of geometry and M is defined by

$$M = \frac{2}{3} \frac{Ra}{U_\infty}.$$

Equation 38 shows that for one set of sodium conditions and one sphere velocity, the equation predicts a heat flux independent of sphere surface temperature. The experimental heat fluxes do not exhibit this trend. Figure 40 shows experimental points taken from Figs. 35 and 36 compared to a plot of Eq. 38. The sphere velocity is 10.0 ft/sec, and the temperature of the sodium is 572°F. Two cases for Eq. 38 are shown in Fig. 40. The line

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Fig. 40. Comparison of Experimental Data and Film-boiling Theory: $U_{\infty} = 10.0$ ft/sec, $T_{Na} = 572^{\circ}\text{F}$

that the heat flux over the back side of the sphere is negligible since the thermal conductivity of the sodium vapor as compared to that of liquid sodium is very low.

The total heat flux from the sphere, calculated by using Eq. 38, was found by a numerical summation. The surface of the sphere was split into 10° increments, and Eq. 38 was applied in each increment. The following expression was used for the total heat flux:

$$(q/A)_T = \frac{k_l \Delta T_B}{A(\pi M)^{1/2}} \sum_{j=1}^n \frac{\sin^2 \phi_j}{\eta_j} \Delta A_j,$$

where ΔA_j is the incremental sphere surface area associated with each 10° increment and A is the total area of the sphere. For the vapor film maintained to the 180° position, $n = 18$; for 90° separation, $n = 9$.

The properties of liquid sodium were evaluated at the arithmetic average of the liquid-vapor interface temperature and the sodium-bulk temperature. For this case, the liquid-vapor interface temperature was assumed to be the saturation temperature. The properties were taken from Dunning.⁵²

The film-theory results in Fig. 40 begin at a surface temperature of 1700°F , which is approximately 80°F above the boiling temperature of the sodium. Figure 40 shows that the theory predicts the correct order of magnitude for the heat fluxes. However, the experimental results indicate that heat fluxes are not independent of surface temperature as predicted by the theory. The plot of Eq. 38, assuming no separation, which is represented

representing higher heat flux was computed assuming that the vapor film was maintained to the 180° position of the sphere. It is also possible that the vapor film could "separate" from the sphere surface in a manner similar to that noted by Vliet and Leppert⁹ in forced-convection boiling of water around a circular cylinder. Hence the line representing the lower heat flux in Fig. 40 was computed assuming that "separation" of the vapor film occurred at the 90° position on the sphere. In reality, this assumption implies

by the upper line in Fig. 40, appears to agree more closely with the experimental data than the lower line representing the solution of Eq. 38 up to the 90° point, particularly for sphere temperatures sufficiently high that film boiling should occur.

Figure 41 compares Eq. 38 and experimental-data points taken from Fig. 37. The sphere velocity was 6.0 ft/sec, and the sodium temperature was 572°F. Figure 42 compares data obtained at a velocity of 10.0 ft/sec in 842°F sodium to Eq. 38, which was solved for similar conditions. These experimental data were taken from Fig. 38. These plots show the same trend as did Fig. 40.

Fig. 41
Comparison of Experimental Data
and Film-boiling Theory: $U_{\infty} =$
6.0 ft/sec, $T_{Na} = 572^{\circ}\text{F}$

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Fig. 42
Comparison of Experimental Data
and Film-boiling Theory: $U_{\infty} =$
10.0 ft/sec, $T_{Na} = 842^{\circ}\text{F}$

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Average heat flux and an integrated mean heat flux for most of the data runs were compared to resolve the question of whether "separation" occurred while the sphere moved through the liquid sodium. The average heat flux for each run was obtained by the so-called equilibrium manner described in Chapter VI. Briefly, the sphere inlet and exit equilibrium temperatures are found, and the relation

$$\left(\frac{q}{A}\right)_{\text{avg}} = \frac{m\Delta H}{A\Delta t}$$

is used to calculate the heat flux. The integrated mean heat flux was found by using the relation

$$\left(\frac{q}{A}\right)_{\text{mean}} = \frac{1}{\Delta t} \int_0^{\Delta t} \frac{q}{A}(t) dt.$$

This technique was applied to the heat-flux-versus-time plots such as Fig. 30. If "separation" occurred during an experiment, the average heat flux for the experiment should be approximately half the integrated mean heat flux. This is because the instantaneous heat flux is a local heat flux, and the assumption is made that the entire sphere behaves in a manner similar to the measured point. However, if most of the heat transfer takes place on the front half of the sphere, where the temperature measurements were made, this method would predict an overall mean heat flux from the sphere surface approximately twice as high as actually occurred. Table II presents the results of this comparison. The two heat fluxes, computed by different methods, are in excellent agreement for all the data runs that were reduced. Consequently, in the remainder of this work, it was assumed that "separation" did not occur, and all theoretical work was performed using the entire surface of the sphere.

TABLE II. Comparison of Average and Integrated Heat Fluxes

Run	Average Heat Flux, Btu/hr-ft ² x 10 ⁶	Integrated Mean Heat Flux, Btu/hr-ft ² x 10 ⁶	Run	Average Heat Flux, Btu/hr-ft ² x 10 ⁶	Integrated Mean Heat Flux, Btu/hr-ft ² x 10 ⁶
59	10.48	14.00	76	5.71	6.06
60	9.17	8.91	77	5.48	5.51
62	6.28	7.52	79	6.52	8.34
63	12.00	13.32	80	6.39	5.33
64	10.93	10.65	81	6.56	6.76
66	8.16	8.76	82	5.23	5.02
67	8.50	8.26	83	4.60	3.97
68	11.88	9.43	84	8.17	8.97
70	11.15	12.18	85	6.55	4.99
71	12.60	11.80	86	5.33	4.37
74	10.95	9.25	87	10.65	7.95
75	9.75	9.54			

It is instructive to calculate the thickness of the vapor film, δ , required to accomodate the heat fluxes calculated from Eq. 38. Writing the expression for film thickness, one obtains, from Eq. 34,

$$\delta = \frac{k_v}{k_\ell} \frac{\Delta T}{\Delta T_B} \frac{\sqrt{\pi M \eta}}{\sin^2 \phi}.$$

For the case considered in Fig. 40, that is, $U_{\infty} = 10.0$ ft/sec and sodium temperature = 572°F , the film thickness at the 90° position on a $1/2$ -in.-diam sphere at 3472°F is given by

$$\delta = 7.65 \times 10^{-6} \text{ in.}$$

Although the surface of the sphere was kept relatively smooth, it is anticipated that surface roughnesses much larger in magnitude than this calculated film thickness existed. Therefore, it is physically unrealistic that such a thin vapor film could exist adjacent to the sphere surface. This suggests that no vapor film is present at all. The combination of a large amount of subcooling in the liquid sodium and velocity of the sphere could prevent the formation of vapor at the sphere surface.

