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POOL BOILING AND CONDENSATION
OF ELEMENTAL SELENIUM

by

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Table of Contents

<u>Section</u>	<u>Page</u>
I. Abstract	8
II. Introduction	10
III. Theory	12
A. Boiling	12
B. Condensation	18
IV. Corrosiveness of Selenium	23
A. Corrosion Tests	23
V. Apparatus	26
A. Boiling Section	26
B. Condensing Section	30
C. Assembled Apparatus	33
1. Air-Coolant Line	34
2. Inert Gas Line	38
3. Closed Chamber	43
4. Instrumentation	44
VI. Experimental Procedure	46
VII. Results	49
A. Preliminary Experiments	49
1. Water	49
2. Dowtherm A	49
B. Boiling of Selenium	57
1. Heat Transfer Data	57
a. Temperature Profiles in the Boiling Pool	60
b. Temperature Fluctuations	62

MASTER

c. Possible Effects of Corrosion	65
d. Pressure Drop across Auxiliary Condenser .	67
2. Empirical Correlating Equations	70
3. Semi-Empirical Correlating Equations	72
4. Possible Effect of Molecular Structure on Boiling Heat Transfer	80
C. Condensation of Selenium	87
1. Heat Transfer Data	87
2. Nusselt Equation	92
3. Condensation Coefficients	96
4. Possible Effect of Molecular Structure on Condensation Coefficients	100
VIII. Conclusions	105
Appendix A. Experimental Data	108
Appendix B. Calibrations	114
1. Electrical Meters	114
2. Heat Losses	114
3. Thermocouples	118
Appendix C. Calculations	121
1. Boiling Data	121
a. Sample Calculations - run S43	127
2. Condensation Data	129
a. Sample Calculations - run S43	135
Appendix D. Electrical Analogue Study	139
Appendix E. Physical Properties of Selenium	142
1. Vapor Pressure	142
2. Liquid Heat Capacity	142

3. Liquid Density	142
4. Vapor Density	145
5. Heat of Vaporization	145
6. Liquid Thermal Conductivity	149
7. Surface Tension	150
8. Liquid Viscosity	150
Appendix F. Structure of Selenium	153
1. Solid	153
2. Liquid	153
3. Vapor	157
Appendix G. Analysis of Errors	161
1. Boiling Data	161
2. Condensation Data	162
Nomenclature	165
References	168

List of Figures

<u>Figure</u>	<u>Page</u>
V-1. Boiling Section	27
V-2. Condensing Section	31
V-3. Schematic Diagram of Assembled Apparatus	35
V-4. Inside of Closed Chamber	36
V-5. Apparatus	37
V-6. Apparatus	37
V-7. Expansion Chamber	39
V-8. Top of Test Chamber	41
V-9. Bellows Assembly	42
VII-1. Heat Velocity vs. Temperature Difference for the Boiling of H ₂ O, Compared with Published Results at Atmospheric Pressure	51
VII-2. Heat Velocity vs. Temperature Difference for the Boiling of Dowtherm A	54
VII-3. Heat Velocity vs. Temperature Difference for Boiling Selenium	59
VII-4. Temperature Profiles in Selenium Boiling Pool	63
VII-5. Comparison of Experimental P_m and T_v with Vapor Pressure Curve for Selenium	68
VII-6. Constant in Empirical Correlating Equations vs. Pressure	73
VII-7. Correlation of Boiling Heat Transfer Data with the Rohsenow Equation	75
VII-8. Experimental and Predicted Pool Boiling Heat Transfer Data at Atmospheric Pressure	76
VII-9. Ratio of Predicted to Experimental Pool Boiling Heat Velocities vs Pressure, for $q_b = 50,000$ Btu/sq ft-hr	79
VII-10. Mechanism of Bubble Growth	80
VII-11. Ratio of Predicted to Experimental Heat Velocities vs Average Chain Length of Liquid Selenium, for $q_b = 50,000$ Btu/sq ft-hr	82

VII-12.	Nucleate Boiling Heat Transfer Data for Water-Ethylene Glycol Mixtures at Atmospheric Pressure (S6).....	84
VII-13.	Heat Velocity vs. Temperature Difference for Condensing Selenium	89
VII-14.	Condensation Coefficient vs. Saturation Pressure at Surface of Condensate Film	98
VII-15.	Possible Mechanism for Condensation of Selenium Vapor ..	101
VII-16.	Condensation Coefficient vs. Mole Fraction of Dimer in Vapor	103
B-1.	Results of Heat Loss Calibrations	115
C-1.	Location of Thermocouple Junctions in Copper Cylinder ..	122
C-2.	Schematic Diagram of Test Condenser	131
D-1.	Electrical Analogue Network	140
E-1.	Vapor Pressure of Selenium	143
E-2.	Liquid Density of Selenium	144
E-3.	Vapor Density of Selenium	146
E-4.	Low Range Vapor Density of Selenium	147
E-5.	Heat of Vaporization of Selenium	148
E-6.	Surface Tension of Selenium	151
E-7.	Viscosity of Liquid Selenium	152
F-1.	Average Chain Length of Liquid Selenium	155
F-2.	Weight Percent Seg Rings in Liquid Selenium	156
F-3.	Mole Fraction of $\text{Se}_2(\gamma_2)$ in Saturated Selenium Vapor ...	158
F-4.	Average Number of Atoms per Molecule in Saturated Selenium Vapor	160

List of Tables

<u>Table</u>		<u>Page</u>
VII-1.	Boiling Heat Transfer Data for Water	50
VII-2.	Boiling Heat Transfer Data for Dowtherm A	53
VII-3.	Condensation Heat Transfer Data for Dowtherm A	55
VII-4.	Boiling Heat Transfer Data for Selenium	58
VII-5.	Temperature Profiles in Selenium Boiling Pool	61
VII-6.	Effect of Power on P_g	69
VII-7.	Least Squares Equations for the Selenium Constant Pressure Pool Boiling Data	71
VII-8.	Comparison Between Predicted and Experimental Boiling Heat Velocities	77
VII-9.	Results of Condensation Experiments with Selenium	88
VII-10.	Comparison Between Experimental and Theoretical Condens- ing Heat Transfer Coefficients	93
VII-11.	Molecular Condensation Coefficients	97
A-1.	Experimental Data for Boiling Selenium-Temperature and Pressure Measurements	109
A-2.	Experimental Data for Boiling Selenium-Power Measure- ments	110
A-3.	Experimental Data for Condensing Selenium	111
A-4.	Experimental Data for Condensing Selenium	112
B-1.	Relative Calibrations of Thermocouples	119
F-1.	Mole Fractions of Molecular Species in Saturated Selenium Vapor	159

1. ABSTRACT

Experimental and correlating techniques have been developed and successfully employed to obtain and to analyze heat transfer data in the pool boiling and condensation of elemental selenium. The apparatus used to collect the heat transfer data contained a horizontal boiling plate and a horizontal condensing tube. All surfaces in contact with selenium were composed of stainless steel 316, which was found to have the best corrosion resistance towards selenium of several materials that were tested.

Boiling and condensation experiments were first carried out with water and Dowtherm A. The heat transfer data agreed closely with other available data obtained under similar conditions and with data predicted by standard correlations.

Boiling and condensation experiments with selenium were carried out over an absolute pressure range of 10 to 769 Torr, corresponding to a range of saturation temperatures from 437 to 687°C. Boiling heat velocities ranged between 20,000 and 100,000 Btu/sq ft-hr, while condensing heat velocities ranged between 18,000 and 52,000 Btu/sq ft-hr. The reproducibility and internal consistency of the heat transfer data were generally very good. Corrosion of the apparatus during the course of the experiments was shown to have insignificant effect on the heat transfer data.

An empirical correlating equation for the boiling heat transfer data was determined from a least squares fit of the data. The correlating equation,

$$q_b = (0.1956 P^{1/2} + 8.379 \times 10^{-3} P)(T_w - T_v)^{1.981},$$

calculated the boiling heat velocities with a probable error of +11.2% or -9%, where q_b , $(T_w - T_v)$, and P have the units of Btu/sq ft-hr, °F, and Torr, respectively.

The boiling heat transfer data were compared to values predicted by six widely tested semi-empirical correlations for nucleate pool boiling. The agreement between experimental and predicted heat transfer values was satisfactory for the data collected at pressures of approximately 760 Torr, but became progressively worse as the pressure was decreased. At the lowest pressure, 10 Torr, predicted heat velocities exceeded experimental heat velocities by around one to two orders of magnitude for each of the six correlations.

The condensing heat transfer data were correlated with the Nusselt equation for filmwise condensation on a horizontal tube. Experimental heat transfer coefficients were generally 40 to 50% of the values predicted by the Nusselt equation. The kinetic theory of condensation was employed to calculate condensation coefficients for selenium. The condensation coefficients were found to correlate well with pressure according to the equation

$$\log \sigma_c = -0.546 \log P_{sl} - 1.320,$$

where P_{sl} has the units of Torr.

The discrepancies between the experimental boiling and condensing heat transfer data and the data predicted by the standard correlations have been interpreted in terms of the complex structure of liquid and vapor selenium. Possible models for the boiling and condensation of selenium have been presented.

II. INTRODUCTION

The element selenium was discovered in 1817 by J.J. Berzelius. While it is widely distributed in the earth's crust, it occurs naturally only in very small concentrations. Practically all selenium is obtained as a by-product of the electrolytic refining of copper. Because of the expansion in copper mining in many countries, the supply of selenium is expected to remain ample in the future (E2).

Selenium is a very toxic substance (S1) that must be handled with caution, and above the melting point has generally been found to be quite corrosive. It has a melting point of 217°C , a normal boiling point of 686°C , an atomic number of 34, and an atomic weight of 78.96. Elemental selenium exists in several allotropic forms. The liquid and vapor phases are composed of several molecular species.

At the present time, the principal use of selenium is as a semiconductor component in rectifiers. Possible future applications, which provided the motivation for this research program, are as a high temperature heating or cooling fluid with liquid/vapor phase change, and as a thermodynamic working fluid in thermal energy conversion. Selenium's low vapor pressure and high boiling point would appear to make it well suited for use as a thermodynamic working fluid in space power generation; since radiation is the only method of heat rejection in space and weight is a prime consideration in the design of space vehicles, the condenser must be operated at relatively high temperatures and low pressures. Selenium also has the advantage of being much less costly than certain other substances, such as cesium and rubidium, which have similar boiling points and whose applicability as heat transfer and thermodynamic working

fluids has been considered by NASA, AEC, and the Air Force. It is commercially available at \$10 per pound in 99.999% purity; cesium and rubidium are approximately \$400 per pound. Selenium's principal drawback would appear to be its corrosiveness to most commercial metals, though its higher freezing point might also be undesirable.

In spite of selenium's attractive features, its heat transfer properties have not been previously studied. The basic objectives of this research program were to obtain and to correlate heat transfer data in the boiling and condensation of elemental selenium, for possible thermal design purposes.

Selenium has two properties that have made it an interesting, but difficult, material to study: its complex structure in the liquid and vapor phases and its corrosiveness. At the beginning of this research program, corrosion tests were carried out on several materials in liquid selenium. Since stainless steel 316 was found to have the best corrosion resistance towards selenium of the materials tested, the apparatus was constructed in such a way that only stainless steel 316 came into contact with selenium. The good reproducibility of the heat transfer data showed that corrosion of the apparatus during the boiling and condensation experiments did not have significant effect on the heat transfer data. Both the boiling and condensing heat transfer data have been compared to predictions by standard correlations. Discrepancies between the experimental and predicted heat transfer data have been interpreted in terms of the molecular compositions of the liquid and vapor phases.

III. THEORY

A. Boiling

Nucleate pool boiling is characterized by a periodic growth and departure of bubbles at specific nuclei on the heating surface. When the temperature of the heating surface exceeds the saturation temperature of the liquid by a sufficient amount, bubbles grow rapidly in the superheated layer adjacent to the heating surface. When the bubbles are large enough, buoyant forces cause the bubbles to detach from the surface tension pull to the surface, and to rise into the bulk liquid. The high rates of heat transfer associated with nucleate boiling are generally believed to be due to bubble-induced agitation of the fluid next to the heating surface. In a photographic study of the nucleate boiling of water, Gunther and Kreith (G5) measured radial velocities of 8 to 15 fps for the growing bubbles, indicating the importance of bubble agitation in reducing the high thermal resistance of the superheated liquid film at the heating surface.

1. Bubble Nucleation

Nucleate boiling involves two separate processes that control the rate of heat transfer: bubble initiation or nucleation at the heated surface, and bubble growth there and later. Nucleation refers to the formation of a small bubble (nucleus) whose size is barely in excess of that required for thermodynamic equilibrium. For a spherical nucleus of radius r , the conditions for thermodynamic equilibrium at the vapor-liquid interface can be written:

$$P_v - P_l = \frac{2\sigma}{r}, \quad (\text{III-1})$$

where P_v , P_l , and σ represent the vapor pressure inside the nucleus, the pressure in the bulk liquid, and the surface tension, respectively. By combining Eq. III-1 with the Clausius-Clapeyron equation, the amount of superheat required for bubble nucleation is obtained:

$$T_v - T_s = \frac{2RT_s^2\sigma}{\lambda P_s r}. \quad (\text{III-2})$$

A nucleus of radius r will therefore undergo bubble growth when the superheat exceeds $T_v - T_s$ (11).

In perfectly clean systems with extremely smooth surfaces, nucleation may occur first within the bulk liquid. However, when typical heating surfaces are used, nucleation occurs at cavities on them. Corty and Faust (C11) found that surface roughness has a major influence on nucleate boiling heat transfer. Bankoff (B5) examined the mechanism of bubble nucleation and concluded that bubbles originate from vapor pockets that are trapped in cavities on the heating surface. Griffith and Wallis (G4) carried out nucleation experiments from carefully prepared cavities. The superheats that they measured agreed well with the superheats calculated with Eq. III-2 when r was taken to be the radius at the mouth of the cavity. They therefore concluded that the mouth diameter determines the superheat required to initiate bubble growth from a cavity.

2. Effect of Surface Roughness on Boiling Heat Transfer

Examination of Eq. III-2 reveals that rough surfaces, with relatively large cavities, should provide better nucleate boiling heat transfer than

smooth surfaces. Corty and Faust (C11), Berenson (B8), and several other investigators have found that rough surfaces do, in fact, yield less superheat and thus higher nucleate boiling heat transfer coefficients than do smooth surfaces. By varying his surface conditions from very smooth to very rough, Berenson obtained variations in boiling temperature differences up to 600% for a constant heat flux. However, the data of Corty and Faust and of Vachon (V1) show that surface conditions are extremely important for smooth surfaces (with rms roughness between 1 and 10 microinches), while they have considerably less effect for rougher surfaces (with rms roughness greater than approximately 10 microinches). For typical industrial surfaces, which generally have an rms roughness between 30 and 60 microinches, and for surfaces that are usually employed in boiling heat transfer experiments, surface conditions can generally be expected to affect the boiling heat transfer coefficient by no more than 25 to 50%.

3. Effect of Pressure on Boiling Heat Transfer

Farber and Scoria (F1), Cichelli and Bonilla (C7), and several other investigators have found that nucleate boiling heat transfer coefficients increase with pressure. This can be explained by considering the effect of pressure on the terms in Eq. III-2. It can be shown that, for a given value of r , the right side of Eq. III-2 decreases as the pressure is increased; thus, less superheat is required to activate each nucleation site. Nucleate boiling heat transfer coefficients must therefore improve on increasing the pressure.

4. Correlation of Nucleate Boiling Heat Transfer Data

Several semi-empirical equations have been developed to correlate nucleate boiling heat transfer data. Because of the difficulties inherent in characterizing surface conditions, most of these correlations neglect the effect of surface conditions on boiling heat transfer. Others, such as the correlation of Rohsenow (R3), do at least recognize the effect of surface conditions by including parameters that must be determined empirically for each fluid-surface combination.

In view of the significant effect that surface conditions have on nucleate boiling heat transfer, it is clear that any correlation that is to predict nucleate boiling heat transfer data with a high degree of accuracy must account for surface conditions. Nevertheless, several of the correlations that neglect surface conditions have been widely tested and found to predict nucleate boiling heat transfer data with reasonable accuracy.

In deriving an equation to correlate nucleate boiling heat transfer data, Forster and Zuber (F2) developed a theory for the growth of a vapor bubble in a superheated liquid. They obtained analytical expressions for the bubble radius and the bubble growth rate:

$$r = \frac{(T_w - T_s)\rho_l c_{pl}(\pi a_l t)^{1/2}}{\lambda \rho_v}, \quad (\text{III-3})$$

$$\frac{dr}{dt} = \frac{(T_w - T_s)\rho_l c_{pl}(\pi a_l)^{1/2}}{2\lambda \rho_v(t)^{1/2}}. \quad (\text{III-4})$$

The reliability of their analysis is shown by the fact that Eq. III-4

gives good agreement with bubble growth rates that have been measured by others, e.g. Dergarabedian (D1) and Gunther and Kreith (G5).

Since the high heat transfer rates in nucleate boiling, compared to non-boiling forced convection, are considered to be a consequence of the bubble-induced turbulence in the superheated liquid, Forster and Zuber reasoned that bubble dynamics must be included in their correlation. They therefore used the bubble radius and bubble growth rate; given by Eqs. III-3 and III-4, as the characteristic length and velocity in Nusselt and Reynolds numbers which they defined for the boiling liquid. The final form of their semi-empirical correlation is:

$$q_b = \frac{0.0015 k_l \left[\frac{c_{pl}}{\mu_l} \left(\frac{(T_w - T_s) c_{pl} \rho_l (\pi a_l)^{3/2}}{\lambda \rho_v} \right)^2 \right]^{0.62} \left[\frac{c_{pl} \mu_l}{k_l} \right]^{0.33}}{\frac{c_{pl} \rho_l (\pi a_l)^{3/2}}{\lambda \rho_v} \left[\frac{2G}{\Delta P} \right]^{1/2} \left[\frac{c_{pl}}{\Delta P g_c} \right]^{1/2}} \quad \text{(III-5)}$$

Eq. III-5 has been found to predict nucleate boiling heat transfer data for water, benzene, ethanol, methanol, mercury, and sodium with an accuracy that is generally within ± 25 to $\pm 50\%$ (F2, C1, W2).

Chang and Snyder (C2), Forster and Greif (F3), and McNelly (M6) employed dimensional analysis to obtain correlating equations for nucleate boiling heat transfer. Hughmark (H4) analyzed the experimental data of several investigators with a statistical technique to obtain a correlating equation for nucleate boiling. Levy (L3) employed a simplified model of nucleate boiling heat transfer to obtain his correlating equation. The semi-empirical correlating equations of the above investigators are as follows:

(a) Chang and Snyder:

$$q_b = \frac{4 \times 10^{-4} k_l \Delta P^{1.4}}{(\lambda \rho_v)^{0.8} \sigma} [c_{pl} T_s (\rho_l - \rho_v)]^{0.4} \Delta T ; \quad (\text{III-6})$$

(b) Forster and Greif:

$$q_b = 4.3 \times 10^{-5} \frac{a_l c_{pl} \rho_l T_s}{\sigma^{1/2} (\lambda \rho_v)^{3/2}} (c_{pl} T_s a_l^{1/2})^{1/4} \left[\frac{\rho_l}{\mu_l} \right]^{5/8} \left[\frac{\mu_l c_{pl}}{k_l} \right]^{1/3} (\Delta P)^2 ; \quad (\text{III-7})$$

(c) McNelly:

$$q_b = \left[0.225 k_l \left[\frac{c_{pl}}{k_l \lambda} \right]^{0.69} \left[\frac{P}{\sigma} \right]^{0.31} \left[\frac{\rho_l}{\rho_v} - 1 \right]^{0.33} \Delta T \right]^{3.23} ; \quad (\text{III-8})$$

(d) Hughmark:

$$q_b = \frac{2.67 \times 10^{-9} \Delta P^{1.867} (\rho_l - \rho_v)_w^{2.27} (c_{pl})_w^{0.945} T_w^{1.618}}{(\rho_v)_w^{1.385} \lambda_w^{1.15} (\mu_l)_w^{1.630} (P_r)^{0.202}} ; \quad (\text{III-9})$$

(e) Levy:

$$q_b = \frac{k_l c_{pl} \rho_l^2}{\sigma T_s (\rho_l - \rho_v) B} (\Delta T)^3 , \quad (\text{III-10})$$

where B is a function of $\lambda \rho_v$. Equations III-6 through III-10 have generally been found to predict nucleate boiling heat transfer data within a factor of 2 (C3, H4).

The Rohsenow correlation (R3) is the most widely used and tested method of correlating nucleate boiling heat transfer data. In his analysis of the nucleate boiling process, Rohsenow assumed that the movement of bubbles as they are breaking away from the heating surface is of

prime importance. By obtaining analytical expressions for the mass velocity and diameter of bubbles as they break away from the surface, and by defining Reynolds and Nusselt numbers in terms of these expressions, Rohsenow derived the following correlating equation:

$$\frac{c_{pl}\Delta T}{\lambda} = c_{sf} \left[\frac{q_b}{\mu_l \lambda} \sqrt{\frac{g_c \sigma'}{g(\rho_l - \rho_v)}} \right]^u \left[\frac{c_{pl} \mu_l}{k_l} \right]^s, \quad (\text{III-11})$$

where the constants c_{sf} , u , and s must be determined empirically for each fluid-surface combination. Rohsenow employed experimental heat transfer data for several fluid-surface combinations to obtain the constants c_{sf} , u , and s . For the systems that Rohsenow considered, u generally had a value close to 0.33. For clean surfaces, Rohsenow obtained a value of 1.7 for s , whereas s varied between 0.8 and 2.0 for contaminated surfaces. Nucleate boiling heat transfer data for a particular fluid-surface combination are generally correlated within $\pm 20\%$ by Eq. III-11.

B. Condensation

1. Nusselt Equation

Film condensation occurs when the liquid condensate wets the "cooling" surface, forming a continuous film that covers the surface while draining off. For the case of laminar film condensation on a horizontal cylinder, an equation derived by Nusselt (24) can be used to predict the heat transfer rate:

$$q_{Nu} = 0.725 \left[\frac{g k_l^3 \rho_l^2 \lambda}{D \mu_l} \right]^{1/4} (T_v - T_o)^{3/4}, \quad (\text{III-12})$$

where D is the diameter of the cylinder and q_{Nu} is the condensing heat velocity predicted by the Nusselt equation. In developing his theory of film condensation, Nusselt made several simplifying assumptions:

- (1) the gravitational force acting on each particle of the liquid film is in balance with the viscous drag force;
- (2) acceleration of the film is negligible;
- (3) vapor drag at the condensate surface is negligible;
- (4) the temperature profile across the film is linear at all locations;
- (5) only the latent heat of condensation is conducted through the film;
- (6) convection within the film is negligible;
- (7) the total thermal resistance is in the liquid film.

Most of the simplifications made by Nusselt have been checked analytically and found to have insignificant effect on the heat transfer data predicted by the Nusselt equation. Rohsenow (R4) considered the effect of a nonlinear temperature profile due to condensate flow on the heat transfer. Koh (K4) and Chen (C4) generalized the condensation problem to include the effects of convection and acceleration in the liquid film and vapor drag at the condensate surface. For condensing fluids with moderately high Prandtl numbers and low Reynolds numbers (which is the case in the condensation of selenium), the results of the modified-Nusselt analyses were found to differ insubstantially from the results predicted by the Nusselt equation.

To account for the subcooling of the condensate film, the Nusselt equation can be rewritten:

$$q_{Nu} = 0.725 \left[\frac{g k_l^3 \rho_l^2 \lambda'}{D_l \mu_l} \right]^{1/4} (T_v - T_o)^{3/4}, \quad (\text{III-13})$$

where

$$\lambda' = \lambda + c_{pl}(T_v - T_{mix}). \quad (\text{III-14})$$

Colburn (C9) has shown that the mixing cup temperature of the condensate leaving the condensing surface, T_{mix} , should be computed by:

$$T_{mix} = T_v - \frac{3}{8}(T_v - T_o). \quad (\text{III-15})$$

λ should be evaluated at the vapor temperature T_v , while the other physical properties in Eq. III-13 should be evaluated at an average film temperature T_f (C8):

$$T_f = T_o + \frac{1}{4}(T_v - T_o). \quad (\text{III-16})$$

Assumption (7) implies that the thermal resistance at the liquid-vapor interface is negligible. In the film condensation of water and organic liquids, the thermal resistance at the liquid-vapor interface is negligible, and Nusselt heat transfer predictions generally agree with experimental results within 10 to 15%. However, in the condensation of liquid metals, which have very high thermal conductivities, the thermal resistance of the condensate film is very small; the resistance at the liquid-vapor interface therefore becomes very significant and the Nusselt-predicted heat transfer coefficients generally exceed the experimental heat transfer coefficients by factors of 10 or more.

2. Molecular Condensation Coefficient

The molecular condensation coefficient (σ_c) for a vapor in contact

with a liquid is generally defined as "the fraction of molecules striking the condensate surface that actually condenses". To determine the condensation coefficient for a condensing vapor from heat transfer data, a procedure developed by Schrage (S2) can be employed.

According to the kinetic theory of gases, the flux, G , of molecules in any one direction in an ideal gas at equilibrium is given by the equation:

$$G = P \sqrt{\frac{M}{2\pi RT}} \quad (\text{III-17})$$

By applying the kinetic theory of gases to the streams of molecules striking and leaving the condensate surface, Schrage (S2) derived the following equation:

$$\frac{W}{A_c} = \left[\frac{2\sigma_c}{2 - \sigma_c} \right] \left(\frac{M}{2\pi R} \right)^{1/2} \left[\frac{P_v}{T_v^{1/2}} - \frac{P_{sl}}{T_{sl}^{1/2}} \right], \quad (\text{III-18})$$

where W is the mass rate of condensation, T_{sl} is the temperature at the surface of the condensate film, and P_{sl} is the vapor pressure corresponding to temperature T_{sl} . In most condensation experiments the vapor temperature T_v , the wall temperature T_o , and the condensing heat velocity q_c are determined experimentally. W/A_c can then be calculated:

$$\frac{W}{A_c} = \frac{q_c}{\lambda_f} \quad (\text{III-19})$$

By applying the Nusselt equation to the condensate film, the value of T_{sl} can be calculated:

$$T_{sl} = \left(\frac{q_c}{0.725 \left[\frac{g_k^3 \rho_l^2 \lambda^2}{Du_l} \right]^{1/4}} \right)^{4/3} + T_o \quad (\text{III-20})$$

Equation III-18 can then be solved for σ_c .

The kinetic theory of condensation has been employed to correlate heat transfer data in the condensation of liquid metal vapors. Larrain (12) has correlated most of the available heat transfer data in the condensation of liquid metal vapors to obtain a compilation of condensation coefficients for liquid metals. In general, the condensation coefficients for the liquid metals are found to correlate with P_{sl} . At pressures below approximately 10 Torr, the values of σ_c are generally close to 1. At higher pressures, σ_c is found to decrease with increasing pressure, most of the data indicating a linear relationship between $\log \sigma_c$ and $\log P_{sl}$. Barry and Sartor (B6) investigated the condensation of sodium and considered the data of several other investigators. They concluded that the kinetic theory of condensation can be used to successfully correlate heat transfer data in the condensation of liquid metal vapors.

IV. CORROSIVENESS OF SELENIUM

Several investigators of selenium's physical and chemical properties have reported that selenium is generally quite corrosive. An investigation of the corrosiveness of selenium towards stainless steels 304, 316, 410, 430, and 446, and Hastelloys B, C, and F has been made by Cooper (C10). He concluded that for temperatures between 400 and 650°C, stainless steel 316 was the most reliable containing material for selenium of these materials.

A. Corrosion Tests

To determine a suitable material of construction for the apparatus, a series of corrosion tests was carried out at the beginning of this research program.

The first group of samples tested included pure nickel, low carbon nickel, Haynes-25, and the stainless steels 304 and 316. The samples and 100 g of selenium were placed in a mullite tube under argon as cover gas, maintained in an oven at 620°C for six weeks, then removed from the mullite tube and separated from the solid selenium. Most of the selenium was readily chipped from the samples. Any remaining selenium was removed by placing the samples in CS_2 and in a weak alkaline cleaning solution. The pure nickel, low carbon nickel, and Haynes-25 samples had become very brittle.

The stainless steel samples were in apparently good condition. They were analyzed by weighing them before and after the corrosion test to determine their weight changes, and by examining their surfaces under a microscope. The corrosion rates were found to be:

<u>Sample</u>	<u>Corrosion Rate</u>	
	<u>(g/sq in.-month)</u>	<u>(mils/month)</u>
stainless steel 304	1.20	9.1
stainless steel 316	0.82	6.2

Microscopic examination of a polished section of each stainless steel sample revealed a smooth, regular interface between a very thin film (obviously a selenide) and the metal sample. In no case did corrosion penetrate deeply into a sample.

The second group of samples tested included the stainless steels 316, 410, and 446, Hastelloy C, tantalum, tungsten, molybdenum, and graphite, and copper wires electroplated with thick deposits of chromium. The samples and 120 g of selenium were heated under argon as before, except in a fused quartz tube. After one month at 600°C, the samples were removed from the quartz tube and separated from the selenium. The tantalum, tungsten, molybdenum, and chromium-plated copper wire samples had disappeared completely during the corrosion test, showing they were evidently unsuitable. The graphite sample significantly increased in weight during the corrosion test, most likely due to absorption of selenium into the pores of the graphite.

The corrosion rates of the remaining samples were as follows:

<u>Sample</u>	<u>Corrosion Rate</u>	
	<u>(g/sq in.-month)</u>	<u>(mils/month)</u>
Hastelloy C	2.00	13.7
stainless steel 410	2.70	21.3
stainless steel 446	1.41	11.5
stainless steel 316	1.28	9.7

The principal conclusions derived from these corrosion tests were:

- (1) Of the metals tested, stainless steel 316 had the best corrosion resistance towards selenium at temperatures around 600°C.
- (2) Stainless steel 316 had sufficient corrosion resistance towards selenium to be used as the material of construction for the apparatus employed in obtaining the heat transfer data.
- (3) However, no metal tested so far can resist selenium sufficiently at these temperatures to make use of selenium practical in commercial applications or probably even short space power missions.

V. APPARATUS

Because of the superior corrosion resistance of stainless steel 316 towards selenium at the temperatures of interest, the apparatus was designed and constructed in such a way that only stainless steel 316 came into contact with selenium. The boiling section of the apparatus contained a horizontal stainless steel 316 boiling plate, and the test condenser consisted of a horizontal stainless steel 316 tube.

A. Boiling Section

The boiling section of the apparatus that was used to obtain the pool boiling heat transfer data, as well as produce vapor for the condensing section, contained a solid copper cylinder with its bottom $2/3$ heated electrically, the upper part of the cylinder serving to diffuse the heat to a stainless steel 316 boiling plate brazed to the top of the copper cylinder, as shown in Fig. V-1. In designing the heater dimensions to supply a relatively uniform heat velocity to the boiling liquid, an electrical analogue study of the heat conduction was made, as described in Appendix D. It was shown that, except for possible nonuniformities due to conduction of heat along the wall of the test chamber, the boiling heat velocity on top of the stainless steel plate would nowhere deviate from its average value by more than 1% over the range of boiling conditions.

The boiling section was constructed by brazing a $3/4$ in. thick stainless steel 316 disc to the top of the copper cylinder. The brazing was carried out in a dry hydrogen furnace at Temperature Processing Co., North Arlington, N.J. Engaloy 255 brazing alloy, with a composition of

Fig. V-1. Boiling Section

82% Au and 18% Ni and a melting point of 950°C , was used. Ultrasonic tests of the brazed joint were performed at Sperry Products, Danbury, Conn. by immersing the test specimen in water and looking for reflections of the high frequency ultrasonic waves from any unbonded areas. The tests showed that the copper and stainless steel surfaces were perfectly bonded.

The disc was then turned into the shape of a cup, with the dimensions shown in Fig. V-1. From the original thickness of the stainless steel disc and the depth of the machined "cup" below its rim, the thickness of the boiling plate was found to be 0.103 in. The rms roughness of the boiling plate was estimated to be 60 microinches by comparing the surface of the boiling plate to several standard surfaces, both visually (under a microscope) and by feel. It should be noted that, due to the continual slow corrosion of the boiling plate by the liquid selenium, the actual surface conditions of the boiling plate must have varied during the course of the experiments with selenium. However, tests for reproducibility of results showed little apparent change. To reduce the conduction of heat upwards along the wall of the test chamber, which would cause an uncertainty in the boiling heat flux calculated from an energy balance around the copper cylinder, and a decrease in the uniformity of the boiling heat velocity and boiling surface temperature, the wall thickness near the boiling plate was reduced to 0.065 in. on a lathe for a length of 0.4 in.

Six thermocouple holes, 0.042 in. in diameter, were drilled radially into the copper cylinder, parallel to the boiling plate. Three holes were drilled 0.75 in. deep and three holes were drilled 1.25 in. deep into the copper, as shown in Fig. V-1. Sheathed, chromel-alumel thermocouples were placed in the thermocouple holes and the observed

temperatures extrapolated to the copper surface, to determine the boiling heat velocity and boiling surface temperature. A fixed thermocouple was located in the vapor space of the test chamber, while two moveable thermocouples, located in the boiling pool, were used to obtain the liquid temperature profile in the boiling pool.

The power supplied to the heater was controlled by a variable transformer and measured with precision electrical meters. An energy balance, made around the copper cylinder, was used to check the boiling heat velocity that was calculated from the temperature measurements in the copper cylinder.

The heater used Kanthal A-1 resistance wire wound around an alundum core that had been placed around the bottom 4.4 in. of the copper cylinder. The alundum core, which electrically insulated the Kanthal A-1 wire from the copper cylinder, had an ID of 3.0 in., a wall thickness of 0.25 in., 6 turns per inch, and a groove diameter of 1/16 in. The Kanthal A-1 wire had a diameter of 0.051 in., a resistance of 0.338 ohm/ft, and a maximum temperature rating of 1375°C.

The core was cut by two axial planes into four equal sections held tightly against the copper cylinder by two insulated steel straps, which also held the ends of the Kanthal wire in the grooves. The power leads were also A-1 wire, but made of four wires in parallel. The resistance of the heater was measured at room temperature with a Wheatstone Bridge to be 7.67 ohms, while the resistance of the Kanthal and copper lead wires was 0.13 ohms. By utilizing these measured resistances, the measured power was corrected for power dissipation in the lead wires. Due to the small magnitude of the correction and to the near independence of

resistivity of Kanthal wire with temperature (less than a 4% variation from 25 to 1300°C), temperature had a negligible effect on the corrected power.

B. Condensing Section

The test condenser consisted of a horizontal, thin-walled, stainless steel 316 tube, brazed to an inner concentric silver tube. By placing the thermocouples in the silver tube, which has a high thermal conductivity, the error that would be present in the condensing temperature difference due to uncertainty in the effective locations of the thermocouple junctions was minimized. In addition, the temperatures in the condensing wall would become circumferentially very uniform, and thus close to the boundary conditions of the Nusselt equation. The test condenser is shown in Fig. V-2.