If no vapor film exists around the hot sphere then the problem becomes one of single-phase flow of liquid sodium around the sphere with convective heat transfer. This also introduces the possibility of superheating the liquid sodium in small layers near the sphere surface. The liquid sodium will try to assume the temperature of the contacting wall. Since sphere surface temperatures much higher than the boiling temperature of sodium are encountered, this would require superheating of the liquid sodium.

B. Comparison of Experimental Data and Superheating Theory

If superheating of liquid sodium does occur, very high temperature differences could exist between the sodium-sphere interface and the bulk of the sodium. Since the thermal conductivity of sodium is extremely high, tremendous quantities of energy could be transferred into the subcooled bulk of the sodium.

The theory developed in Chapter III for film boiling can be adapted for use in the superheating case. The vapor film need no longer be considered in an energy balance at the sphere surface. The net heat flux becomes the convective flux from the sphere surface into the bulk sodium. For this convective heat flux from the sphere surface into the bulk sodium, Eq. 38 will be used. However, the interface temperature can now assume values other than the saturation temperature. The equation becomes

$$\frac{q}{A} = \frac{k_{\ell} \sin^2 \phi}{\sqrt{\pi M \eta}} (T_{\ell} - T_B), \quad (39)$$

where T_{ℓ} is the temperature of the liquid sodium at the solid-liquid interface. The assumption of potential flow of sodium around the sphere will be retained.

Equation 39 also is based on the assumption of laminar flow of liquid over the sphere. The Reynolds number for these experiments varied from approximately 56,000 to 220,000. The corresponding Peclet-number range is from 190 to 1600. The Reynolds number and Prandtl number were evaluated at the arithmetic mean of the sphere surface temperature and the bulk sodium temperature.

The maximum heat flux for the superheating case would occur if the liquid sodium instantly assumed the temperature of the sphere surface as it contacted the sphere surface.

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Fig. 43. Comparison of Experimental Data and Superheating Theory: $U_{\infty} = 10.0$ ft/sec, $T_{Na} = 572^{\circ}\text{F}$, $T_l = T_w$

Figure 43 shows heat-flux values calculated from Eq. 39 plotted against surface temperature, assuming that $T_l = T_w$. Also shown in Fig. 43 are points taken from Figs. 35 and 36 representing experimental data. Although somewhat higher than the experimental data, the superheating theory correctly predicts the trend of the experimental data. The superheating theory predicts an increasing heat flux with increasing sphere surface temperature, which agrees with the experimental-data trend.

The theory can be brought into better agreement with the data if it is assumed that the expression for heat flux can be written as

$$\frac{q}{A} = K \frac{k_l \sin^2 \phi}{\sqrt{\pi M \eta}} (T_w - T_B). \quad (40)$$

The factor K could be thought of as a correction factor for the neglect of viscosity effects in the boundary layer, and for the situation in which the liquid sodium near the wall does not immediately assume the wall temperature. Figure 44 shows Eq. 40 plotted for $K = 0.54$ and $K = 0.60$. The plot for $K = 0.54$ agrees well with the lower-temperature data, whereas the plot for $K = 0.60$ agrees well with the higher-temperature data. It is apparent that K is a function of surface temperature. This suggests that the data perhaps could be correlated by an expression of the form

$$q/A = C(T_w - T_B)^n.$$

Fig. 44

Comparison of Experimental Data and
Superheating Theory: $U_{\infty} = 10.0$ ft/sec,
 $T_{Na} = 572^{\circ}\text{F}$

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C. Correlation of Data with Temperature Difference

The data were plotted on log-log coordinates to evaluate C and n . Figures 45-47 show these plots. Figure 45 represents data points taken from Figs. 35 and 36. These data could be correlated by the expression

$$q/A = 2.98 \times 10^4 (T_w - T_B)^{0.88},$$

where temperature is in $^{\circ}\text{F}$ and heat flux in Btu/hr-ft^2 . Figure 47 shows data points taken from Fig. 39 obtained at $U_{\infty} = 10.0$ ft/sec and $T_{Na} = 842^{\circ}\text{F}$. These data could be correlated by the relation

$$q/A = 4.94 \times 10^4 (T_w - T_B)^{0.77}.$$

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Fig. 45. Heat Flux vs. Temperature Difference:
 $U_{\infty} = 10.0$ ft/sec, $T_{Na} = 572^{\circ}\text{F}$

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Fig. 46. Heat Flux vs. Temperature Difference:
 $U_{\infty} = 6.0$ ft/sec, $T_{Na} = 572^{\circ}\text{F}$

Fig. 47
Heat Flux vs. Temperature Difference:
 $U_{\infty} = 10.0 \text{ ft/sec}$, $T_{Na} = 842^{\circ}\text{F}$

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The exponent, n , for heat flux measured in 842°F sodium differs considerably from that obtained from data taken in 572°F . Data at conditions other than those associated with Figs. 45-47 were sparse and were not plotted on log-log coordinates.

The reduction of the experimental data to an equation of the form

$$q/A = C(T_W - T_B)^n$$

provides a convenient tool for engineers and designers to calculate heat fluxes from spheres to liquid sodium with minimum effort. All that is required is the knowledge of the wall and bulk temperatures. These expressions are the result of curve fitting and not the result of theoretical considerations.