The silver was machined from a solid silver rod into a tube of the dimensions shown in Fig. V-2. The thermocouple holes were drilled 0.042 in. in diameter, 90° apart, and 1 in. deep from each end of the silver tube for the placement of chromel-alumel thermocouples. By measuring the temperatures at these eight positions, a good overall average was obtained.

Grooves were machined, both inwardly and outwardly, on both ends of the silver tube. The grooves on the outside surface of the tube were used to facilitate the brazing of the silver tube inside the stainless steel condensing tube. The grooves on the inside surface were utilized for shrink-fitting extension tubes, used to pass coolant along the axis of the test condenser.

Fig. V-2. Condensing Section

The brazing of the silver tube into the stainless steel tube was carried out in a very dry hydrogen furnace at Kern Mfg. Co., Long Island City, N.Y. The brazing alloy, Lithobraz BT, had a composition of 71.8% Ag, 28% Cu, and 0.2% Li and a melting point of 760°C ; this brazing alloy was recommended because of its ability to wet stainless steels. Strips of the brazing alloy were placed in the top groove between the silver and stainless steel 316 tubes, which were held vertically during the brazing process. The radial clearance between the silver and the outer stainless steel tubes was 0.00075 in. at the brazing temperature (approximately 1500°F). The thermocouple holes were painted with titanium oxide to prevent their filling with brazing alloy. Ultrasonic tests of the brazed joint were performed at Sperry Products and showed that the bonding between the silver and stainless steel surfaces was excellent over the entire length of the brazed joint.

The test condenser was welded to the wall of the test chamber, which consisted of a stainless steel 316 tube with an ID of 2.75 in., an OD of 3.00 in., and a length of 9 in. To eliminate any possibility of damaging the brazed joint during the welding, auxiliary pieces of tubing were first heliarc welded to the test chamber. These pieces of tubing were machined to give a snug fit with the test condenser. The test condenser was then lined up with and fusion welded to the auxiliary pieces of tubing at both ends of the test condenser (see Fig. V-2).

A stainless steel 316 disc, $1/8$ in. thick, was heliarc welded to the top of the test chamber. A stainless steel 316, 1 in. ID, Swagelok tube connector and a stainless steel 316 bushing had been welded to the top of the disc. The Swagelok tube connector and the bushing made tight

seals with an auxiliary condenser and with the sheath of the fixed thermocouple in the vapor space of the test chamber.

To pass coolant through the test condenser, extension tubes were shrunk-fit into the grooves on the inside surface of the test condenser. Each extension tube was constructed by brazing a 1.5 in. long silver tube to a 2 in. long stainless steel 316 tube. The ends of the silver tubes were machined to give a slight interference fit with the grooves. The composite tubes were then lined up with the test condenser, put in liquid nitrogen, and mechanically rammed into place. Since silver was shrunk-fit into silver, loosening of the extension tubes by thermal expansion during the heat transfer experiments was prevented. The use of the stainless steel sections prevented the ends of the extension tubes from reaching 500°F. This was important because the extension tubes were connected to silicone rubber tubing, whose upper temperature limit was 500°F.

Compressed air was used as the coolant for the test condenser. From the flow rate and temperature rise of the air, the condensing heat velocity was determined. The eight thermocouples in the test condenser wall and the fixed thermocouple in the vapor space of the test chamber were used to determine the average condensing temperature difference.

C. Assembled Apparatus

The boiling and condensing sections were incorporated into a single apparatus for obtaining boiling and condensing heat transfer data simultaneously. The bottom of the test chamber was heliarc welded to the rim of the stainless steel 316 "cup", approximately 3/4 in. above the boiling

plate (see Fig. V-1). The center line of the horizontal condensing tube was located 7 in. above the boiling plate and 3 in. below the top of the test chamber.

A schematic diagram of the assembled apparatus is shown in Fig. V-3. The apparatus is also shown in Figs. V-5 and V-6.

1. Air Coolant Line

The compressed air that was used as the coolant for the test condenser was first passed through a Cuno filter. A Moore Nullamatic pressure regulator was used to control the air flow rate. By setting the pressure at the downstream side of the regulator, the desired air flow rate could be obtained. The pressure regulator was found to be very effective in eliminating pressure fluctuations downstream from the regulator, although the pressure upstream from the regulator varied by several psi. A steady air flow rate was therefore obtained. The downstream side of the pressure regulator consisted of 1.5 in. ID pipe, in which the air temperature was measured with a chromel-alumel thermocouple. The pipe was connected to 3/8 in. OD tubing that was inserted into the closed chamber through a Conax gland.

To allow for thermal expansion of the apparatus during heat transfer experiments, flexible silicone rubber tubing, with an ID of 3/8 in. and an upper temperature limit of 500°F, was used to connect the tubing of the air coolant line with the extension tubes that had been shrunk-fit into the test condenser.

The temperature of the air coolant at the outlet of the test condenser was measured in an expansion chamber. The air velocities in the

Fig. V-3. Schematic Diagram of Assembled Apparatus

Fig. V-4. Inside of Closed Chamber

O R N L PHOTOGRAPH NO. 3012-72

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COLOR ☐

Fig. V-5. Apparatus

Fig. V-6. Apparatus

O R N L PHOTOGRAPH NO. 3013-72

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tubing of the air coolant line reached values of 500 fps and higher during the heat transfer experiments. At these high velocities, accurate measurement of the air temperature would be very difficult because of the large differences between the stagnation and static temperatures of the air stream. By reducing the velocity of the air to 40 fps or less in the expansion chamber, the correction in the temperature measurement due to the kinetic energy of the air stream was made negligible. The expansion chamber, which was made of brass, is shown in Fig. V-7. Rods and screens were placed near the inlet of the expansion chamber to increase mixing and spreading of the air throughout the chamber. Conical sections at the inlet and outlet of the chamber helped achieve smooth slowing down of the air stream in the chamber. Three chromel-alumel thermocouples were located at different radial positions within the chamber to check the uniformity of the air temperature. The thermocouple wires were inserted through a Conax gland with teflon sealant.

The hot air coming out of the expansion chamber was cooled in a Thrush shell and tube heat exchanger to approximately room temperature. The air was then passed through a wet test meter, which measured its volumetric flow rate.

2. Inert Gas Line

Argon cover gas of 99.995% purity was used to control the pressure in the test chamber. To keep the argon out of the test chamber, an auxiliary reflux condenser was connected to the top of the test chamber, and the boiler was always set to produce a slight excess of selenium vapor that was not condensed by the test condenser. This excess vapor

Fig. V-7. Expansion Chamber

stream acted like a diffusion pump, carrying the argon in the vapor space out into the auxiliary condenser.

The auxiliary condenser consisted of a 4 ft. long, 1 in. OD, stainless steel 316 tube that was cooled by the atmosphere in the room. To prevent condensate in the auxiliary condenser from dripping onto the test condenser, the auxiliary condenser was located near the edge of the disc at the top of the test chamber (see Fig. V-8). A short piece of stainless steel 316 tubing, whose bottom was cut at an angle, was welded to the bottom of the disc directly below the auxiliary condenser to induce dripping to occur farther away from the test condenser.

The feed tube, welded near the top of the auxiliary condenser and sealed with a Swagelok cap, was used to charge the test chamber. A bellows assembly was welded to the top of the auxiliary condenser. The bellows was used to raise and lower two long thermocouples that passed through the bellows and auxiliary condenser into the boiling pool, thereby providing the temperature profile in the boiling liquid pool. A schematic diagram of the bellows assembly is shown in Fig. V-9. The bellows, made from stainless steel 321, had a total travel of 0.5 in. Since the hot junctions of the two long thermocouples were located 0.7 in. apart, the total distance that the two thermocouple junctions could cover was 1.2 in. A screw mechanism was used to compress and stretch the bellows, to vary the positions of the hot junctions. The thermocouple sheaths passed through Conax glands welded to a disc at the top of the bellows. The fixed plate was held stationary by attaching it to the auxiliary condenser. The distances of the thermocouple junctions from the boiling plate were determined by measuring the distance between

Fig. V-8. Top of Test Chamber

Fig. V-9. Bellows Assembly

the top of the screw and the top of the nut. A reference point had been determined by noting the position of the screw at which electrical contact was made between the lower thermocouple sheath and the boiling plate with the test chamber empty.

For argon pressures above 103 Torr, the pressure was measured with an open-end mercury manometer. For pressures of 103 Torr or less, a closed-end mercury manometer was used. When the open-end manometer was employed, the barometric pressure was also measured. A reservoir with a volume ten times that of the test chamber was inserted into the inert gas line to facilitate pressure control. A Genco Hypervac 23 vacuum pump was employed for evacuating the test chamber. Hoke packless-diaphragm valves, types 411 and 413, were used in the inert gas line. The tubing in the inert gas line was stainless steel 316 with 3/8 in. OD. All connections were either soldered or made with Swagelok tube fittings.

3. Closed Chamber

Since selenium is very toxic, the test chamber was placed inside an outer closed chamber, minimizing the danger of selenium vapor leakage into the atmosphere. To prevent oxidation of the test apparatus, especially the copper cylinder, the outer chamber was filled with argon gas at a slight positive pressure.

The outer chamber consisted of a welded steel box with sides ribbed to stand vacuum and a removeable front cover sealed with a rubber gasket. The thermocouple wires and the copper power wires were inserted into the closed chamber through Conax glands welded to the top of the chamber.

The 3/8 in. OD tubing in the air coolant line passed through the side walls of the closed chamber, also through Conax glands. The auxiliary condenser was inserted through a Swagelok tube fitting welded to the top of the closed chamber. The closed chamber was grounded as a safety precaution in case of a power circuit fault. The closed chamber, with the front cover removed, is shown in Fig. V-4.

To reduce heat losses, the copper cylinder and the test chamber were thermally insulated with a 1.5 in. thick layer of Fiberfrax (bulk fiber and Lo-Con blanket insulation). Fiberfrax consists of alumina-silica fibers stable to 2300°F; the blanket form, 3/8 in. thick, was found very convenient for covering the cylindrical surfaces. The air coolant line and expansion chamber were also thoroughly insulated to minimize heat losses.

4. Instrumentation

The six thermocouples in the copper cylinder and the eight thermocouples in the test condenser were chromel-alumel, grounded-hot-junction, special accuracy thermocouples with stainless steel 304 sheaths of 0.040 in. OD. The fixed thermocouple in the vapor space of the test chamber and the two moveable thermocouples in the boiling pool were chromel-alumel, insulated-hot-junction, special accuracy thermocouples with stainless steel 316 sheaths of 0.062 in. OD. The thermocouples used to measure the air coolant temperatures at the inlet and outlet of the test condenser were made from 20 gage chromel-alumel thermocouple wire that had been calibrated; the beaded junctions were made by gas welding. All sheathed thermocouples and thermocouple wire were obtained from Thermo-Electric Company, Saddle Brook, N.J.

Each thermocouple was connected through multipoint two-pole switches to a Leeds and Northrup portable precision potentiometer. By adjusting the switches, the desired single thermocouple and differential thermocouple emfs could be read.

A Pt/Pt-10Rh thermocouple whose junction was located at the bottom (the hottest level) of the copper cylinder was connected to a temperature controller that would open a relay and cut off the power to the heater in case the copper cylinder ever reached 1050°C (which never happened).

A Sargent recorder, with variable millivolt range and chart speed, was used to record temperature fluctuations within the copper cylinder and the boiling liquid. Other instruments included:

American Wet Test Meter, Model A1 23, 20 scfm range;

Westinghouse A.C. Wattmeter, Type PY-5, 0 to 3000 watt range;

Weston A.C. Ammeters, Model 433, 0.4 to 15 and 0.8 to 30 ampere ranges;

Weston A.C. Voltmeters, Model 155, 30 to 125 and 80 to 300 volt ranges.

The calibrations of the instruments and thermocouples are discussed in Appendix B.

VI. EXPERIMENTAL PROCEDURE

The apparatus was assembled and the test chamber was cleaned by soaking with trichloroethylene, then draining and flushing with distilled water. The Swagelok tube fitting at the top of the test chamber was then connected to the auxiliary condenser and the apparatus was leak-tested with a soap solution and a halogen leak detector under positive pressure, and a helium leak detector under vacuum. After repairing all leaks, the boiling plate was heated to 500°C and the apparatus outgassed to vacuum for several hours. The apparatus was not considered to be sufficiently tight until it could hold a vacuum for 18 hours with less than 0.2 Torr pressure change. Whenever the apparatus was taken apart and reassembled, the leak-testing procedure was repeated.

Heat loss calibrations (Appendix B), heat transfer experiments with water and Dowtherm A (Sect. VII-A), and relative in-place calibrations of thermocouples (Appendix B) were then carried out.

Selenium of 99.999+% purity, from the American Smelting and Refining Company, was used. Four-hundred-fifty-two grams of selenium were charged into the test chamber. The outer chamber was outgassed for several hours and filled with argon to a slight positive pressure. The test chamber was evacuated and flushed three times with argon, after which it was brought to the desired argon pressure.

During each run, the power and air flow rate were adjusted until excess vapor condensed in the auxiliary condenser, as shown by the height of the hot (condensing) wall. Temperatures in the copper cylinder and the test condenser were monitored until they remained constant for several minutes, indicating that steady-state conditions had been reached.

Then the following data were recorded: the readings of the manometers, the wet test meter, and the electrical meters, the height of the condensing ring in the auxiliary condenser, and the thermocouple emfs. Temperatures in the vapor and in the boiling liquid, temperature differences between the vapor and the thermocouples in the test condenser and the copper cylinder, and temperature differences between the expansion chamber and the inlet of the test condenser were measured. During several runs, the output poles of the switch box were connected to the Sargent recorder and temperature fluctuations in the copper cylinder and the boiling liquid were recorded.

At the conclusion of a series of runs, the power was turned off. Before the next series was started, the apparatus was outgassed, checked for leaks, and flushed with argon. The power was then turned on and the above procedure repeated. Except for the first of a series of runs, which required two to three hours, each run required thirty to ninety minutes to reach a steady state.

Steady-state conditions in the boiling section could not be reached for runs S29 through S33. During these runs the copper cylinder became steadily hotter until the power had to be reduced to avoid damaging the boiling section. The test chamber and the auxiliary condenser were disconnected after run S33. Most of the selenium had frozen in the auxiliary condenser, while only a 1/8 to 1/4 in. thick layer remained on the boiling plate. The selenium (and selenides) in the test chamber and auxiliary condenser was dissolved in nitric acid. After removing all traces of selenium, the test chamber and the auxiliary condenser were flushed with a steady stream of tap water, followed by thorough flushing with distilled water.

The test chamber was then charged with 444 g of selenium and runs S34 to S49 were carried out. Several of the earlier runs were reproduced to check for any possible effect of corrosion on the heat transfer data.

Inerts were effectively eliminated from the test chamber by welding most of the joints in the apparatus, by thorough leak-testing of the apparatus, and by utilizing the auxiliary condenser to flush the argon cover gas and retain it out of the test chamber. To test for any possible effect of inerts on the heat transfer, the height of the condensing ring in the auxiliary condenser was varied by varying the power while the pressure and the air flow rate were held constant. Alternatively, in other tests the pressure, power, and air flow rate were all held constant and the condensing ring was raised by thermally insulating the bottom portion of the auxiliary condenser from the atmosphere. Since the height of the condensing ring was the length of the path for the back-diffusion of argon into the test chamber, the effect of varying the height of the condensing ring on the heat transfer data (especially the condensation data) could be used to determine the effect of inerts on the heat transfer data.

VII. RESULTS

A. Preliminary Experiments

To confirm the reliability of heat transfer data obtained with the assembled apparatus and to develop an efficient operating procedure with selenium, boiling and condensing experiments were first carried out with distilled water and with Dowtherm A.

1. Water

The results of the water boiling experiments are presented in Table VII-1. The water boiling data that were obtained at a pressure of approximately 1 atm are shown in Fig. VII-1, where the boiling heat velocity is plotted against the boiling temperature difference. The solid lines represent the data of Lyon (L6) and Vachon (V1). The data of Lyon were obtained by boiling water on a heated, horizontal, stainless steel 316 tube at a pressure of 1 atm; the surface conditions of the stainless steel 316 tube were not specified. The data of Vachon were obtained by boiling water on a heated, horizontal, stainless steel 304 plate at a pressure of 1 atm; the surface of the boiling plate, which had been mechanically polished, was determined by means of a profilometer to have an rms roughness of 61 microinches. It can be seen from Fig. VII-1 that these data generally agree with the data of both Lyon and Vachon within 2 to 3°F. The small differences between the three sets of data could easily be due to differences in surface conditions.

2. Dowtherm A

Experiments with Dowtherm A were carried out between 105 and 954 Torr, corresponding to a range of vapor temperatures from 362 to 519°F.