The variation of the exponent n with T_B suggests that the actual phenomena may be more complicated than was recognized in the development of the film theory and the superheat theory. However, the idealization employed in the development of the theories were necessary in order to obtain solutions for this complicated problem.

D. Effect of Velocity

Figure 48 presents data obtained in 572°F sodium at various velocities. These measurements were made to determine the effect of sphere velocity upon the heat flux. The lines representing data at 3.48, 12.00, and 14.06 ft/sec were obtained by fairing curves through the data points representing the respective velocities in Fig. 38. The lines representing data at 10 and 6 ft/sec are plots of the correlating equations obtained from Figs. 45 and 46, respectively.

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Fig. 48. Heat Flux vs. Surface Temperature:
 $T_{Na} = 572^{\circ}\text{F}$, Various Velocities

to account for this result. Therefore the results at 14.06 ft/sec must be considered anomalous. The lines representing velocities of 6.0 and 10.0 ft/sec represent many individual data runs, whereas the lines for the other three velocities represent only single data runs. Consequently, the 6.0- and 10.0-ft/sec lines must be considered to represent statistically a much better indication of the effect of sphere velocity upon heat flux. The measurements at these two velocities indicate that the heat fluxes follow the square root of velocity dependence as predicted by both the film-boiling and superheat theories.

The fact that the heat flux depends upon the square root of velocity in both theories is caused by the assumption of potential flow of the liquid around the sphere. Vliet and Leppert⁵³ found that the heat flux from spheres to water varies with velocity to the 0.66 power. This value (0.66) was found from experimental data where the effects of viscosity were present. A similar variation of velocity dependence was not observed in this study. The scatter of the experimental data could mask any such variation. However, the data trends indicate that the assumption of potential (frictionless) flow is valid for the range of experimental conditions investigated in this study.

E. Conclusions and Recommendations

This investigation has provided a new insight into the basic mechanism of heat transfer from extremely hot spheres into highly subcooled liquids such as molten sodium. The combined effects of large degrees of subcooling and of motion of the sphere tend to prevent the formation of vapor at the surface of the sphere. A theory was developed in this study by assuming that liquid sodium could be in contact with a surface that is much hotter than the boiling temperature of sodium. The limiting case for maximum heat transfer from a sphere to liquid sodium can be calculated with this superheat theory by assuming that the sodium immediately assumes

Figure 48 shows the trend of increasing heat flux with increasing sphere velocity up to approximately 12.0 ft/sec. At high surface temperatures, the heat-flux curve for $U_{\infty} = 12.0$ ft/sec falls lower than that for $U_{\infty} = 10.0$ ft/sec. The variation between these curves is within the limits of experimental error, however. The heat-flux curve obtained at $U_{\infty} = 14.06$ ft/sec falls lower than even the heat flux curve for $U_{\infty} = 6.00$ ft/sec. The 20% maximum experimental error that could be present fails

the sphere surface temperatures as it contacts the surface. The heat fluxes calculated from the experimental data were in every case less than this limiting heat flux. Even better agreement with the experimental data could be achieved with a slight modification of the superheat theory.

The film-boiling theory represents only a first-order approximation of the actual physical process for heat transfer from a sphere to liquid sodium. The many restrictive assumptions necessary in the formulation of the film theory contribute to its inability to describe the actual heat-transfer mechanism. Furthermore, the extremely thin vapor-film thicknesses calculated from this theory support the superheat theory.

The superheat theory provides a much simpler tool than the film-boiling theory for calculating heat fluxes from heated surfaces into highly subcooled liquids such as sodium. The superheat theory is essentially one of nonboiling convective heat transfer from the heated surface to the flowing liquid. This case is much more amenable to analysis than the relatively complicated problem of solving two boundary-layer systems, which is necessary in the film-boiling problem.

The heat fluxes calculated from the experimental data could be correlated by an expression of the form

$$q/A = C(T_W - T_B)^n.$$

The values of C and n were determined for three sets of experimental conditions:

U_∞ , ft/sec	T_B , °F	C	n
10.0	572	2.98×10^4	0.88
6.0	572	2.23×10^4	0.88
10.0	842	4.94×10^4	0.77

An expression of this type represents a curve fit for the experimental data and is not the result of any theory. Consequently these values of C and n should be used only over the range of conditions established by the experiments.

This study indicates that very large quantities of energy can be transferred from extremely hot particles to highly subcooled sodium without vapor formation. However, superheating of thin layers of liquid sodium near the sphere could occur. Many small particles in close proximity with each other, dispersed in liquid sodium, could provide a matrix of highly superheated sodium. Vapor nucleation could result in vapor formation of such magnitude to create damaging pressure pulses in a reactor core that is undergoing a destructive power excursion. Vapor nucleation would depend

upon the number and spacing of particles, their total energy dissipation, and the energy storage capability of the surrounding coolant. A study of these effects upon energy transfer and pressure generation would be of value in determining the consequences of a destructive nuclear excursion.

In conclusion, it can be stated that:

1. A valuable set of experimentally obtained heat fluxes from spheres to liquid sodium has been obtained in this study. To the author's knowledge no other data of this type have been obtained. The demonstrated accuracy of the data is good.

2. The superheat theory developed in this study provides a new, simpler technique for the calculation of heat transfer from extremely hot particles moving through highly subcooled fluids of high thermal conductivity.

3. The experimental and theoretical work performed in this study indicate that large amounts of energy can be transferred from a hot spherical particle to highly subcooled liquid sodium without vapor being formed at the sphere surface.

4. The film-boiling theory developed in this study represents a first-order approximation of the experimental data.

5. The experimental data were put into the simple form,

$$q/A = C(T_w - T_B)^n,$$

for use by reactor safety analysts. Heat fluxes can be calculated by knowing only the sphere-surface and bulk-sodium temperatures and the sphere velocity.

The following recommendations for possible future investigations are suggested as a result of the present study:

1. Further experimental work will be required to verify the absence of vapor around a hot sphere passing through highly subcooled liquid sodium.

2. Further experimental work should be performed to obtain the effect of subcooling upon the heat flux by making measurements in sodium up to its saturation temperature.