Table VII-1. Boiling Heat Transfer Data for Water

Run	Power (corrected) (watts)	T_v (°F)	P (Torr)	q_b (Btu/ft ² -hr)	$(T_w - T_v)$ (°F)	h_b (Btu/ft ² - hr-°F)
W1	188.5	217.40	848.7	13,968	14.44	967.6
W2	251.0	218.15	853.8	18,053	15.43	1170.2
W3	300.0	218.63	859.9	21,195	15.78	1342.9
W4	158.5	211.80	757.9	11,129	14.58	763.3
W5	179.0	212.00	760.4	12,023	15.38	782.0
W6	205.0	211.73	757.9	13,267	15.86	836.5
W7	222.5	211.76	759.6	14,821	15.98	927.3
W8	255.0	212.07	761.5	18,565	15.20	1221.6
W9	283.0	211.93	761.7	18,862	16.99	1109.9
W10	300.0	211.90	761.1	18,548	17.09	1037.1
W11	199.0	250.25	1524.7	14,741	12.60	1170.1
W12	252.0	250.23	1526.0	17,184	14.44	1190.4
W13	305.0	250.20	1520.0	22,141	13.94	1588.5
W14	360.0	250.30	1517.5	24,448	15.85	1542.7
W15	255.0	211.80	759.9	18,391	15.83	1162.0
W16	255.0	249.87	1513.4	17,183	14.48	1186.3
W17	256.0	211.60	757.1	17,142	17.60	974.0

Fig. VII-1. Heat Velocity vs. Temperature Difference for the Boiling of H_2O , Compared with Published Results at Atmospheric Pressure

The boiling data are presented in Table VII-2. It should be noted that the pressure was not measured during runs D14 through D21. Thus, the values of P reported in Table VII-2 for runs D14 through D21 are not measured values, but are the saturation pressures corresponding to the measured vapor temperatures. In all other cases, the tabulated values of P were experimentally determined.

The boiling data that were obtained at pressures of approximately 1 and 1/2 atm are shown in Fig. VII-2, where the boiling heat velocity is plotted against the boiling temperature difference. The solid lines represent the heat transfer data predicted by the Forster-Zuber equation (Eq. III-5) at saturation pressures of 1 and 1/2 atm. It can be seen from Fig. VII-2 that the agreement between the experimental and predicted heat transfer data is generally within 25 to 50%.

The condensation data are presented in Table VII-3, where H represents the height of the condensing ring in the auxiliary condenser and h_{Nu} represents the condensation heat transfer coefficient predicted by the Nusselt equation (Eq. III-13). The last column in Table VII-3 shows that the experimental heat transfer coefficients (h_c) generally agree with the Nusselt-predicted heat transfer coefficients (h_{Nu}) within 10 to 15%, which is within the usual accuracy of the Nusselt equation. McAdams compiled condensation heat transfer data for several organic compounds (M5). He found that the average value of h_c/h_{Nu} for eighteen organic compounds undergoing filmwise condensation on horizontal tubes was 0.94, compared to the average value of 0.91 for the Dowtherm A data in Table VII-3.

In testing for the possible effect of noncondensable gas on the heat transfer data, the height of the condensing ring was varied from the

Table VII-2. Boiling Heat Transfer Data for Dowtherm A

Run	Power (corrected) (watts)	T_v (°F)	P (Torr)	q_b (Btu/ft ² -hr)	$(T_w - T_v)$ (°F)	h_b (Btu/ft ² - hr-°F)
D2	450.0	499.50	754.6	29,876	28.44	1050.4
D3	495.0	498.40	744.1	35,808	26.94	1329.4
D4	535.0	498.40	745.3	38,385	28.42	1350.6
D5	552.0	499.73	755.3	37,790	30.25	1249.3
D6	514.0	518.10	943.0	32,194	29.27	1099.7
D7	605.0	519.10	954.0	42,904	29.06	1476.4
D8	324.0	518.70	949.3	23,518	24.08	976.7
D10	395.0	445.30	370.4	26,169	37.26	702.4
D11	460.0	446.40	374.9	31,419	37.21	844.5
D12	510.0	447.23	378.4	33,121	40.77	812.4
D13	340.0	445.25	371.9	20,567	38.68	531.7
D14	500.0	362.67	105.0	28,053	61.42	456.7
D15	600.0	405.60	209.0	34,634	54.45	636.1
D16	692.0	447.30	380.0	50,279	40.52	1240.9
D17	630.0	427.73	288.0	39,497	50.87	776.4
D18	498.0	362.90	105.0	31,763	58.40	543.9
D19	590.0	398.60	188.0	39,549	52.62	751.6
D20	700.0	435.73	325.0	47,102	46.33	1016.6
D21	635.0	415.65	245.0	43,179	51.67	835.7

Fig. VII-2. Heat Velocity vs. Temperature Difference for the Boiling of Dowtherm A

Table VII-3. Condensation Heat Transfer Data for Dowtherm A

Run	T_v (°F)	P (Torr)	Power (watts)	H (in.)	q_c (Btu/ft ² -hr)	$(T_v - T_o)$ (°F)	h_c (Btu/ft ² -hr-°F)	h_{Nu} (Btu/ft ² -hr-°F)	(h_c/h_{Nu})
D1	500.1	760.9	297	9	20,234	87.2	231.9	254.6	0.91
D2	499.5	754.6	450	18	21,218	88.8	238.8	253.5	0.94
D3	498.4	744.1	495	23	22,225	94.9	234.1	249.4	0.94
D4	498.4	745.3	535	27	23,114	100.8	229.2	246.2	0.93
D5	499.7	755.3	552	38	15,329	58.3	263.0	280.2	0.94
D6	518.1	943.0	514	22	21,567	91.8	235.0	251.5	0.93
D7	519.1	954.0	605	33	21,395	91.5	233.7	251.8	0.93
D8	518.7	949.3	324	9	21,356	91.3	234.0	252.0	0.93
D9	519.0	954.7	256	0	17,011	166.4	102.2	219.0	0.47
D10	445.3	370.4	395	19	18,530	78.6	235.7	260.6	0.90
D11	446.4	374.9	460	29	18,551	77.6	238.9	261.2	0.91
D12	447.2	378.4	510	38	18,437	77.8	237.1	260.7	0.91
D13	445.3	371.9	340	11	18,513	77.1	240.2	261.5	0.92
D14	362.7	105.0	500	50	14,939	61.9	241.4	270.2	0.89
D15	405.6	209.0	600	50	17,347	70.8	245.2	265.2	0.92
D16	447.3	380.0	692	50	18,019	80.8	223.1	258.7	0.86
D17	427.7	288.0	630	50	16,717	71.6	233.4	265.4	0.88
D18	362.9	105.0	498	50	15,570	68.9	225.9	262.4	0.86
D19	398.6	188.0	590	50	17,572	77.3	227.2	258.6	0.88
D20	435.7	325.0	700	50	19,719	87.2	226.0	253.7	0.89
D21	415.6	245.0	635	50	18,483	81.0	228.2	257.0	0.89
D22	499.2	751.9	540	23	24,218	109.5	221.1	241.1	0.92

bottom of the auxiliary condenser to the top of the bellows. During run D9 the bottom of the auxiliary condenser was cold, indicating that inerts must have been present in the top of the test chamber during this run. Runs D14 through D21 were carried out without argon cover gas. A valve was inserted in the line between the auxiliary condenser and the reservoir. By pumping the argon out of the system and closing the valve, it was possible to ensure the absence of noncondensables from the test chamber. Thus, during runs D14 through D21 Dowtherm A condensed throughout the auxiliary condenser and in the bellows.

Close scrutiny of the data in Table VII-3 for runs D6 through D8 and D10 through D13 reveals that wide variations in H had insignificant effect on the condensation heat transfer data. It was therefore concluded that noncondensables were effectively eliminated from the test chamber during the Dowtherm A tests, and consequently had no effect on the condensation process. This conclusion was also confirmed, of course, by the excellent agreement between the experimental and theoretical condensation heat transfer data.

Dowtherm A is actually a eutectic mixture of 26.5% diphenyl and 73.5% diphenyl oxide. Although it is not composed of a single compound, Dowtherm A can be used as a heat transfer standard because it is very nearly a constant boiling mixture. For the range of pressures investigated, the relative volatility ranged from 0.95 to 1.05, there being practically no separation of the components on boiling; the liquid-vapor equilibrium data at atmospheric pressure were found to obey Raoult's law. Dowtherm A can therefore be considered to behave as a pure component with respect to its heat transfer properties (D4). The close agreement between

the experimental and predicted heat transfer data was evidence of the reliability of the apparatus.

The physical property data for Dowtherm A that were employed in Eqs. III-5 and III-13 were obtained from the Dowtherm Handbook (D4). The data for ρ_L , c_{pL} , λ , ρ_V , μ_L , and k_L were considered to be known with good accuracy. Surface tension data were available only up to 140°F. The values of σ employed in Eq. III-5 were estimated to have an uncertainty of $\pm 15\%$.

B. Boiling of Selenium

1. Heat Transfer Data

The power measurements and the pressure and temperature data that were recorded during the boiling experiments are presented and discussed in Appendix A.

The results of the boiling selenium experiments are presented in Table VII-4. Several runs were made at pressures of approximately 10, 30, 100, 300, and 760 Torr. Some data at intermediate pressures were also obtained. Runs CAL1, CAL2, and CAL3 were made during the relative in-place calibrations of the thermocouples in the test condenser and the vapor space (see Appendix B). The pool depth during these runs was approximately 0.5 in. Runs S1 through S49 were carried out with two different charges of selenium, a new charge being added to the test chamber prior to run S34 (see Sects. VI and VII-B-1c). The initial pool depth for both sets of runs was approximately 1.25 in.

The boiling data obtained at pressures of approximately 10, 30, 100, 300, and 760 Torr are shown in Fig. VII-3, where the boiling heat

Table VII-4. Boiling Heat Transfer Data for Selenium

Run	T_v (°F)	P (Torr)	q_b (Btu/ft ² -hr)	$(T_w - T_v)$ (°F)	h_b (Btu/ft ² -hr-°F)
S1	1024.25	101.25	53,171	177.9	298.8
S2	1025.15	101.75	58,715	199.7	294.0
S3	1026.50	101.75	81,407	179.5	453.6
S4	1025.60	100.00	88,844	196.3	452.7
S5	1026.50	100.75	98,688	204.9	481.7
S6	1022.90	100.00	69,514	170.8	406.9
S7	1021.64	98.75	55,760	157.3	354.5
S8	1022.45	98.75	51,489	160.3	321.2
S9	1022.00	98.75	40,103	143.1	280.4
S10	1024.70	99.75	75,870	189.1	401.2
S11	829.85	11.30	42,971	232.9	184.5
S12	851.45	14.80	46,647	245.9	189.7
S13	882.95	21.90	55,935	239.8	233.2
S14	826.25	10.80	34,889	230.6	151.3
S15	825.35	10.60	29,581	232.1	127.0
S16	825.80	10.70	29,148	236.4	123.3
S17	853.25	15.40	47,279	249.5	189.5
S18	832.55	11.70	36,046	246.2	146.4
S20	904.28	29.00	46,730	199.7	233.9
S23	903.92	29.00	38,902	156.3	248.9
S24	904.28	29.00	28,013	135.9	206.1
S25	818.15	28.50	20,300	206.0	98.6
S27	905.90	29.50	38,866	180.1	215.9
S28	905.90	29.50	44,687	184.8	241.8
S34	1024.25	99.50	55,783	176.1	316.8
S35	1024.25	100.00	68,503	170.4	402.1
S36	1023.35	100.00	53,636	147.5	363.7
S37	904.64	29.50	40,982	173.4	236.4
S38	907.25	29.50	58,994	194.0	304.1
S39	822.65	10.00	43,049	228.1	188.8
S40	1146.20	294.50	66,729	109.2	610.9
S41	1146.20	298.40	81,517	121.9	668.6
S42	1146.65	297.40	84,237	130.7	644.5
S43	1147.10	299.80	98,556	135.3	728.5
S44	1146.20	298.80	59,818	111.2	538.0
S45	1022.00	99.50	54,012	150.1	359.8
S46	1272.20	763.00	69,072	82.8	833.8
S47	1272.20	764.50	74,996	83.3	900.4
S48	1272.20	768.50	89,647	79.0	1134.5
S49	1272.20	768.70	97,317	84.1	1157.1
CAL1	852.12	15.25	15,197	154.1	98.7
CAL2	1026.50	102.50	18,410	126.5	145.6
CAL3	1215.95	514.25	26,796	85.5	313.4

Fig. VII-3. Heat Velocity vs. Temperature Difference for Boiling Selenium

velocity (q_b) is plotted against the boiling temperature difference ($T_w - T_v$) on log-log paper. The solid lines were obtained by means of a least squares fit of the heat transfer data with the constraint that all lines were to have the same slope (see Sect. VII-B-2). It can be seen from Fig. VII-3 that the heat transfer coefficients for boiling selenium increased with the pressure, as expected (see Sect. III-A). The relatively high boiling temperature differences for runs S1, S2, and S34 can be explained by the fact that they were the first runs carried out after the apparatus had been charged, indicating that wetting of the boiling plate was evidently incomplete during these runs. The runs obtained at intermediate pressures are not shown in Fig. VII-3.

The values of q_b and ($T_w - T_v$) reported in Table VII-4 and shown in Fig. VII-3 were calculated from the measured temperature profiles in the copper cylinder. An energy balance was made around the copper cylinder to check the boiling heat velocities. The methods of calculating the boiling heat velocities and temperature differences from the experimental data, and the good consistency of the calculated q_b are discussed in Appendix C; sample calculations are also presented. It should be noted that large temperature fluctuations in the copper cylinder during the non-boiling run prevented accurate calculation of q_b by Eq. C-4. The heat velocity reported for run S25 was therefore calculated from the energy balance around the copper cylinder (Eq. C-12).

1a. Temperature Profiles in the Boiling Pool

The temperature profile in the boiling pool was measured during several runs. The data are presented in Table VII-5, where L and $T_\ell - T_v$ represent the distance above the boiling surface and the

Table VII-5. Temperature Profiles in Selenium Boiling Pool

Run	P (Torr)	q_b (Btu/ft ² -hr)	$T_g - T_v$ (°F)								
			L (in.)								
			0.00	0.04	0.20	0.37	0.54	0.74	0.90	1.07	1.24
S1	101.25	53,171	177.9	7.97	2.03	2.03	1.42	2.03	1.85	1.21	1.12
S2	101.75	58,715	199.7	1.71	0.68	0.34	-0.07	0.77	0.77	0.34	-0.07
S3	101.75	81,407	179.5	2.90	0.56	0.34	0.34	0.68	0.34	0.27	0.14
S4	100.00	88,844	196.3	5.45	0.36	0.02	-0.16	-0.50	-0.50	-0.70	-1.01
S5	100.75	98,688	204.9	6.75	0.36	0.36	-0.50	-0.07	-0.50	-0.70	-0.92
S6	100.00	69,514	170.8	7.56	1.21	0.77	0.56	0.36	0.14	0.14	-0.16
S7	98.75	55,760	157.3	6.69	1.21	0.56	-0.07	0.14	0.14	-0.07	-0.41
S8	98.75	51,489	160.3	7.97	1.84	0.77	0.02	0.14	0.14	-0.50	-0.50
S9	98.75	40,103	143.1	9.00	1.42	0.77	0.77	0.14	0.14	-0.07	-0.50
S10	99.75	75,870	189.1	9.00	1.62	0.14	-0.07	-0.50	-0.50	-0.83	-0.70
S11	11.30	42,971	232.9	21.06	4.55	3.71	3.28	1.58	1.17	0.76	0.11
S12	14.80	46,647	245.9	12.69	6.75	2.95	1.67	1.26	0.85	0.20	-0.43
S13	21.90	55,935	239.8	11.52	3.04	0.94	0.29	-0.34	-0.56	-1.40	-1.62
S14	10.80	34,889	230.6	20.16	7.52	5.83	4.97	2.86	2.02	1.17	0.76
S15	10.60	29,481	232.1	21.06	9.63	5.40	4.14	2.86	2.02	2.45	2.00
S17	15.40	47,279	249.5	13.95	-	3.37	-	1.26	-	0.22	-
S18	11.70	36,046	246.2	7.09	-	3.71	-	2.02	-	0.76	-
S20	29.00	46,730	199.7	5.15	-	-	-	1.35	-	-	-
S23	29.00	38,902	156.3	5.58	-	-	-	0.94	-	-	-
S24	29.00	28,013	135.9	3.46	-	-	-	-0.77	-	-	-
S25	28.50	20,300	206.0	98.82	-	-	-	50.00	-	-	-
S27	29.50	38,866	180.1	5.58	-	-	-	0.94	-	-	-
S28	29.50	44,687	184.8	6.84	-	-	-	0.94	-	-	-
CAL1	15.25	15,197	154.1	43.29	24.03	6.44	2.18	0.99	0.20	-0.09	-0.09
CAL2	102.50	18,410	126.5	16.47	9.27	5.85	2.39	1.73	1.17	0.36	0.27
CAL3	514.25	26,796	85.5	7.74	0.59	0.34	0.04	0.54	0.41	0.04	-0.09

temperature difference between the liquid in the pool and the vapor over the pool, respectively. $T_L - T_V$ at the boiling surface, where L equals 0, has been assumed to be equal to $T_W - T_V$. The measurements at L equal to 0.04, 0.20, 0.37, and 0.54 in. were made with the lower moveable thermocouple, while the measurements at L equal to 0.74, 0.90, 1.07, and 1.24 in. were made with the upper moveable thermocouple (see Sect. V-C-2). For runs S1 through S49, the 1.24 in. position was close to the free liquid surface, while for runs CAL1 through CAL3 the 0.54 in. position was close to the free liquid surface.

The temperature profiles for two typical runs, S6 and S14, are shown in Fig. VII-4. It can be seen that most of the temperature drop occurred within 0.04 in. of the boiling surface, while the temperature drop between the 0.04 in. height and the free liquid surface was very small. Prior to run S34, the moveable thermocouple closer to the boiling plate had failed. While temperature profiles for runs S34 to S49 could not be obtained, the liquid temperatures 1.07 in. from the boiling surface were measured. In most cases, the liquid temperature in the vicinity of the free liquid surface was within 1°F of the vapor temperature, and in no case did the difference exceed 2°F.

1b. Temperature Fluctuations

Fluctuations in the temperature differences ($T_1 - T_V$) were recorded during several runs, where T_1 was the temperature measured by the thermocouple located closest to the top and center line of the copper cylinder (see Fig. C-1). Since the vapor temperatures were found to be steady, any fluctuations in the values of $T_1 - T_V$ must have been due to fluctuations in T_1 . In all cases, fluctuations observed with the recorder were

Fig. VII-4. Temperature Profiles in Selenium Boiling Pool

consistent with fluctuations observed with the potentiometer during the temperature measurements.

For the boiling experiments carried out at pressures near 30, 100, 300, and 760 Torr, fluctuations in T_1 were negligible. For the runs carried out at approximately 10 Torr, small erratic fluctuations in T_1 were observed. These fluctuations ranged between 0.5 and 1.0°F. Due to the erratic nature and the small magnitude of the fluctuations, a frequency for the oscillations could not be determined. In all cases, including the 10 Torr runs, fluctuations in the temperatures measured 0.85 and 1.60 in. from the top of the copper cylinder (T_2 , T_3 , T_5 , T_6) were negligible.

In contrast to the boiling runs, relatively large temperature fluctuations in T_1 were observed during run S25, the non-boiling run. The fluctuations ranged between 10 and 30°F, with an average frequency of approximately 4 cycles/min. Due to the relatively low frequency of temperature fluctuations at the heating surface during the non-boiling run, the temperature wave was able to penetrate the copper cylinder. In the case of the boiling experiments, the relatively high frequency of the temperature fluctuations at the heating surface and the high thermal capacity of the boiling plate and copper cylinder resulted in an almost complete damping of the fluctuations in T_1 . Hyman and Bonilla (H5) observed significant temperature fluctuations near the surface of a horizontal cylinder transferring heat by natural convection to several different liquids. Several investigators (M1, M3) have observed large temperature fluctuations in the boiling of liquid metals; this can be explained by the fact that the frequency of surface-temperature fluctuations in the boiling of liquid metals is relatively low, due to the

unusually high thermal diffusivity of liquid metals. In boiling experiments with water and organic liquids, which have very high frequencies of surface-temperature fluctuations, significant temperature fluctuations have not been observed.

1c. Possible Effects of Corrosion

Due to the corrosiveness of selenium, it was expected that corrosion of the boiling plate might have a significant effect on boiling heat transfer. Thus, several boiling runs made during the first set of experiments (S1 to S33) were duplicated during the second set of experiments (S34 to S49). By successfully reproducing several runs, it was found that corrosion did not have a significant effect on the boiling heat transfer.

Since steady-state conditions in the boiling section of the apparatus could not be obtained during runs S29 through S33 (see Sect. VI), the test chamber and the auxiliary condenser were disconnected before the second set of runs were carried out. After run S33 had been completed, several futile attempts were made to reach steady-state conditions in the copper cylinder. It should be noted that steady-state conditions did exist around the test condenser during runs S29 through S33.

After the apparatus had been disassembled, several selenium samples were removed from the test chamber and auxiliary condenser and tested for the presence of impurities. Spectrographic and x-ray analyses of these samples were performed at Lucius Pitkin, Inc., New York, N.Y., and the Central Research Laboratory of the American Smelting and Refining Co., South Plainfield, N.J. The samples taken from the test chamber contained a total of approximately 10% Fe, Ni, and Cr, the ratios of the

impurities being consistent with the composition of stainless steel 316. Only traces of impurities were found in a sample taken from the auxiliary condenser.

After the second charge of selenium had been inserted into the test chamber, runs S34 through S49 were carried out. Before proceeding with the runs at the highest temperatures, S40 through S49, several runs that were made at the beginning and at the end of the first set of experiments were duplicated. The heat transfer data obtained for runs S35, S36, and S45 were in close agreement with the heat transfer data obtained for runs S6 and S7; run S37 closely reproduced Run S27, while run S38 was consistent with the other runs made at pressures around 30 Torr; and run S39 closely reproduced run S11.

The reproducibility of the heat transfer data that had been obtained at the beginning of the first set of runs indicated that the condition of the boiling plate, with regard to boiling heat transfer, had not changed appreciably during the course of runs S1 through S29 and was essentially the same for the two sets of runs. The reproducibility of the heat transfer data obtained near the end of the first set of runs indicated that the impurities that had accumulated in the selenium did not have a significant effect on the saturation temperature or the boiling heat transfer.

The average thickness of the stainless steel boiling plate that had dissolved in the boiling selenium during the first set of runs was estimated to be 0.004 in. Since the solubilities of the impurities in liquid selenium were very low, the impurities most likely formed a thin selenide layer on top of the boiling plate. The solubility of Fe in liquid

selenium is only 40 ppm at 585°C and 600 ppm at 790°C (D5, D6), while the Ni-Se phase diagram shows Ni to be insoluble in liquid selenium (K8). Data on the solubilities of Cr and Mo in liquid selenium were not available. Before the second selenium charge was added to the test chamber, the boiling plate was visually inspected through the Swagelok fitting at the top of the test chamber. The surface appeared to be somewhat smoother than the original 60 microinch rms surface, with one small pitted area close to the wall of the test chamber.

After run S49 had been completed, temperature measurements in the copper cylinder became erratic. While the power and the pressure had remained constant, the temperatures measured in the copper cylinder dropped nearly 100°F very rapidly, indicating that the selenium may have eaten through the small pitted area of the boiling plate and attacked the copper.

1d. Pressure Drop Across Auxiliary Condenser

During runs S11 through S19, carried out at measured pressures around 10 Torr, it was found that increasing the power beyond approximately 800 watts resulted in a significant pressure drop across the auxiliary condenser. A comparison between the pressure (P_m) observed in the inert gas system and the vapor space temperature (T_v) and the saturation curve for elemental selenium is shown in Fig. VII-5. All of the boiling data obtained at pressures above 10 Torr and some of the data at approximately 10 Torr agree very closely with the saturation curve, while other 10 Torr points show significant disagreement. The points shown in Fig. VII-5 are typical of the data at these pressures. The runs that deviated significantly from the saturation curve were

Fig. VII-5. Comparison between Experimental P_m and T_v and Vapor Pressure Curve for Selenium

S25, the non-boiling run, and S12, S13, S17, and S19.

A comparison between the measured pressures (P_m) and the saturation pressures (P_s) corresponding to the measured T_v for runs S11 through S19 is made in Table VII-6.

Table VII-6. Effect of Power on P_s

<u>Run</u>	<u>Power (watts)</u>	<u>T_v (°F)</u>	<u>P_s (Torr)</u>	<u>P_m (Torr)</u>	<u>H (in.)</u>
S11	796	829.85	11.3	10.25	20.0
S12	955	851.45	14.8	10.75	26.0
S13	1085	882.95	21.9	11.00	30.0
S14	695	826.25	10.8	11.00	16.5
S15	602	825.35	10.6	11.00	13.0
S16	602	825.80	10.7	11.00	23.5
S17	945	853.25	15.4	11.00	26.0
S18	777	832.55	11.7	11.00	18.5
S19	1165	896.36	26.0	11.00	34.0

It can be seen that P_m and P_s agree very closely at the lowest powers, with the difference increasing as the power, and consequently H, are increased. This behavior can be explained by the existence of a pressure drop across the auxiliary condenser and its inlet, caused by relatively high vapor velocities at the low pressures and high powers.

The pressures for runs S11 through S19 reported in Tables A-1 and A-3 were taken as P_s , since P_s for these runs was considered more indicative of actual conditions in the test chamber than the measured P_m . For all other runs, the reported P refers to the measured value. It

should be noted that the boiling heat velocities that could be obtained at pressures around 10 Torr were limited by the pressure drop across the auxiliary condenser, since increasing the power beyond a certain value resulted in an increase in the measured T_v and the corresponding P_s .

2. Empirical Correlating Equations

Empirical correlating equations for the boiling heat transfer data shown in Fig. VII-3 have been developed. The solid lines in Fig. VII-3 were obtained by means of a least squares procedure developed by Ergun (E4) for fitting a series of parallel straight lines to data.

The pool boiling data of several investigators (C12, B13, C7, A2, M3), obtained with a variety of test liquids and over a wide range of pressures, show that pressure generally has an insignificant effect on the slope of the boiling curve, when $\log q_b$ is plotted against $\log (T_w - T_v)$. It has also been found that experimental log-log boiling curves are usually linear over most of the nucleate boiling region, curvature being limited to the critical heat velocities and to relatively low heat velocities, where boiling is not fully developed. In addition, the semi-empirical equations for predicting nucleate boiling heat transfer that were discussed in Sect. III generally result in a series of nearly parallel straight lines when $\log q_b$ is plotted against $\log (T_w - T_v)$, with pressure as a parameter. In correlating the boiling heat transfer data all runs shown in Fig. VII-3 (except S1, S2, S34, and S25) were therefore utilized to determine the best average slope of the boiling curves obtained at the approximately constant pressures of 10, 30, 100, 300, and 760 Torr, considered as in separate groups.

In carrying out the least squares calculations, the sums of the squares of the differences between the logarithms of the experimental and predicted heat velocities were minimized. Due to the high values of $(T_w - T_v)$, the implicit assumption that the experimental values of $(T_w - T_v)$ were free of error was reasonable. The least squares equations that were determined for the constant pressure data shown in Fig. VII-3 are presented in Table VII-7.

Table VII-7. Least Squares Equations for the Selenium Constant Pressure Pool Boiling Data

Pressure Group (Torr)	Runs Included	Boiling Heat Velocity, q_b (Btu/sq ft-hr)
10	S11, S14, S15, S16, S18, S39	$0.716 (T_w - T_v)^{1.981}$
30	S20, S23, S24, S27, S28, S37, S38	$1.518 (T_w - T_v)^{1.981}$
100	S3, S4, S5, S6, S7, S8, S9, S10, S35, S36, S45	$2.507 (T_w - T_v)^{1.981}$
300	S40, S41, S42, S43, S44	$5.730 (T_w - T_v)^{1.981}$
760	S46, S47, S48, S49	$13.155 (T_w - T_v)^{1.981}$

The common slope (1.981) and the intercept of each constant pressure line at $\Delta T = 1$ were determined from the equations derived by Ergun (E4).

The general equation for calculating q_b can be written:

$$q_b = A(T_w - T_v)^{1.981}, \quad (\text{VII-1})$$

where A is a function of pressure. The functional relationship between A and P was determined by computing the value of A for each data point (including runs S12, S13, and S17, clearly not in any of the pressure

groups in Table VII-7) and by plotting A versus P, as shown in Fig.

VII-6. The least squares straight line through the data is:

$$A = 0.1521P^{0.6422} \quad (\text{VII-2})$$

By substituting Eq. VII-2 into Eq. VII-1, the following correlating equation for the boiling heat transfer data was obtained:

$$q_b = 0.1521(T_w - T_v)^{1.981} (P)^{0.6422}, \quad (\text{VII-3})$$

where q_b , $(T_w - T_v)$, and P have the units of Btu/sq ft-hr, °F, and Torr, respectively. The probable error in the value of the boiling heat velocity calculated by Eq. VII-3 was computed to be +12.5% or -11%; the probable limits can be found by multiplying and dividing the calculated value by 1.125.

The data in Fig. VII-6 were also fit (by least squares) with a parabola, with insignificant improvement in the fit. An improved fit was obtained by the least squares curve,

$$A = 0.1956P^{1/2} + 8.379 \times 10^{-3}P, \quad (\text{VII-4})$$

resulting in the following correlating equation:

$$q_b = (T_w - T_v)^{1.981} (0.1956P^{1/2} + 8.379 \times 10^{-3}P). \quad (\text{VII-5})$$

The probable error in the value of q_b calculated by Eq. VII-5 was found to be +11.2% or -9%, the geometric limits of probable error being $\pm 1.112^x$.

3. Semi-Empirical Correlating Equations

The Rohsenow equation (Eq. III-11) was used to correlate the boiling heat transfer data. The constants c_{sf} , u , and s in Eq. III-11 were

Fig. VII-6. Constant in Empirical Correlating Equation vs. Pressure

determined graphically by the procedure developed by Rohsenow (R3). The final correlation of the data, as shown in Fig. VII-7, resulted in the following correlating equation:

$$\frac{c_{pl}(T_w - T_v)}{\lambda} = 0.035 \left\{ \frac{q_b}{\mu_l \lambda} \sqrt{\frac{g_c \sigma}{g(\rho_l - \rho_v)}} \right\}^{0.33} \left\{ \frac{c_{pl} \mu_l}{k_l} \right\}^{0.92} \quad (\text{VII-6})$$

The value of u , determined from the slope of the line in Fig. VII-7, is in perfect agreement with the value of 0.33 that was obtained by Rohsenow for the fluid-surface combinations that he investigated, while the value of s (0.92) is within the range (0.8 to 2.0) that Rohsenow obtained. Equation VII-6 fits the data in Fig. VII-7 with a standard deviation of 22%.

The experimental boiling heat transfer data have been compared to the data predicted by Eqs. III-5 through III-10. The predicted and experimental heat velocities obtained at a pressure of approximately 1 atm are plotted against the boiling temperature difference in Fig. VII-8. The heat velocities predicted by the equations of Forster and Zuber (F2), Forster and Greif (F3), and Levy (L3) are within approximately 25% of the experimental heat velocities, while the heat velocities predicted by the equations of McNelly (M6), Hughmark (H4), and Chang and Snyder (C2) exceed the experimental heat velocities by approximately 100%. In all cases, the agreement between the predicted and experimental heat velocities was considered to be satisfactory.

A comparison between the predicted and experimental heat velocities at pressures of 10, 30, 100, 300, and 760 Torr is presented in Table VII-8, where q_b , q_{FZ} , q_{FG} , q_M , q_L , q_H , and q_C represent the experimental

Fig. VII-7. Correlation of Boiling Heat Transfer Data with the Rohsenow Equation (R3)

Fig. VII-8. Experimental and Predicted Pool Boiling Heat Transfer Data at Atmospheric Pressure

Table VII-8. Comparison Between Predicted and Experimental Boiling Heat Velocities

P (Torr)	$T_w - T_v$ (°F)	q_b (Btu/ft ² -hr)	$\left(\frac{q_{FZ}}{q_b}\right)$	$\left(\frac{q_{FG}}{q_b}\right)$	$\left(\frac{q_M}{q_b}\right)$	$\left(\frac{q_L}{q_b}\right)$	$\left(\frac{q_H}{q_b}\right)$	$\left(\frac{q_G}{q_b}\right)$
760	70	60,000	0.73	0.68	1.69	0.75	1.79	1.73
760	80	78,000	0.76	0.73	2.00	0.86	2.00	1.92
760	90	99,000	0.78	0.79	2.33	0.97	2.22	2.08
300	110	63,500	1.15	1.32	7.13	2.04	2.70	3.72
300	120	75,000	1.21	1.45	8.32	2.22	3.03	4.17
300	130	89,000	1.25	1.57	9.08	2.44	3.45	4.55
300	140	102,000	1.30	1.70	10.00	2.63	4.00	5.00
100	150	51,500	1.75	2.63	25.6	5.55	4.34	7.70
100	160	58,500	1.85	2.94	27.8	5.88	5.00	8.33
100	170	66,000	1.96	3.22	30.3	6.25	5.88	9.10
100	180	74,000	2.04	3.57	32.3	6.67	6.67	10.00
100	190	82,500	2.13	4.00	34.4	7.13	7.70	11.10
30	160	35,500	1.64	3.03	47.5	9.10	3.22	9.10
30	170	40,000	1.76	3.45	52.7	10.00	3.85	10.00
30	180	44,500	1.85	3.85	55.5	10.55	4.76	10.75
30	190	50,000	1.96	4.34	58.8	11.10	5.55	11.75
30	200	55,000	2.08	5.00	62.5	11.75	6.67	13.15
10	220	31,000	3.22	10.00	154.0	28.6	11.80	25.6
10	230	34,000	3.45	11.78	164.0	29.4	14.50	28.5
10	240	37,000	3.70	13.50	172.5	31.3	17.60	31.2
10	250	40,500	3.85	15.15	182.0	32.2	21.3	34.4

heat velocities and the heat velocities predicted by the equations of Forster and Zuber, Forster and Greif, McNelly, Levy, Hughmark, and Chang and Snyder, respectively. The experimental heat velocities were obtained from the least squares lines shown in Fig. VII-3. The ratio of the predicted to experimental heat velocities is plotted against pressure in Fig. VII-9 for q_b equal to 50,000 Btu/sq ft-hr. In all cases, the ratio q_{pred}/q_b increases markedly as the pressure is decreased; at the lowest pressure investigated (10 Torr), the predicted heat velocities exceed the experimental heat velocities by factors of approximately 4, 44, 15, 250, 25, and 35 for Eqs. III-5 to III-10, respectively.

It can be seen from Figs. VII-8 and VII-9 that the internal agreement between the six predicting equations is relatively poor, especially at the lower pressures. No reason is evident for the division of the predicted heat transfer data into two groups at atmospheric pressure. Each of the six predicting equations was based on heat transfer data for a variety of liquids, nearly all of the data being obtained from horizontal boiling surfaces; in addition, the orientation and geometry of pool boiling surfaces have generally been found to have only a slight effect on the heat transfer (C2, C3, M6). As shown in Fig. VII-9, the division of the predicted heat transfer data into two groups did not occur at the lower pressures.