3. The effect of velocity should be more thoroughly investigated by performing experiments over a wider range of velocities. The effect of sphere diameter upon heat flux should be obtained.

4. More refined theoretical studies of the problem of heat transfer from an extremely hot sphere into liquid sodium are recommended. In particular, an analysis relaxing the assumption of constant fluid properties is needed. In general, better methods for treating flow around spheres are needed.

APPENDIX A

Solution to Liquid and Vapor Boundary-layer Equations1. Liquid Boundary Layer

The reduced energy equation for liquid flowing around a spherical vapor space can be written as

$$u_{r^*} \frac{\partial T}{\partial r^*} + u_{\phi} \frac{1}{r^*} \frac{\partial T}{\partial \phi} = \alpha \frac{\partial^2 T}{\partial r^{*2}} \quad (A1)$$

Equation A1 is written in spherical coordinates (r^*, ϕ, θ^*) by assuming axial symmetry and neglecting conduction in the ϕ -direction as compared to convection in that direction. It is also assumed that heat is transferred in a very thin liquid layer. The velocities, u_{r^*} and u_{ϕ} , represent velocity in the radial and angular directions, respectively. T is temperature, and α is the thermal diffusivity of the liquid.

The boundary conditions applicable to the case of liquid overriding a vapor film are given by

$$\begin{aligned} T &= T_B, & r^* &= \infty, & \phi &\geq 0; \\ T &= T_{\text{sat}}, & r^* &= R, & \phi &> 0; \\ T &= T_B, & \infty &\geq r^* \geq R, & \phi &= 0; \end{aligned} \quad (A2)$$

where T_B represents the temperature in the bulk liquid flowing past the spherical region, T_{sat} is the saturation temperature of the liquid, and R is the radius of the spherical region. If the vapor-film thickness is assumed thin, the radius of the solid sphere may be used for R .

Defining y by $y = r^* - R$, one may write

$$u_{r^*} = -3U_{\infty} \frac{y}{R} \cos \phi; \quad u_{\phi} = (3/2)U_{\infty} \sin \phi.$$

Transformation variables can be defined as

$$\begin{aligned} \Delta T &= T - T_{\text{sat}}, \\ \psi &= y \sin^2 \phi, \end{aligned}$$

and

$$\eta = \int_0^{\phi} \sin^3 \phi \, d\phi. \quad (A3)$$

Equation A1 can now be written as

$$\frac{\partial \Delta T}{\partial \eta} = M \frac{\partial^2 \Delta T}{\partial \psi^2}, \quad (\text{A4})$$

where M is defined by

$$M = (2/3)R\alpha/U_\infty.$$

The boundary conditions (A2) transform into

$$\begin{aligned} \Delta T &= T_B - T_{\text{sat}}, & \psi &= \infty, & \eta &\geq 0; \\ \Delta T &= 0, & \psi &= 0, & \eta &> 0; \\ \Delta T &= T_B - T_{\text{sat}}, & \infty &\geq \psi \geq 0, & \eta &= 0. \end{aligned} \quad (\text{A5})$$

Equation A4 and the associated boundary conditions constitute a system of equations identical to the heat-conduction equation for which solutions are known. A solution is

$$\frac{T - T_B}{T_{\text{sat}} - T_B} = \text{erfc}\left(\frac{\psi}{2\sqrt{M\eta}}\right), \quad (\text{A6})$$

where the complementary error function is defined by

$$\text{erfc}(a) = 1 - \text{erf}(a) = 1 - \frac{2}{\sqrt{\pi}} \int_0^a e^{-\lambda^2} d\lambda.$$

The heat flux into the liquid from the liquid-vapor interface can be found by differentiating Eq. A6. The heat flux can be written as

$$\frac{q}{A} = -k \left(\frac{\partial T}{\partial y} \right)_{y=0}. \quad (\text{A7})$$

The temperature gradient $\partial T / \partial y$ can be written as

$$\frac{\partial T}{\partial y} = \frac{\partial T}{\partial \psi} \frac{\partial \psi}{\partial y} + \frac{\partial T}{\partial \eta} \frac{\partial \eta}{\partial y}.$$

However, $\partial \eta / \partial y$ is zero, and the gradient becomes

$$\frac{\partial T}{\partial y} = \frac{\partial T}{\partial \psi} \frac{\partial \psi}{\partial y},$$

or alternately

$$\frac{\partial T}{\partial y} = \frac{\partial \Delta T}{\partial y} = \frac{\partial \Delta T}{\partial \psi} \frac{\partial \psi}{\partial y}.$$

The term $\partial \Delta T / \partial \psi$ is evaluated as

$$\frac{\partial \Delta T}{\partial \psi} = - \frac{\Delta T_B}{(\pi M \eta)^{1/2}},$$

where

$$\Delta T_B = T_{\text{sat}} - T_B,$$

and

$$\frac{\partial \psi}{\partial y} = \sin^2 \phi.$$

Therefore the heat flux into the bulk liquid from the liquid-vapor interface becomes

$$\frac{q}{A} = \frac{k_\ell \Delta T_B \sin^2 \phi}{(\pi M \eta)^{1/2}}. \quad (\text{A8})$$

2. Vapor Boundary Layer

The momentum and energy-conservation equations for a vapor film flowing around a sphere can be written as

$$u \frac{\partial u}{\partial x} + v \frac{\partial u}{\partial y} = -\frac{1}{\rho} \frac{\partial p}{\partial x} + \frac{\mu}{\rho} \frac{\partial^2 u}{\partial y^2}, \quad (\text{A9})$$

and

$$u \frac{\partial T}{\partial x} + v \frac{\partial T}{\partial y} = \alpha \frac{\partial^2 T}{\partial y^2}, \quad (\text{A10})$$

where u and v are the components of vapor velocity in the x -direction (along the sphere surface) and the y -direction (normal to the sphere surface), respectively. The pressure at any point in the vapor film is represented by p . The boundary conditions applicable to this case are

$$\begin{aligned} y = 0, \quad u = 0, \quad T = T_w, \\ y = \delta, \quad u = U(x), \quad T = T_{\text{sat}}, \end{aligned} \quad (\text{A11})$$

where δ is the film thickness and $U(x) = (3/2)U_\infty \sin \phi$. By assuming that inertia effects and energy convection in the vapor boundary layer are negligible, one can write Eqs. A9 and A10 as

$$\frac{\partial p}{\partial x} = \mu \frac{\partial^2 u}{\partial y^2} \quad (\text{A12})$$

and

$$\frac{\partial^2 T}{\partial y^2} = 0. \quad (\text{A13})$$

The solution to Eq. A13 can be written as

$$T = C_1 y + C_2.$$

Evaluations of the constants C_1 and C_2 from the boundary conditions give

$$T = T_w + (T_{\text{sat}} - T_w) y/\delta.$$

By rearranging this equation, we obtain

$$\frac{T - T_{\text{sat}}}{T_w - T_{\text{sat}}} = 1 - \frac{y}{\delta}. \quad (\text{A14})$$