Equations III-5 through III-10 have been tested for a wide variety of liquids and over wide ranges of pressure (N4, F2, F3, C2, M6, L3, C1, C3, B4, N3). These equations have generally been found to satisfactorily predict pool boiling heat transfer data for liquid metals, as well as for water and organic liquids (see Sect. III-A-4). Experimental heat transfer

Fig. VII-9. Ratio of Predicted to Experimental Pool Boiling Heat Velocities vs. Pressure, for $q_b = 50,000$ Btu/sq ft-hr

data obtained at low, moderate, and high pressures (from 6 Torr to 50 atm, and higher) have generally been predicted with the same degree of accuracy. A possible explanation for the discrepancy between the predicted and experimental heat velocities at the lower pressures, as shown in Fig. VII-9, is discussed below.

4. Possible Effect of Molecular Structure on Boiling Heat Transfer

As described in Appendix F, liquid selenium is composed of Se_8 (and possibly Se_6) rings and long chains of selenium atoms. For the purposes of the present discussion, it has been assumed that the rings consist primarily of Se_8 molecules. At the conditions of the boiling experiments, the rings constituted 14 to 20 wt % of the liquid. Because the rings are smaller and consequently more volatile than the long chains, and because Se_8 rings are the largest molecules found in the vapor phase, the rings were considered to be the principal species that evaporate into bubbles during nucleate boiling.

The probable mechanism of bubble growth is shown in Fig. VII-10.

Fig. VII-10. Mechanism of Bubble Growth

As rings enter the vapor at the bubble interface, they must be replaced for continued bubble growth. The rings can be replaced in two ways: by diffusion of rings from the liquid surrounding the bubble, and by the splitting of long chains at the bubble interface into shorter chains and rings.

After the bubble departs from the heating surface, the liquid surrounding the bubble rushes to cover the vacated spot at the heating surface, entrapping some vapor in the nucleation site. As power is supplied to the heating surface, its temperature and superheat increase until the vapor pressure inside the nucleus is sufficient to again initiate bubble growth (see Sect. III-A and Eq. III-1). Since the concentration of rings in the liquid surrounding the nucleation site would be reduced as rings enter the vapor, the rate of pressure increase in the nucleus would be limited by the availability of rings at the vapor-liquid interface, and consequently by the diffusional resistance and the reaction kinetics in the liquid. With the liquid in the vicinity of the vapor-liquid interface being partially stripped of rings, the superheat required to produce enough vapor pressure inside the cavity to cause bubble nucleation would be increased. Since the resistance to diffusion of rings through the long chains and the rate of formation of rings by chemical reaction are functions of the liquid structure, the liquid composition could have a significant effect on both bubble nucleation and bubble growth, and consequently on nucleate boiling heat transfer.

The ratio of the predicted to experimental heat velocities is plotted against the average chain length of the liquid (Z) in Fig. VII-11 for $q_b = 50,000$ Btu/sq ft-hr. The predicted heat velocities were evaluated at the ΔT corresponding to q_b . Z was evaluated at the saturation temperature and

Fig. VII-11. Ratio of Predicted to Experimental Heat Velocities vs. Average Chain Length of Liquid Selenium, for $q_b = 50,000$ Btu/sq ft-hr (Both q_{pred} and q_b at same ΔT)

obtained from Fig. F-1. For each of the six predicting equations, the ratio q_{pred}/q_b approaches 1 as Z decreases. This result is consistent with the above model, since the rate of production of rings by chemical reaction should be increased and the diffusional resistance should be decreased by a reduction in the average chain length. Bubble nucleation and growth should then be enhanced by a reduction in the average chain length of the liquid, with the experimental heat velocities approaching the predicted heat velocities at the lower average chain lengths.

Sternling and Tichacek (S6) obtained nucleate pool boiling heat transfer data at atmospheric pressure for fourteen binary mixtures of high relative volatility. Their boiling heat transfer data for pure water, pure ethylene glycol, and water-ethylene glycol mixtures, and for pure methanol, pure amyl alcohol, and methanol-amyl alcohol mixtures are shown in Fig. VII-12. Such boiling mixtures are analogous to boiling selenium, with the volatile component A and the nonvolatile component B corresponding to the Se_8 rings and the long polymer chains of Se atoms, respectively. However, unlike boiling selenium, the binary systems are non-reacting. For all the mixtures tested, the boiling heat transfer coefficients decreased markedly as material B was added to the pure volatile component A; h_b for the mixtures was generally considerably lower than the average of the h_b for the pure components at atmospheric pressure.

The surprisingly low h_b for the H_2O -glycol and methanol-amyl alcohol mixtures, as shown in Fig. VII-12, can be explained by a partial separation of the volatile and nonvolatile components in the liquid layer surrounding nucleation sites and by the presence of a resistance to mass

(a) BINARY SYSTEM-METHANOL AND AMYL ALCOHOL

(b) BINARY SYSTEM-WATER AND ETHYLENE GLYCOL

Fig. VII-12. Nucleate Pool Boiling Heat Transfer Data for Binary Mixtures of High Relative Volatility, at Atmospheric Pressure (S6)

transfer as the volatile component diffused towards growing bubbles. As component A would be preferentially stripped from the liquid in the vicinity of nucleation sites, nonvolatile material would tend to accumulate around the sites. The saturation temperature T_s for a mixture with significant amounts of both A and B is considerably lower than T_s for pure B. Since the mole fraction of A in the liquid at the vapor-liquid interface is much lower than that in the bulk liquid (due to the stripping of A from the liquid), and since the vapor pressure of B at T_s of the mixture is relatively low, the superheat required to produce enough vapor pressure inside cavities to cause bubble nucleation (see Eq. III-1) can be considerably higher for a mixture than for pure B.

In addition to the effect on bubble nucleation, the presence of significant amounts of both A and B in the liquid can also result in a diffusional resistance in the liquid layer surrounding growing bubbles. Except for mixtures with mole fractions of B close to 1, the saturated vapor would consist primarily of A. The presence of B in the liquid would then cause a resistance to the diffusion of A towards growing bubbles, thereby limiting bubble growth rates. Van Stralen (V2, V3) has, in fact, predicted and confirmed experimentally the existence of a minimum bubble growth rate at a certain low concentration of the more volatile component in the nucleate boiling of binary mixtures.

It is interesting to note that, for the binary systems investigated by Sternling and Tichacek, the slopes of the boiling curves are generally lower for the mixtures than for the pure components. In the case of a pure component, the reciprocal slope of a boiling curve is a measure of the resistance of the nuclei to boiling. The larger reciprocal slope for

a mixture is then a measure of the resistance of the nuclei plus the diffusional resistance of the more volatile component to reaching the bubble interface. In the case of the constant-pressure boiling data for selenium, shown in Fig. VII-3, the estimated reciprocal slopes for the constant-pressure data groups increase continuously with decreasing pressure from 760 to 30 Torr, as expected (in view of the increase in the diffusional and kinetic resistance with decreasing pressure), but then decrease at 10 Torr. However, due to the scatter in the data and the limited number of data points at each constant pressure, it is not possible to accurately determine the slope for each constant pressure group. It should be noted that in correlating the boiling heat transfer data, the constant pressure groups in Fig. VII-3 were assumed to have the same slope.

The variation of the ratio q_{pred}/q_b with P for boiling selenium (Fig. VII-9) and the boiling curves for binary mixtures of high relative volatility (Fig. VII-12) can both be explained in terms of the availability of volatile components in the liquid layer surrounding nucleation sites and growing bubbles, and consequently in terms of the compositions of the boiling liquids. In the case of the binary mixtures studied by Sternling and Tichacek, the concentration of volatile material in the liquid was varied directly. In the case of boiling selenium, the diffusional resistance and the reaction kinetics in the liquid were varied by varying P_s and T_s , and consequently the composition of the liquid phase. Due to the correlation between the ratio q_{pred}/q_b and the average chain length of liquid selenium (Fig. VII-11), and due to the similarities in the results obtained for binary mixtures of high relative volatility and for selenium, the mechanism proposed for bubble growth and nucleation in the boiling of selenium (see Fig. VII-10) was considered to be a

reasonable explanation for the high values of q_{pred}/q_b at the lower pressures investigated.

C. Condensation of Selenium

1. Heat Transfer Data

The experimental data that were recorded during the condensation experiments are presented and discussed in Appendix A.

The results of the condensation experiments are presented in Table VII-9, where H is the height of the condensing ring in the auxiliary condenser. Most runs were made at pressures of approximately 10, 30, 100, 300, and 760 Torr, with only runs S12, S13, S17, and S19 being at intermediate pressures (see Sect. VII-B-1). The condensation data in the 10, 30, 100, 300, and 760 Torr groups are shown in Fig. VII-13, where the condensing heat velocity (q_c) is plotted against the condensing temperature difference ($T_v - T_o$). Since nearly identical heat transfer data were obtained for several runs, many of the data points in Fig. VII-13 actually represent more than one run. The duplicating runs shown in Fig. VII-13 were made to check the reproducibility of results and to test for the possible effect of noncondensables on the heat transfer data.

It can be seen from Fig. VII-13 that the heat transfer coefficient (h_c) increased as the pressure (and saturation temperature) was increased. For liquids whose condensing heat transfer data agree closely with the Nusselt equation (Eq. III-13), h_c generally increases slightly with temperature and pressure, since the viscosity, which overshadows the effect of the other properties, decreases with temperature, favoring thin condensate films and higher heat transfer coefficients. In condensation

Table VII-9. Results of Condensation Experiments with Selenium
(Runs are in Chronological Order)

Run	T_v (°F)	P (Torr)	H (in.)	q_c (Btu/ft ² -hr)	$T_v - T_o$ (°F)	h_c (Btu/ft ² -hr-°F)
S1	1024.25	101.25	10.5	37,846	281.5	134.5
S2	1025.15	101.75	16.0	36,755	271.9	135.2
S3	1026.50	101.75	24.5	33,870	262.2	129.2
S4	1025.60	100.00	31.0	32,331	272.4	118.7
S5	1026.50	100.75	35.0	35,444	316.4	112.0
S6	1022.90	100.00	18.0	33,757	317.6	106.3
S7	1021.64	98.75	14.0	34,522	337.4	102.3
S8	1022.45	98.75	14.5	27,634	245.7	112.5
S9	1022.00	98.75	11.0	27,611	247.3	111.6
S10	1024.70	99.75	25.0	30,927	307.6	100.5
S11	829.85	11.30	20.0	23,517	267.5	87.9
S12	851.45	14.80	26.0	25,536	289.7	88.1
S13	882.95	21.90	30.0	27,359	311.7	87.8
S14	826.25	10.80	16.5	20,701	231.7	89.4
S15	825.35	10.60	13.0	18,597	192.9	96.4
S16	825.80	10.70	23.5	18,654	192.6	96.9
S17	853.25	15.40	26.0	25,119	300.2	83.7
S18	832.55	11.70	18.5	22,910	273.7	83.7
S19	896.36	26.00	34.0	25,618	271.8	94.2
S20	904.28	29.00	16.0	28,471	332.0	85.7
S21	910.40	29.00	27.0	25,506	270.2	94.4
S22	908.60	29.00	22.0	25,213	274.1	92.0
S23	903.92	29.00	12.5	25,010	272.8	91.7
S24	904.28	29.00	9.0	25,019	273.1	91.6
S25	818.15	28.50	0.0	19,046	318.5	59.8
S26	906.80	29.00	19.5	25,126	272.3	92.3
S27	905.90	29.50	17.0	20,681	207.7	99.6
S28	905.90	29.50	14.0	27,596	347.7	79.4
S29	906.35	30.00	25.0	27,435	348.5	78.7
S30	1024.25	100.25	9.0	32,953	348.8	94.5
S31	1024.25	100.25	9.5	27,008	253.7	106.5
S32	923.72	30.00	10.5	20,824	223.6	93.1
S33	927.95	29.50	9.5	21,071	213.8	98.6
S34	1024.25	99.50	14.0	34,857	276.9	125.9
S35	1024.25	100.00	19.0	35,472	277.3	127.9
S36	1023.35	100.00	13.0	36,931	294.7	125.3
S37	904.64	29.50	18.0	22,083	190.3	116.1
S38	907.25	29.50	21.0	26,727	250.0	106.9
S39	822.65	10.00	21.0	24,060	252.6	95.3
S40	1146.20	294.5	14.0	43,473	269.4	161.4
S41	1146.20	298.4	19.0	40,910	242.2	168.9
S42	1146.65	297.4	23.0	41,043	242.9	169.0
S43	1147.10	299.8	27.0	41,078	245.1	167.6
S44	1146.20	298.8	12.0	41,108	241.7	170.1
S45	1022.00	99.50	14.0	35,880	327.9	109.4
S46	1272.20	763.0	9.0	50,932	273.2	186.4
S47	1272.20	764.5	14.0	51,481	265.7	193.8
S48	1272.20	768.5	18.0	44,139	196.8	224.3
S49	1272.20	768.7	22.0	44,140	193.8	227.7

Fig. VII-13. Heat Velocity vs. Temperature Difference for Condensing Selenium

studies with liquid metals, Sukhatme and Rohsenow (S7), Aladyev (A3), and Labuntsov and Smirnov (L1) found that h_c increased with pressure, whereas Misra (M8) found that h_c decreased with increasing pressure.

The q_c reported in Table VII-9 and shown in Fig. VII-13 was determined from the sensible heat gained by the air coolant as it passed through the test condenser. $T_v - T_o$ was obtained from the eight measured temperatures in the silver layer of the condensing tube, corrected to the surface of the condensing tube. Both q_c and $T_v - T_o$ represent average values over the surface of the condensing tube. The methods and sample calculations are discussed in Appendix C.

To test for the presence, and any appreciable effect, of noncondensables on the condensing heat transfer data, the height of the condensing ring in the auxiliary condenser (H) was varied while the selenium vapor pressure and the cooling air flow rate were held constant. This would change the path length for the backward diffusion of inert gas into the test chamber after it had been swept to the top of the auxiliary condenser, and thus change its steady state concentration in the chamber. In some runs the heater power was varied to achieve the change in condensing ring position. In other runs, the power, as well as the pressure and air flow rate, was held constant and the condensing ring was raised by adding thermal insulation around the bottom portion of the auxiliary condenser. The following groups of runs were made at constant pressure and air flow rate, but with varying power: S8 and S9; S21, S22, S23, S24, and S26; S41, S42, S43, and S44; S46 and S47; and S48 and S49. Runs S16 and S29 were carried out under the same conditions of pressure, air flow rate, and power as runs S15 and S28, respectively, but with the bottom portion of the auxiliary condenser well-insulated with Fiberfrax Lo-Con blanket

over a 13 in. length. Close scrutiny of the heat transfer data in Table VII-9 for these runs reveals that wide variations in H had insignificant effect on the condensation heat transfer data. It has therefore been concluded that noncondensables had no effect on the condensation heat transfer data (see Sect. VI). The reproducibility of results is further evidence that noncondensables were effectively eliminated from the test chamber during these experiments. Had noncondensables been present in even a minute amount, the process of vapor migration towards the test condenser would have considerably concentrated them on its surface, yielding an exaggerated effect on the apparent condensing heat transfer coefficient; in view of the variations in condensing and refluxing conditions, results as reproducible as these would seem very unlikely, in agreement with previous studies (E3, MJ).

During the non-boiling run S25 (see Sect. VII-B-1), the bottom of the auxiliary condenser was cold. The relatively low h_c for run S25 must have been due to the presence of inerts in the test chamber during this run, since the rate of boiling did not reach the possible rate of condensation on the test condenser. The relatively high h_c for runs S1, S2, S3, S34, S35, and S36, in the 100 Torr group, and also for runs S37, S38, and S39, in the 30 and 10 Torr groups, must be due to the fact that they were the first runs after the two fresh charges of selenium had been added to the test chamber (see Sect. VII-B-1c). Dropwise condensation over portions of the condensing tube, resulting from incomplete wetting of the condensing surface, could therefore have occurred, causing higher values of h_c .

The reproducibility of the heat transfer data collected after run S3 indicated that nearly complete wetting of the condensing surface had

occurred after run S3 had been completed, and that filmwise condensation therefore occurred over most of the condensing surface during runs S4 through S33. In view of the reproducibility of results during the second set of experiments, after run S39, runs S40 through S49 were also considered to have been predominantly filmwise condensation. Due to the relatively high condensation rates and condensate temperatures during runs S40 through S49, filmwise condensation should indeed have been enhanced during these runs.

2. Nusselt Equation

The experimental condensing heat transfer data have been compared to heat transfer results predicted by the Nusselt equation (Eq. III-13). The experimental and theoretical heat transfer coefficients are presented in Table VII-10, where h_{Nu} represents the Nusselt result. The ratio h_c/h_{Nu} generally fell between 0.4 and 0.5, there being no significant variation in the ratio with pressure or with q_c . The linear correlation coefficients between h_c/h_{Nu} and P and between h_c/h_{Nu} and q_c were computed to be 0.18 and 0.19, respectively, showing that the linear correlation between the above pairs of variables was not significant at the 95% probability level; a correlation coefficient of approximately 0.32 would be required to show significant linear correlation at this level. In contrast, the linear correlation coefficient between q_c and P was computed to be 0.88, indicating a strong linear relationship between q_c and P , which is obvious from Fig. VII-13.