The temperature distribution in the vapor film turns out to be linear.

The solution to Eq. A12 can be written as

$$u = \frac{1}{2\mu} \frac{\partial p}{\partial x} y^2 + C_3 y + C_4.$$

C_4 is zero from the condition that $u = 0$ at $y = 0$. Evaluation of the constant C_3 gives

$$u = \frac{3}{2} U_\infty (\sin \phi) y/\delta + \frac{1}{2\mu} \frac{\partial p}{\partial x} (y^2 - y\delta). \quad (\text{A15})$$

If it is assumed that the pressure in the liquid layer is "impressed" upon the vapor layer, the pressure-gradient term $\partial p/\partial x$ can be replaced by a velocity-gradient term. Neglecting height changes, Bernoulli's equation for the liquid is

$$p_0 = p + (1/2)\rho U^2 = \text{constant}.$$

By taking the derivative with respect to x , we obtain

$$\frac{\partial p}{\partial x} = -\rho U \frac{\partial U}{\partial x},$$

where U is the velocity of the liquid at the liquid-vapor interface. Therefore $\partial p / \partial x$ can be written as

$$\frac{\partial p}{\partial x} = -\left(\frac{3}{2}\right)^2 \frac{\rho U_{\infty}^2}{R} \cos \phi \sin \phi.$$

Substitution of this result into Eq. A15 gives

$$u = \frac{3}{2} U_{\infty} \sin \phi \left[\frac{y}{\delta} + \frac{3}{4} \frac{\rho U_{\infty}}{\mu R} (\cos \phi)(y\delta - y^2) \right]. \quad (\text{A16})$$

Equation A16 represents the velocity at any point in the vapor film.

APPENDIX B

Experimental Technique

The sequence of events followed when the heat-transfer apparatus was used to obtain data is as follows:

1. Turn on all electrical equipment to allow for warmup.
2. Check the water and oxygen content of the argon atmosphere. Normally these should be approximately 5 ppm.
3. Check for water in the oil reservoir for the sodium-tank cooling system.
4. Turn sodium-tank cooling-oil recirculating system on.
5. Check the thermocouples monitoring the sodium temperature and the sodium-tank heat-exchanger temperature to see that they are in place and connected.
6. Check the pressure in the glovebox.
7. Turn the electrical heaters on, and begin heating the sodium.
8. As the sodium is being heated, continually check the pressure in the box. (A good indication of a sudden pressure rise in the box is the manner in which the gloves hang. If they stand out rigidly, the pressure is high.) Also, monitor the temperature of the sodium-tank heat exchanger. If this temperature exceeds 100°C, turn on the cooling water to the oil-water heat exchanger.
9. Remove any oxide film from the top of the sodium pool.
10. Allow the sodium to reach the desired steady-state conditions.
11. Check the data-recording system for readiness.
12. Turn on the oil-recirculating system for the induction heating coil.
13. Check the induction-heating-system flowmeter for oil flow and also for any indication of oil-water mixing.
14. Position the sphere in the induction heating coil.
15. Heat the sphere to the desired temperature (observed visually on the Visicorder galvanometer system).

16. Actuate the motor, and take the data run.
17. Check the pressure in the glovebox.
18. Repeat steps 9, 10, 11, 14, 15, 16, and 17 until the required data range is covered.
19. When data taking is complete, turn the electrical sodium-tank heaters off. Increase the cooling-water rate to the sodium-tank oil-cooling system so that maximum cooling can occur.
20. When the induction-coil oil system has reached approximately room temperature, turn off the pump.
21. After 5 min, shut down the induction generator.
22. Monitor the sodium temperature until the sodium has solidified.
23. Turn off the sodium-tank oil-recirculating system and cooling water.

APPENDIX C

Experimental Temperature-Time Data1. Experimental Reproducibility Data

Run	Velocity, ft/sec	Sodium Temp, °F	T. C. No.	Temp In, °F	Temp Out, °F	Time, sec
28	10.36	577	TCP #1	3434		
29	9.83	572	TCP #1	3020		
30	6.82	572	TCP #1	3326		
31	6.04	572	TCP #1	3317		
32	11.50	572	TCP #3	3317		
33	10.70	568	TCP #3	3542	2174	0.0729
34	11.30	572	TCP #5	3427	2290	0.0655
35	11.30	572	TCP #5	3308	2201	0.0655
36	10.15	577	TCP #5	3056	2035	0.0729
37	10.60	568	TCP #3	3398	2183	0.0770
38	10.25	577	TCP #3	2967	1931	0.0785
39	10.00	572	TCP #3	2957	2021	0.0778
40	10.00	570	TCP #3	2732	1832	0.0778
41	10.00	572	TCP #3	2264	1616	0.0778
42	9.80	574	TCP #5	3416	2048	0.0793
43	9.92	572	TCP #5	2480	1654	0.0783
44	10.70	574	TCP #5	2111	1544	0.0726
45	10.10	574	TCP #3	3389	2237	0.0770
46	10.26	572	TCP #3	2948	1976	0.0758
47	10.45	572	TCP #3	2597	1841	0.0744
48	10.45	576	TCP #3	2228	1652	0.0744
49	10.69	570	TCP #5	3384	2336	0.0735
50	10.10	572	TCP #5	2660	1814	0.0770
51	10.30	577	TCP #5	3074	2062	0.0755
52	10.45	577	TCP #3	3074	2147	0.0665
53	9.60	572	TCP #3	2786	1895	0.0724
54	10.45	572	TCP #3	2417	1787	0.0665
55	9.50	576	TCP #5	3155	2120	0.0732
56	10.80	570	TCP #5	2750	1934	0.0644
57	10.45	572	TCP #5	2426	1792	0.0664

2. Instantaneous Temperature-Time Data

Run 59 Date 7/19/66 Velocity 9.60 ft/sec Temp. Na 572°F

Thermocouple No. 93 Time in Sodium .115 sec T.C. Shape Straight

Thermocouple Position 0.828 Sphere Diameter 0.500 in.

Time (sec)	0.00	0.01	0.02	0.03	0.04	0.05	0.06	0.07	0.08
Temp (°F)	3182	3083	2804	2534	2318	2138	2174	1877	1778

Time (sec)	0.09	0.10	0.11	0.12	0.13	0.14	0.15	0.16	0.17
Temp (°F)	1688	1616	1548	1486	1450				

Time (sec)	0.18	0.19	0.20	0.21	0.22	0.23	<u>Min</u> .136	<u>Equil.</u> .7
Temp (°F)							1450	2183

Run 60 Date 7-19-66 Velocity 9.60 ft/sec Temp. Na 577°F

Thermocouple No. 93 Time in Sodium .115 sec T.C. Shape Straight

Thermocouple Position 0.828 Sphere Diameter 0.500 in.

Time (sec)	0.00	0.01	0.02	0.03	0.04	0.05	0.06	0.07	0.08
Temp (°F)	2759	2678	2444	2196	2030	1900	1787	1674	1587

Time (sec)	0.09	0.10	0.11	0.12	0.13	0.14	0.15	0.16	0.17
Temp (°F)	1515	1472	1427	1400	1368				

Time (sec)	0.18	0.19	0.20	0.21	0.22	0.23	<u>Min</u> .135	<u>Equil.</u> .7
Temp (°F)							1368	1742

Thermocouple Position 0.828 Sphere Diameter 0.500 in.

Thermocouple Position	0.828	Sphere Diameter	0.500	in.
-----------------------	-------	-----------------	-------	-----

Time (sec)	0.18	0.19	0.20	0.21	0.22	0.23	<u>Min</u> .142	<u>Equil.</u> 1.20
Temp (°F)							1535	1760

Run 71 Date 7/28/66 Velocity 10.45 ft/sec Temp. Na 570°F

Thermocouple No. 91 Time in Sodium 0.11 sec T.C. Shape Bent

Thermocouple Position 0.828 Sphere Diameter 0.500 in.

Time (sec)	0.00	0.01	0.02	0.03	0.04	0.05	0.06	0.07	0.08
Temp (°F)	3141	3092	2876	2660	2516	2354	2228	2107	1994

Time (sec)	0.09	0.10	0.11	0.12	0.13	0.14	0.15	0.16	0.17
Temp (°F)	2084	1814	1733	1688	1643	1634	1620		

Time (sec)	0.18	0.19	0.20	0.21	0.22	0.23	Min .15	Equil. ~.7	
Temp (°F)							1620	1778	

Run 72 Date 7/28/66 Velocity 10.1ft/sec Temp. Na 572°F

Thermocouple No. 91 Time in Sodium 0.116 sec T.C. Shape Bent

Thermocouple Position 0.828 Sphere Diameter 0.500 in.

Time (sec)	0.00	0.01	0.02	0.03	0.04	0.05	0.06	0.07	0.08
Temp (°F)	2678	2674	2570	2219	2075	1931	1832	1724	1634

Time (sec)	0.09	0.10	0.11	0.12	0.13	0.14	0.15	0.16	0.17
Temp (°F)	1562	1481	1418	1364	1328	1310	1292	1283	

Time (sec)	0.18	0.19	0.20	0.21	0.22	0.23	Min .162	Equil. 1.90	
Temp (°F)							1283	1670	

Thermocouple Position	0.828	Sphere Diameter	0.500	in.
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Time							<u>Min</u>	<u>Equil.</u>
(sec)	0.18	0.19	0.20	0.21	0.22	0.23	15	21.05
Temp							1778	1868
(°F)

Thermocouple Position 0.828 Sphere Diameter 0.500 in.

Time (sec)	0.18	0.19	0.20	0.21	0.22	0.23	<u>Min</u> .14	<u>Equil.</u> 1.13
Temp (°F)							1499	1724

Run 80 Date 8/22/66 Velocity 10.2 ft/sec Temp. Na 577°F
 Thermocouple No. 91 Time in Sodium .114 sec T.C. Shape Bent
 Thermocouple Position 0.828 Sphere Diameter 0.500 in.

Time (sec)	0.00	0.01	0.02	0.03	0.04	0.05	0.06	0.07	0.08
Temp (°F)	2016	1980	1868	1724	1607	1508	1414	1350	1292

Time (sec)	0.09	0.10	0.11	0.12	0.13	0.14	0.15	0.16	0.17
Temp (°F)	1247	1202	1166	1139	1126				

Time (sec)	0.18	0.19	0.20	0.21	0.22	0.23	Min	Equil.
Temp (°F)							.13	2.0
							1126	1256

Run 81 Date 8/22/66 Velocity 6.38 ft/sec Temp. Na 570°F
 Thermocouple No. 91 Time in Sodium .192 sec T.C. Shape Bent
 Thermocouple Position 0.828 Sphere Diameter 0.500 in.

Time (sec)	0.00	0.01	0.02	0.03	0.04	0.05	0.06	0.07	0.08
Temp (°F)	2669	2588	2457	2318	2156	2074	1863	1742	1652

Time (sec)	0.09	0.10	0.11	0.12	0.13	0.14	0.15	0.16	0.17
Temp (°F)	1580	1517	1454	1400	1355	1319	1292	1256	1220

Time (sec)	0.18	0.19	0.20	0.21	0.22	0.23	Min	Equil.
Temp (°F)	1202	1184	1166				.20	2.0
							1166	1382

Run 82 Date 8-22-66 Velocity 5.45 ft/sec Temp. Na 572°F
 Thermocouple No. 91 Time in Sodium 219 sec T.C. Shape Bent
 Thermocouple Position 0.828 Sphere Diameter 0.500 in.