In the condensation of liquid metals, h_c is usually around an order of magnitude smaller than h_{Nu} . The generally accepted explanation for the discrepancy between h_c and h_{Nu} is the presence of a thermal resistance

Table VII-10. Comparison Between Experimental and Theoretical Condensing Heat Transfer Coefficients

Run	P (Torr)	q_c (Btu/ft ² -hr)	h_c (Btu/ft ² -hr-°F)	h_{Nu} (Btu/ft ² -hr-°F)	$\frac{h_c}{h_{Nu}}$
S1	101.25	37,846	134.5	252.1	0.533
S2	101.75	36,755	135.2	257.9	0.524
S3	101.75	33,870	129.2	264.2	0.490
S4	100.00	32,331	118.7	257.9	0.460
S5	100.75	35,444	112.0	234.8	0.477
S6	100.00	33,757	106.3	232.4	0.457
S7	98.75	34,522	102.3	222.1	0.461
S8	98.75	27,634	112.5	272.1	0.413
S9	98.75	27,611	111.6	270.9	0.412
S10	99.75	30,927	100.5	238.4	0.422
S11	11.30	23,517	87.9	167.4	0.525
S12	14.80	25,536	88.1	166.8	0.528
S13	21.90	27,359	87.8	170.9	0.514
S14	10.80	20,701	89.4	183.2	0.488
S15	10.60	18,597	96.4	204.4	0.472
S16	10.70	18,654	96.9	204.8	0.473
S17	15.40	25,119	83.7	163.0	0.513
S18	11.70	22,910	83.7	165.7	0.505
S19	26.00	25,618	94.2	195.6	0.482
S20	29.00	28,471	85.7	171.5	0.500
S21	29.00	25,506	94.4	203.0	0.465
S22	29.00	25,213	92.0	200.2	0.460
S23	29.00	25,010	91.7	198.7	0.462
S24	29.00	25,019	91.6	198.7	0.461
S25	28.50	19,046	59.8	158.5	0.377
S26	29.00	25,126	92.3	200.3	0.461
S27	29.50	20,681	99.6	235.9	0.422
S28	29.50	27,596	79.4	165.6	0.479
S29	30.00	27,435	78.7	165.4	0.476
S30	100.25	32,953	94.5	217.9	0.434
S31	100.25	27,008	106.5	268.1	0.397
S32	30.00	20,824	93.1	235.2	0.396
S33	29.50	21,071	98.6	243.4	0.405
S34	99.50	34,857	125.9	254.7	0.494
S35	100.00	35,472	127.9	254.4	0.503
S36	100.00	36,931	125.3	244.5	0.513
S37	29.50	22,083	116.1	246.6	0.471
S38	29.50	26,727	106.9	212.1	0.504
S39	10.00	24,060	95.3	171.1	0.557
S40	294.5	43,473	161.4	321.4	0.502
S41	298.4	40,910	168.9	339.5	0.498
S42	297.4	41,043	169.0	339.3	0.498
S43	299.8	41,078	167.6	337.9	0.496
S44	298.8	41,108	170.1	339.9	0.500
S45	99.50	35,880	109.4	226.9	0.482
S46	763.0	50,932	186.4	385.6	0.484
S47	764.5	51,481	193.8	390.8	0.496
S48	768.5	44,139	224.3	446.0	0.503
S49	768.7	44,140	227.7	448.8	0.508

at the liquid-vapor interface; due to the very high thermal conductivity of the liquid metal condensate film, any appreciable resistance at the liquid-vapor interface can become controlling. Misra (M8) and Engelbrecht (E3) found wide variations in the ratio h_c/h_{Nu} with pressure in the condensation of liquid metals. Other possible explanations for the low h_c/h_{Nu} for liquid metals are: a "contact" resistance at the solid-liquid interface, an unexplained thickening of the condensate film, and a resistance in the vapor phase due to the presence of noncondensable gas. It has been concluded by several investigators that a solid-liquid interfacial resistance and an unexpected thickening of the condensate film can generally not be significant in the condensation of liquid metal vapors, and that noncondensables can be eliminated by the careful design of experiments. Thus, a thermal resistance at the vapor-liquid interface is generally considered to be the only reasonable explanation for the low values of h_c/h_{Nu} obtained by several investigators (S7, W3, K6, B6, L2).

In these condensation experiments with selenium, it was concluded that noncondensables were effectively eliminated from the test chamber and therefore had no effect on the heat transfer data (see Sects. VI and VII-C-1). A possible thickening of the condensate film, caused by the countercurrent flow of selenium vapor moving towards the auxiliary condenser or by dripping of condensate from the auxiliary condenser, was ruled out by the fact that the heat transfer data did not vary with the power, and consequently with the vapor velocity or the rate of condensation in the auxiliary condenser. The apparatus was, in fact, constructed in such a way as to prevent dripping from the auxiliary condenser onto

the test condenser (see Sect. V-C-2). Due to the high temperatures of the condensate, the reactivity of liquid selenium, and the reproducibility of the heat transfer data, it is very unlikely that any contaminant on the surface of the condensing tube could have caused a significant thermal resistance. It should be noted that for those runs where dropwise condensation may have been significant, the values of h_c/h_{Nu} would be higher than for complete filmwise condensation, not lower.

Analytical expressions have been derived to account for several of the simplifying assumptions made by Nusselt (N4) in deriving Eq. III-12 (see Sect. III-B-1). The effects of these simplifying assumptions on the values of h_c/h_{Nu} for selenium, shown in Table VII-10, have been determined. Rohsenow (R4) calculated the actual temperature distribution in the condensate film. His correction to Eq. III-13, when applied to the case of condensing selenium, increased h_{Nu} by 2% or less between 10 and 760 Torr. Chen (C4) considered the effects of convection and acceleration in the liquid film and vapor drag at the surface of the condensate film. His correction increased h_{Nu} for condensing selenium by less than 2% over the ranges of q_c and P investigated in this research program. The increase in h_{Nu} , due to the corrections of Rohsenow and Chen, ranged from approximately 2.4% (at 760 Torr) to 3.9% (at 10 Torr). Thus, these corrections resulted in a slight decrease in h_c/h_{Nu} , not an increase. It should be noted that Eq. III-13, a modified form of the Nusselt equation that was used to calculate h_{Nu} , accounts for the sub-cooling of the condensate film. The physical property data of selenium that were employed in Eq. III-13 were considered to be reliable (see Appendix E).

In view of the above, it was concluded that a liquid-vapor interfacial resistance was the most reasonable explanation for the relatively low h_c/h_{Nu} for selenium.

3. Condensation Coefficients

The theoretical liquid-vapor interfacial resistance was analyzed by means of the kinetic theory of condensation, employed recently to interpret and correlate heat transfer data in the condensation of liquid metal vapors (see Sect. III-B). The condensation coefficients (σ_c) were calculated from the condensing heat transfer data by the following equation, derived from Eqs. III-18 and III-19:

$$\sigma_c = \frac{\frac{2q_c}{\lambda'}}{\frac{q_c}{\lambda'} + \left[\frac{2M}{\pi R} \right]^{1/2} \left[\frac{P_v}{T_v^{1/2}} - \frac{P_{s\ell}}{T_{s\ell}^{1/2}} \right]} \quad (\text{VII-7})$$

The σ_c calculated is presented in Table VII-11. T_v , T_o , and P_v were obtained experimentally, while $T_{s\ell}$ and $P_{s\ell}$ were calculated by Eqs. III-20 and E1, respectively. The molecular weight of the vapor (M), evaluated at $T_{s\ell}$, was obtained from Fig. F-4. σ_c was also calculated with M evaluated at T_v and at $(T_v + T_{s\ell})/2$. In all cases, the calculated values of σ_c agreed within $\pm 5\%$.

The condensation coefficient is generally considered to be a function of the state of the condensate surface. σ_c has therefore been plotted against $P_{s\ell}$ in Fig. VII-14, where the dashed lines represent data for sodium, potassium, and mercury. The data for the liquid metals were obtained from a compilation of condensation coefficients made by

Table VII-11. Molecular Condensation Coefficients

Run	T _v (°F)	T _{sat} (°F)	T _o (°F)	P _v (Torr)	P _{sat} (Torr)	σ _c
S1	1024.25	873.26	742.75	101.25	19.51	0.01161
S2	1025.15	876.13	753.25	101.75	20.19	0.01136
S3	1026.50	872.42	764.30	101.75	19.31	0.01035
S4	1025.60	857.68	753.20	100.00	16.13	0.009572
S5	1026.50	839.54	710.10	100.75	12.85	0.009730
S6	1022.90	828.65	705.30	100.00	11.17	0.009108
S7	1021.64	817.79	684.24	98.75	9.70	0.009165
S8	1022.45	858.02	776.75	98.75	16.19	0.008402
S9	1022.00	856.27	774.70	98.75	15.85	0.008346
S10	1024.70	824.47	717.10	99.75	10.58	0.008305
S11	829.85	688.32	562.35	11.30	1.46	0.05752
S12	851.45	699.85	561.75	14.80	1.76	0.04682
S13	882.95	714.74	571.25	21.90	2.22	0.03298
S14	826.25	691.60	594.55	10.80	1.54	0.05478
S15	825.35	708.40	632.45	10.60	2.01	0.05453
S16	825.80	709.27	633.20	10.70	2.04	0.05427
S17	853.25	692.87	553.05	15.40	1.57	0.04310
S18	832.55	682.45	558.85	11.70	1.33	0.05285
S19	896.36	737.52	624.56	26.00	3.15	0.02722
S20	904.28	721.21	572.28	29.00	2.46	0.02517
S21	910.40	747.23	640.20	29.00	3.63	0.02444
S22	908.60	741.73	634.50	29.00	3.35	0.02380
S23	903.92	738.46	631.12	29.00	3.19	0.02347
S24	904.28	738.53	631.18	29.00	3.19	0.02348
S25	818.15	626.25	499.65	28.50	0.50	0.01579
S26	906.80	741.34	634.50	29.00	3.33	0.02373
S27	905.90	768.78	698.20	29.50	4.96	0.02131
S28	905.90	708.52	558.20	29.50	2.02	0.02325
S29	906.35	707.31	557.85	30.00	1.98	0.02265
S30	1024.25	804.17	675.45	100.25	8.09	0.008328
S31	1024.25	850.49	770.55	100.25	14.75	0.007854
S32	923.72	770.47	700.12	30.00	5.08	0.02087
S33	927.95	783.01	714.15	29.50	6.06	0.02267
S34	1024.25	863.84	747.35	99.50	17.39	0.01059
S35	1024.25	866.14	746.95	100.00	17.89	0.01079
S36	1023.35	859.34	728.65	100.00	16.46	0.01092
S37	904.64	788.15	714.34	29.50	6.51	0.02476
S38	907.25	765.82	657.25	29.50	4.76	0.02679
S39	822.65	696.18	570.05	10.00	1.66	0.07006
S40	1146.20	990.32	876.80	294.50	71.50	0.005002
S41	1146.20	1004.08	904.00	298.40	82.16	0.004949
S42	1146.65	1004.28	903.75	297.40	82.32	0.004991
S43	1147.10	1002.93	902.00	299.80	81.22	0.004904
S44	1146.20	1005.12	904.50	298.80	83.02	0.004988
S45	1022.00	830.85	694.10	99.50	11.50	0.009761
S46	1272.20	1107.48	999.00	763.00	216.03	0.002410
S47	1272.20	1115.28	1006.50	764.50	231.18	0.002520
S48	1272.20	1156.25	1075.40	768.50	326.51	0.002751
S49	1272.20	1158.93	1078.40	768.70	333.76	0.002805

Fig. VII-14. Condensation Coefficient vs. Saturation Pressure at Surface of Condensate Film

Larrain (12), who correlated most of the available heat transfer data on the condensation of liquid metal vapors to obtain his compilation. The data for sodium were quite consistent, the scatter in the values of σ_c around the line shown in Fig. VII-14 being within approximately $\pm 25\%$, whereas the scatter in the data for potassium were generally within $\pm 50\%$. The data for mercury showed scatter of several hundred percent for P_{sl} greater than approximately 10 Torr, with σ_c being consistently close to 1 below 10 Torr. The line shown in Fig. VII-14 was estimated by eliminating the most inconsistent data points for mercury, thereby reducing the scatter to within approximately a factor of 2. For P_{sl} below approximately 10 Torr, σ_c was close to 1 for sodium, potassium, and mercury.

The best straight line through the data points shown in Fig. VII-14, determined by least squares analysis, was represented by the following equation:

$$\log \sigma_c = -0.546 \log P_{sl} - 1.320, \quad (\text{VII-8})$$

where P_{sl} has the units of Torr. The probable error in the values of σ_c calculated by Eq. VII-8 was computed to be $+20\%$ or -17% ; the probable limits can be found by multiplying and by dividing the calculated values by 1.20.

It can be seen from Fig. VII-14 that σ_c for selenium decreases with increasing P_{sl} , as is the case for the liquid metals. σ_c for selenium is generally one to two orders of magnitude smaller than σ_c for sodium, potassium, and mercury at the same P_{sl} .

The analytical expressions which account for the nonlinear temperature profile in the condensate film (R4) and for convection and acceleration

in the liquid film and vapor drag at the condensate surface (C_4) were found to increase h_{Nu} by 2.4% (at 760 Torr) to 3.9% (at 10 Torr), as shown in Sect. VII-C-2. The subcooling of the condensate film is accounted for in Eq. III-13. The resulting decrease in the calculated T_{sg} , due to the above increase in h_{Nu} , ranged from approximately 1.9°F (at 760 Torr) to 5.0°F (at 10 Torr), causing an increase in $(T_v - T_{sg})$ of approximately 1.7% (at 760 Torr) to 2.5% (at 10 Torr). Due to the slight reduction in T_{sg} , σ_c calculated by Eq. VII-7 would be decreased by approximately 0.5% (at 10 Torr) to 1.4% (at 760 Torr). These corrections would therefore result in a slight reduction in the calculated σ_c shown in Fig. VII-14 and Table VII-11, not an increase. A possible explanation for the relatively low σ_c for selenium and for the decrease in σ_c with increasing P_{sg} is presented in the next section.

4. Possible Effect of Molecular Structure on Condensation Coefficients

Paul (P1) has made a compilation of condensation coefficients for a wide variety of substances, most of which were obtained by evaporating the substances into vacua and by comparing the measured rates of evaporation with the theoretical maximum rates of evaporation (calculated by Eq. III-17). Paul concluded that most substances have σ_c close to 1. However, several substances whose vapor molecules differ from the condensate molecules because of association or dissociation were found to have very small σ_c , sometimes less than 1×10^{-4} . It is therefore apparent that σ_c is a function of the liquid-vapor structure.

A possible mechanism for the condensation of selenium vapor is shown in Fig. VII-15. As described in Appendix F, selenium vapor

Fig. VII-15. Possible Mechanism for Condensation of Selenium Vapor

consists of all molecular species from Se_2 to Se_8 , while liquid selenium is composed of long chains and rings. Although Se_2 is a major component of the vapor phase at the conditions of the experiments carried out in this research program, there is no evidence for the existence of Se_2 molecules in the liquid phase. The condensate surface might therefore be impermeable to Se_2 molecules, whereas Se_6 and Se_8 molecules could condense at the surface. Since the net flow of vapor is towards the condensate surface, Se_2 molecules would be carried to the interface by convection, where they would accumulate and consequently present a diffusional resistance to those molecules that actually condense. In this case, the Se_2 molecules in the vapor would have an effect almost similar to that of an inert gas on a condensing vapor, resulting in a resistance to mass transfer in the vapor boundary layer adjacent to the condensate surface and a greatly reduced value of the calculated σ_c (S5, B1). The accumulation of Se_2 in the boundary layer would be limited by diffusion of Se_2 from the boundary layer back into the bulk vapor and by the

kinetics of the various reactions that would occur in the boundary layer and in the bulk vapor, with the mole fraction of Se_2 reacting to approach the equilibrium value. It should be noted that the mechanism described in Fig. VII-15 is highly simplified, since all molecular species other than Se_2 , Se_6 , and Se_8 have been neglected. Nevertheless, since Se_2 and Se_6 are major components of the vapor phase, and Se_8 is a major component of the liquid phase, the above mechanism was considered to be a reasonable representation of the condensation process.

σ_c has been plotted against the mole fraction of dimer in the vapor phase (y_2) in Fig. VII-16. While sodium, potassium, and mercury vapor at these pressures are composed primarily of monomer, with some dimer present (M2, M7, W3), the liquid phases are generally considered to be monatomic (S3, S4). Wilhelm (W3) considered the liquid structure and the mechanism of dimerization to be uncertain. If dimerization occurs only in the vapor, with no dimer present in the liquid, then dimer molecules could accumulate at the condensate surface during condensation and present a diffusional resistance to monomer molecules. Thus, while the molecular species present in the liquid and vapor phases of selenium and the liquid metals are different, the mechanisms of condensation may be analogous. It can be seen from Fig. VII-16 that σ_c for selenium and each of the liquid metals decreases with increasing mole fraction of dimer, which is consistent with the proposed mechanisms of condensation. The fact that σ_c for selenium forms a nearly continuous straight line with the data for sodium and potassium, when correlated in terms of y_2 , would seem to be significant. The mechanism proposed for the condensation of selenium vapor (see Fig. VII-15) seems therefore to be a reasonable

Fig. VII-16. Condensation Coefficient vs. Mole Fraction of Dimer in Vapor

explanation for the relatively low σ_c for selenium, as well as for the decrease in σ_c with increasing $P_{s\ell}$ observed for selenium (and the liquid metals).

For condensing systems with different molecular species in the liquid and vapor phases, the σ_c calculated by Eq. VII-7 does not represent the "fraction of molecules striking the condensate surface that actually condenses." Nevertheless, while the calculated σ_c for such systems does not have the physical significance generally attributed to condensation coefficients, it would be useful for correlating or predicting condensing heat transfer data. The σ_c values shown in Figs. VII-14 and VII-16 should therefore be regarded as "effective" or "apparent" condensation coefficients.