Time (sec)	0.00	0.01	0.02	0.03	0.04	0.05	0.06	0.07	0.08
Temp (°F)	2426	2390	2300	2152	1980	1832	1706	1616	1544

Time (sec)	0.09	0.10	0.11	0.12	0.13	0.14	0.15	0.16	0.17
Temp (°F)	1454	1418	1364	1328	1292	1243	1220	1202	1166

Time (sec)	0.18	0.19	0.20	0.21	0.22	0.23	<u>Min</u> .23	<u>Equil.</u> ~ 2.01
Temp (°F)	1148	1112	1094	1076	1067	1058	1058	1238

Run 83 Date 8/22/66 Velocity 5.99 ft/sec Temp. Na 572°F
 Thermocouple No. 91 Time in Sodium .194 sec T.C. Shape Bent
 Thermocouple Position 0.828 Sphere Diameter 0.500 in.

Time (sec)	0.00	0.01	0.02	0.03	0.04	0.05	0.06	0.07	0.08
Temp (°F)	2057	2030	1922	1809	1652	1562	1418	1382	1328

Time (sec)	0.09	0.10	0.11	0.12	0.13	0.14	0.15	0.16	0.17
Temp (°F)	1274	1238	1202	1166	1148	1112	1085	1067	1049

Time (sec)	0.18	0.19	0.20	0.21	0.22	0.23	<u>Min</u> .21	<u>Equil.</u> ~ 2.0
Temp (°F)	1031	1013	1004	991			991	1134

Run 86 Date 8/25/66 Velocity 3.48 ft/sec Temp. Na 577°F
Thermocouple No. 91 Time in Sodium .187 sec T.C. Shape Bent
Thermocouple Position 0.828 Sphere Diameter 0.500 in.

Time (sec)	0.00	0.01	0.02	0.03	0.04	0.05	0.06	0.07	0.08
Temp (°F)	2363	2350	2264	2152	2012	1868	1733	1638	1562

Time (sec)	0.09	0.10	0.11	0.12	0.13	0.14	0.15	0.16	0.17
Temp (°F)	1499	1436	1373	1337	1292	1265	1238		

Time (sec)	0.18	0.19	0.20	0.21	0.22	0.23	<u>Min</u>	<u>Equil.</u>
Temp (°F)								1346

Run 87 Date 8/31/66 Velocity 14.06 ft/sec Temp. Na 572°F
Thermocouple No. 91 Time in Sodium .080 sec T.C. Shape Bent
Thermocouple Position 0.828 Sphere Diameter 0.500 in.

Time (sec)	0.00	0.01	0.02	0.03	0.04	0.05	0.06	0.07	0.08
Temp (°F)	2534	2498	2386	2237	2066	1904	1760	1661	1580

Time (sec)	0.09	0.10	0.11	0.12	0.13	0.14	0.15	0.16	0.17
Temp (°F)	1517								

Time (sec)	0.18	0.19	0.20	0.21	0.22	0.23	<u>Min</u> .10	<u>Equil.</u>
Temp (°F)							1454	1670

APPENDIX D

FORTRAN Computer Programs for
Data-reduction Methods

```

PROGRAM SP TR CL
C  SPHERE TRANSIENT COOLING
  DIMENSION X(20),F(100),G(100),DELQ(100),THETA(100),Q(100)
  DIMENSION TEMP(100)
C  READ IN PARAMETERS
  5 READ 1, ALPHA,RHO,CP,R,RHOST,EXPNO
    IF(EOF,60)999,17
  1  FORMAT (5F12,6,I4)
  17 READ 2,(X(I),I=1,15)
  2  FORMAT (5F12,6)
    READ 3,TEMP1,TIME,DELTM,NUM
  3  FORMAT (3F12,6,I6)
    READ4,(TEMP(I),I=1,NUM)
  4  FORMAT (6F12,6)
    Z = RHO * CP * R / 3.0
    P =(R*R/ALPHA) *(RHOST * RHOST/6.0 *.10)
    S =(R*R/ALPHA) * 0.06667
C  CALCULATE RESPONSE FUNCTION F
    F(1)=0.0
    A = DELTM
    DO 11 I=1,NUM
      K=I+1
      SUM = 0.0
      DO 10 J=1,15
        B =0.0
        W =SIN(X(J)*RHOST)/(RHOST *SIN(X(J)))
        Y =EXP(-X(J)*X(J)*ALPHA/(R*R) *A)/(X(J)*X(J)*ALPHA/(R*R))
        B = W*Y
        SUM = SUM +B
  10  CONTINUE
      H=A+P-.667*SUM
  15  F(K)=H
      A=A+DELTM
  11  CONTINUE
C  CALCULATE RESPONSE FUNCTION G
    G(1)=0.0
    A1=DELTM
    DO 20 I=1,NUM
      L=I+1
      SUM1 = 0.0
      DO 21 J=1,15
        B1=0.0
        B1 =EXP(-X(J)*X(J) *ALPHA/(R*R) *A1)/(X(J)*X(J) *ALPHA/(R*R))
        SUM1 = SUM1 +B1
  21  CONTINUE
      V=A1+S-.667*SUM1
  24  G(L)=V
      A1 =A1 + DELTM
  20  CONTINUE
C  CALCULATE DELQ(I)
    U=Z*(TEMP1-TEMP(1))/F(2)
    DELQ(1)=U
    DO 30 I=2,NUM
      A2=Z * (TEMP1 -TEMP(I))
      M=I-1
      L1=I+2

```

```

SUM2 = 0.0
DO 31 J=1,M
  B2=DELQ(J)*F(L1=J)
31 SUM2 = SUM2 + B2
  V1=(A2-SUM2)/F(2)
30 DELQ(I)=V1
C   CALCULATE Q(I)
  SUM4 = 0.0
  DO 50 I=1,NUM
    B4=DELQ(I)
    SUM4 = SUM4 + B4
50 Q(I)=SUM4
C   CALCULATE SURFACE TEMPERATURE
  DO 40 I=1,NUM
    SUM3=0.0
    I3=I+2
    DO 41 J=1,I
      B3=DELQ(J)*G(I3=J)
41 SUM3 =SUM3 + B3
    V2=SUM3/Z
40 THETA(I)=V2
  CONTINUE
C   PRINT OUT PARAMETERS
  NUM1=NUM+1
  PRINT 60 ,EXPNO
60 FORMAT (45H SPHERE TRANSIENT COOLING   EXPERIMENT NUMBER, 14 )
  PRINT 61,ALPHA,CP,R,RHOST
61 FORMAT (7H ALPHA=,F8.4,10X,4H CP=,F8.4,10X,3H R=,F8.4,10X,7H RHOST
1=,F8.4)
  PRINT 62 ,TEMPI,DELTM,NUM
62 FORMAT (7H TEMPI=,F10.4,5X,7H DELTM=,F10.6,5X,5H NUM=,15.//)
  PRINT 73, 2
73 FORMAT (10X,*Z=*,F12.6)
  PRINT 63
63 FORMAT(10X,*RESPONSE FUNCTION*)
  PRINT 64,(F(I),I=1,NUM1)
64 FORMAT (10X,6F12.6)
  PRINT 69
69 FORMAT (10X,*RESPONSE FUNCTION G*)
  PRINT 70, (G(I),I=1,NUM1)
70 FORMAT (10X,6F12.6)
  PRINT 65
65 FORMAT(10X,*SURFACE HEAT FLUX*)
  PRINT 66,(Q(I),I=1,NUM)
66 FORMAT (10X,6F12.6)
  PRINT 67
67 FORMAT(10X,*SURFACE TEMPERATURE DIFFERENCE*)
  PRINT 68, (THETA(J),J=1,NUM)
68 FORMAT (10X,6F12.6 )
  GO TO 5
999 STOP
END

```

```

PROGRAM SP TR CL
C TRANSIENT COOLING OF SPHERE WITH KNOWN COEFFICIENT
  DIMENSION B(100),SLM(100),THETA(20),C1(100),C2(100),C3(200)
  DIMENSION C4(100),C5(200)
  READ 1,ALPHA,RHO,H,R,THCON,DELTM
1  FORMAT (6F12,6)
  READ 2,M,N
2  FORMAT (2I6)
  READ 3,(B(I),I=1,M)
3  FORMAT (12F6,3)
  READ 4,(C1(I),I=1,M)
4  FORMAT (6F12,6)
  READ 5,(C2(I),I=1,M)
5  FORMAT (6F12,6)
  READ 6,(C3(I),I=1,M)
6  FORMAT (6F12,6)
  READ 7,(C4(I),I=1,M)
7  FORMAT (6F12,6)
  READ 8,(C5(I),I=1,M)
8  FORMAT (6F12,6)
  DO 31 L=1,4
    THT = ALPHA*DELTM/(R*R)
    S=THT
    B1 = H*R/THCON
    IF (B1=0.1) 999,100,100
100 CONTINUE
    IF (B1=31.0) 101,101,999
101 CONTINUE
    PRINT 102
102 FORMAT (10X,'SPHERE COOLING WITH KNOWN COEFFICIENT',/)
    PRINT 103,H
103 FORMAT (10X,'H=',F12,6)
    PRINT 104,ALPHA,THCON,DELTM
104 FORMAT (10X,'ALPHA=',F12,6,' THCON=',F12,6,' DELTM=',F12,6)
    PRINT 105,R,RHO,M,N
105 FORMAT (10X,'R=',F12,6,' RHO=',F12,6,' M=',I4,' N=',I4,/)
    DO 10 J=1,M
      A=B(J)
      IF (B1=A) 11,11,10
10 CONTINUE
11 CA = C1(J)-(B(J)-B1)*(C1(J)-C1(J-1))/(B(J)-B(J-1))
    CB = C2(J)-(B(J)-B1)*(C2(J)-C2(J-1))/(B(J)-B(J-1))
    CC = C3(J)-(B(J)-B1)*(C3(J)-C3(J-1))/(B(J)-B(J-1))
    CD = C4(J)-(B(J)-B1)*(C4(J)-C4(J-1))/(B(J)-B(J-1))
    CE = C5(J)-(B(J)-B1)*(C5(J)-C5(J-1))/(B(J)-B(J-1))
    DO 30 I=1,N
      G=RHO/R
      C CALCULATE REDUCED TEMPERATURE
      DO 20 K=1,2
        A1=(SINF(CA)-CA*COSF(CA))*EXP(-CA*CA*THT)*SINF(CA*G)/
1(CA*(CA-SINF(CA)*COSF(CA)))
        A2 = (SINF(CB)-CB*COSF(CB))*EXP(-CB*CB*THT)*SINF(CB*G)/
1(CB*(CB-SINF(CB)*COSF(CB)))
        A3=(SINF(CC)-CC*COSF(CC))*EXP(-CC*CC*THT)*SINF(CC*G)/
1(CC*(CC-SINF(CC)*COSF(CC)))
        A4=(SINF(CD)-CD*COSF(CD))*EXP(-CD*CD*THT)*SINF(CD*G)/

```

```

1(CD*(CD*SINF(CD)*COSF(CD)))
A5=(SINF(CE)*CE*COSF(CE))*EXPF(-CE*CE*THT)*SINF(CE*G)/
1(CE*(CE*SINF(CE)*COSF(CE)))
SUM(K)=A1+A2+A3+A4+A5
THETA(K)=2,0*SUM(K)/G
G=1,0
20 CONTINUE
PRINT 21,1,THETA(1)
21 FORMAT (10X,*THETA(1) AFTER*,14,*TIME INCREMENTS*,F12.6)
PRINT 22,1,THETA(2)
22 FORMAT (10X,*THETA(2) AFTER*,14,*TIME INCREMENTS =*,F12.6)
D=1
30 THT=THT+D*S
31 H=H+0,1
999 STOP
END

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