VIII. CONCLUSIONS

1. Corrosion tests showed that stainless steel 316 had the best corrosion resistance towards selenium of several materials tested, with sufficient corrosion resistance to be used as the material of construction for the apparatus. While a significant amount of corrosion of the apparatus was found to have occurred during the heat transfer experiments with selenium, the good reproducibility of the heat transfer data showed that corrosion did not have a significant effect on the heat transfer data.

2. Preliminary boiling and condensing experiments, carried out with water and Dowtherm A as test fluids, showed that the assembled apparatus was capable of producing accurate and reliable heat transfer data.

3. The boiling and condensing heat transfer data obtained with selenium were internally consistent and reproducible.

4. Nucleate pool boiling heat transfer data were obtained over pressure and heat velocity ranges of approximately 10 to 760 Torr and 20,000 to 100,000 Btu/sq ft-hr, respectively. The following correlating equation for the heat transfer data was obtained from a least squares fit of the data:

$$q_b = (0.1956P^{1/2} + 8.379 \times 10^{-3} P)(T_w - T_v)^{1.981}$$

where q_b , $(T_w - T_v)$, and P have units of Btu/sq ft-hr, °F, and Torr, respectively. The geometric limits of probable error are $\pm 1.112^x$.

5. Most of the temperature drop across the boiling pool (between the boiling surface and the vapor space) occurred within 0.04 in. of the boiling surface, the bulk liquid in the boiling pool being at a nearly constant temperature.

6. Temperature fluctuations in the copper cylinder at the closest position to the boiling surface (0.10 in.) were negligible for most boiling runs, never exceeding 1°F. During the only non-boiling run (S25), large temperature fluctuations were observed.

7. Measured pressures, P_m , of the cover gas over the vapor were generally in close agreement with the corresponding selenium saturated vapor pressures at the boiling pool temperatures (see Fig. VII-5). For several runs at the lowest P_m (10 Torr), a significant pressure drop occurred across the auxiliary condenser, thereby limiting the maximum q_b that could be obtained at the lower pressures investigated.

8. Comparisons between the boiling heat transfer data and data predicted by six widely tested semi-empirical correlations for nucleate pool boiling showed satisfactory agreement at atmospheric pressure. However, the agreement became progressively worse as the pressure decreased, and predicted heat velocities exceeded experimental values by one to two orders of magnitude at 10 Torr.

9. Discrepancies between experimental and predicted boiling heat transfer data at the lower pressures investigated have been explained in terms of the complex structure of liquid selenium, with nonvolatile long chains of Se atoms in the liquid having an adverse effect on the boiling heat transfer.

10. Condensing heat transfer data were obtained over pressure and heat velocity ranges of approximately 10 to 760 Torr and 18,000 to 52,000 Btu/sq ft-hr, respectively. The observed h_c was generally only 40 to 50% of the prediction by the Nusselt equation for filmwise condensation on a horizontal tube.

11. A thermal resistance at the liquid-vapor interface was the most reasonable explanation for the differences between the values of h_c and h_{Nu} that were obtained for the condensation of selenium.

12. The condensation coefficient for selenium, determined from the condensing heat transfer data and the kinetic theory of condensation, decreased with increasing pressure according to the equation:

$$\log \sigma_c = -0.546 \log P_{sl} - 1.320 ,$$

where P_{sl} is in Torr. The geometric limits of probable error are ± 1.20 .

13. The large thermal resistance present at the liquid-vapor interface during the condensation experiments with selenium has been explained in terms of the molecular compositions of liquid and vapor selenium, with dimer in the vapor having an effect almost similar to that of a noncondensable gas, causing reductions in h_c and the calculated σ_c .

14. Commercial utilization of selenium as a thermodynamic working fluid or as a heat transport fluid would seem to require the development of materials with much greater corrosion resistance towards liquid selenium, while still having sufficient strength and thermal conductivity. Nitrides or similar surfaces on refractory metals may well be appropriate, but have not been investigated in this research program.

Appendix A. Experimental Data

The experimental data recorded during the boiling experiments with selenium are presented in Tables A-1 and A-2. The pressure and temperature data for the boiling experiments are presented in Table A-1, where $T_1 - T_v$ through $T_6 - T_v$ represent the six measured temperature differences between the thermocouples in the copper cylinder and the vapor space of the test chamber.

The power measurements are presented in Table A-2. The products of the voltage (E) times the current (I), measured by the voltmeters and ammeters, were calculated to check the readings of the wattmeter. The averages of the two current and two voltage measurements were employed in the computations of EI. The agreement between the wattmeter readings and the EI products was generally within $\pm 1\%$, the difference never exceeding 2%. All meters were calibrated (see Appendix B); the measurements reported in Table A-2 are the calibration-corrected values. The wattmeter readings, corrected for power dissipation in the lead wires, were used to determine the power input to the heater.

The experimental data recorded during the condensation experiments with selenium are presented in Tables A-3 and A-4, where T_{in} , the three values of $(T_{out} - T_{in})$, V_m , T_{wtm} , and H represent the air coolant temperature at the inlet of the test condenser, the three measurements of the temperature difference between the outlet and the inlet of the condensing tube, the measured volumetric flow rate of the air coolant, the temperature of the wet test meter, and the height of the condensing ring in the auxiliary condenser, respectively; $T_v - T_7$ through $T_v - T_{14}$ represent the eight measured temperature differences between the thermocouples

Table A-1. Experimental Data For Boiling Selenium - Temperature and Pressure Measurements
(Location of Thermocouples Shown in Figs. V-1 and C-1)

Run	P (Torr)	T _v (°F)	(T ₁ - T _v) (°F)	(T ₂ - T _v) (°F)	(T ₃ - T _v) (°F)	(T ₄ - T _v) (°F)	(T ₅ - T _v) (°F)	(T ₆ - T _v) (°F)
S1	101.25	1024.25	208.89	224.32	238.86	206.78	220.95	237.33
S2	101.75	1025.15	234.00	251.64	269.19	229.50	247.72	267.48
S3	101.75	1026.50	228.82	252.45	274.68	227.02	249.48	273.38
S4	100.00	1025.60	250.11	276.08	300.73	247.86	272.52	299.25
S5	100.75	1026.50	264.82	293.76	321.30	261.22	289.60	319.95
S6	100.00	1022.90	212.62	232.56	250.99	209.39	228.60	248.85
S7	98.75	1021.64	190.26	206.32	221.31	187.56	203.31	219.36
S8	98.75	1022.45	190.35	205.65	220.72	187.42	202.28	218.70
S9	98.75	1022.00	165.69	177.75	189.90	163.35	174.82	187.65
S10	99.75	1024.70	234.72	257.13	278.82	232.20	254.02	277.63
S11	11.30	829.85	258.84	271.35	283.50	256.95	269.03	281.92
S12	14.80	851.45	273.96	280.18	302.94	271.80	285.30	301.95
S13	21.90	882.95	273.82	290.48	307.12	271.44	287.32	305.55
S14	10.80	826.25	250.92	261.00	270.68	248.62	258.12	269.28
S15	10.60	825.35	248.62	257.18	265.50	246.15	254.48	263.88
S16	10.70	825.80	252.68	261.18	269.46	250.38	257.94	267.75
S17	15.40	853.25	277.88	292.14	306.68	274.50	289.26	305.32
S18	11.70	832.55	267.08	278.35	290.52	264.24	276.30	288.90
S20	29.00	904.28	227.66	241.42	254.92	225.72	238.68	253.15
S23	29.00	903.92	179.28	190.35	200.70	177.75	187.74	198.68
S24	29.00	904.28	151.42	159.66	167.85	149.94	157.50	165.60
S25	28.50	818.15	219.60	225.90	234.45	219.60	225.90	234.45
S27	29.50	905.90	202.77	214.20	225.45	199.60	211.28	223.65
S28	29.50	905.90	211.50	224.10	235.62	208.08	220.50	234.00
S34	99.50	1024.25	208.80	225.00	240.30	207.18	222.75	238.95
S35	100.00	1024.25	211.50	231.30	249.84	209.70	229.95	248.04
S36	100.00	1023.35	179.10	195.12	211.05	177.30	193.95	209.70
S37	29.50	904.64	197.64	209.52	220.95	195.84	208.08	219.42
S38	29.50	907.25	230.31	247.14	260.80	228.15	245.25	261.00
S39	10.00	822.65	254.16	266.85	279.45	252.00	265.68	278.10
S40	294.50	1146.20	148.14	167.58	185.85	146.25	165.60	184.05
S41	298.40	1146.20	170.10	193.95	216.45	170.28	192.78	215.10
S42	297.40	1146.65	180.45	205.38	229.32	178.20	202.95	227.70
S43	299.80	1147.10	193.95	222.75	249.75	192.15	220.50	248.40
S44	298.80	1146.20	145.62	163.08	179.55	144.90	161.28	177.75
S45	99.50	1022.00	181.98	198.00	213.75	180.72	196.92	212.40
S46	763.00	1272.20	121.68	141.75	160.20	120.15	139.68	158.40
S47	764.50	1272.20	125.82	148.05	169.20	124.38	145.80	167.40
S48	768.50	1272.20	130.68	156.60	180.18	129.42	154.62	178.38
S49	768.70	1272.20	140.40	168.75	194.85	138.42	166.05	193.05
CAL1	15.25	852.12	160.65	164.61	167.76	160.29	163.80	167.04
CAL2	102.50	1026.50	134.46	139.50	143.82	132.53	137.09	142.38
CAL3	514.25	1215.95	98.06	106.13	114.10	96.79	103.77	112.68

Table A-2. Experimental Data for Boiling
Selenium - Power Measurements

Run	I ^(a) (amp)	I ^(b) (amp)	E ^(c) (v)	E ^(d) (v)	EI (v)	Wattmeter (v)
S1	11.07	11.05	90.0	-	995	994
S2	12.16	12.20	98.1	-	1194	1195
S3	13.28	13.30	107.0	-	1420	1417
S4	14.30	14.36	112.4	-	1611	1607
S5	-	14.89	119.0	-	1770	1757
S6	12.16	12.20	97.7	-	1190	1195
S7	11.07	11.05	90.0	-	995	994
S8	11.03	11.05	89.1	-	984	984
S9	10.08	9.95	79.7	-	797	787
S10	13.19	13.20	106.0	-	1397	1387
S11	10.01	10.04	79.7	79.5	799	796
S12	10.95	10.96	88.3	86.0	965	955
S13	11.63	11.68	94.2	92.1	1095	1085
S14	9.37	9.47	-	73.9	695	695
S15	8.75	8.79	-	69.7	610	602
S16	8.75	8.79	-	69.7	610	602
S17	10.88	10.86	87.4	85.7	949	945
S18	9.91	9.95	79.7	78.9	787	777
S20	10.56	10.59	86.4	84.5	904	894
S23	9.40	9.47	-	75.1	708	697
S24	8.35	8.40	-	66.7	555	548
S25	7.48	7.50	-	59.9	448	449
S27	9.79	9.76	-	77.9	760	755
S28	10.06	10.04	80.7	79.7	805	797
S34	11.02	11.00	89.1	88.1	985	994
S35	12.05	12.10	97.3	95.8	1180	1195
S36	11.02	10.96	90.0	88.1	990	994
S37	9.68	9.66	-	76.9	745	747
S38	10.93	10.86	89.1	87.8	971	974
S39	9.89	9.85	79.7	79.2	784	787
S40	11.88	11.89	96.9	95.2	1150	1165
S41	13.05	13.00	105.0	104.0	1370	1382
S42	13.62	13.51	109.8	109.0	1490	1497
S43	14.40	14.36	116.0	115.4	1665	1677
S44	11.22	11.15	90.8	90.0	1015	1029
S45	11.02	10.96	89.1	88.3	976	984
S46	12.08	12.10	98.1	96.3	1185	1195
S47	12.79	12.70	103.0	101.7	1318	1331
S48	13.52	13.51	109.8	108.7	1482	1492
S49	14.12	14.15	114.1	114.2	1611	1617
CAL1	6.30	6.25	-	-	-	315
CAL2	7.08	7.05	-	-	-	405
CAL3	8.40	-	-	-	-	577

(a) - Ammeter with range of 0.4 to 15 amperes.

(b) - Ammeter with range of 0.8 to 30 amperes.

(c) - Voltmeter with range of 80 to 300 volts.

(d) - Voltmeter with range of 30 to 125 volts.

Table A-3. Experimental Data for Condensing Selenium

Run	P (Torr)	T _{in} (°F)	(T _{out} - T _{in})			V _m (cfm)	T _{wtm} (°F)	Power (w)	H (in.)
			1 (°F)	2 (°F)	3 (°F)				
S1	101.25	85.19	103.95	105.71	105.98	14.50	80.0	994	10.5
S2	101.75	84.88	108.94	110.47	110.70	13.45	80.0	1195	16.0
S3	101.75	84.20	114.75	116.01	116.19	11.75	79.0	1417	24.5
S4	100.00	83.75	113.04	114.30	114.66	11.40	79.0	1607	31.0
S5	100.75	83.75	102.42	102.82	103.05	13.95	79.0	1757	35.0
S6	100.00	83.75	101.88	102.60	102.82	13.30	78.0	1195	18.0
S7	98.75	83.75	96.53	97.42	97.65	14.38	78.0	994	14.0
S8	98.75	83.75	123.66	126.22	125.78	8.82	78.0	984	14.5
S9	98.75	83.75	123.05	125.78	125.33	8.85	78.0	787	11.0
S10	99.75	84.20	106.88	108.36	108.58	11.55	78.0	1387	25.0
S11	11.30	83.30	80.82	82.58	82.44	11.68	79.5	796	20.0
S12	14.80	83.75	79.20	80.10	80.10	13.04	79.0	955	26.0
S13	21.90	83.30	79.20	80.10	80.10	14.00	79.0	1085	30.0
S14	10.80	83.30	91.51	93.60	93.15	9.00	78.5	695	16.5
S15	10.60	81.28	102.24	105.52	104.85	7.10	76.0	602	13.0
S16	10.70	80.81	102.53	105.75	105.30	7.10	76.0	602	23.5
S17	15.40	81.50	78.08	79.20	79.20	12.91	76.0	945	26.0
S18	11.70	81.50	81.00	82.58	82.58	11.27	76.0	777	18.5
S19	26.00	81.50	91.44	92.70	92.70	11.18	77.0	1165	34.0
S20	29.00	81.72	78.30	79.38	79.38	14.67	77.0	894	16.0
S21	29.00	81.72	95.94	97.65	97.65	10.57	77.0	1064	27.0
S22	29.00	81.05	94.95	96.75	96.75	10.53	76.0	994	22.0
S23	29.00	81.50	94.05	95.85	95.40	10.53	75.0	697	12.5
S24	29.00	81.50	93.60	95.85	95.40	10.52	74.0	548	9.0
S25	28.50	81.50	72.45	73.80	73.80	10.57	74.0	449	0.0
S26	29.00	80.60	94.05	95.85	95.85	10.50	73.0	896	19.5
S27	29.50	81.50	114.30	117.99	117.45	7.05	76.0	755	17.0
S28	29.50	81.50	76.28	77.40	77.40	14.60	76.0	797	14.0
S29	30.00	83.30	76.50	77.40	77.40	14.50	76.0	797	25.0
S30	100.25	82.40	94.86	95.40	95.76	14.03	78.0	994	9.0
S31	100.25	81.50	121.32	123.30	122.94	8.80	77.0	790	9.5
S32	30.00	81.50	115.56	118.08	117.72	7.10	77.0	750	10.5
S33	29.50	81.50	117.72	120.60	120.60	7.00	76.0	757	9.5
S34	99.50	80.15	105.75	106.65	106.65	13.00	74.0	994	14.0
S35	100.00	80.60	108.90	108.90	109.35	12.90	74.0	1195	19.0
S36	100.00	80.60	105.12	106.20	105.75	13.85	73.0	994	13.0
S37	29.50	80.60	121.05	123.30	123.30	7.10	74.0	747	18.0
S38	29.50	80.60	99.45	100.35	100.80	10.57	73.0	974	21.0
S39	10.00	81.05	85.05	85.05	85.95	11.20	72.0	787	21.0
S40	294.70	81.50	126.45	129.15	128.70	13.30	72.0	1165	14.0
S41	298.40	81.05	135.45	138.15	138.15	11.65	72.0	1382	19.0
S42	297.40	81.50	135.45	138.15	137.70	11.70	72.0	1497	23.0
S43	299.80	81.50	135.90	137.70	138.15	11.70	72.0	1677	27.0
S44	298.80	81.50	135.00	137.70	138.15	11.73	72.0	1029	12.0
S45	99.50	81.50	99.45	100.35	100.35	14.25	72.0	984	14.0
S46	763.00	81.50	141.48	144.00	144.00	13.90	72.0	1195	9.0
S47	764.50	81.50	143.28	146.25	145.80	13.85	72.0	1331	14.0
S48	768.50	81.50	163.80	168.75	167.40	10.28	72.0	1492	18.0
S49	768.70	81.50	164.70	168.30	168.30	10.25	72.0	1617	22.0

Table A-4. Experimental Data for Condensing Selenium

Run	T_v (°F)	$(T_v - T_7)$ (°F)	$(T_v - T_8)$ (°F)	$(T_v - T_9)$ (°F)	$(T_v - T_{10})$ (°F)	$(T_v - T_{11})$ (°F)	$(T_v - T_{12})$ (°F)	$(T_v - T_{13})$ (°F)	$(T_v - T_{14})$ (°F)
S1	1024.25	307.80	307.48	309.06	306.68	280.44	282.38	284.40	282.17
S2	1025.15	297.68	297.00	299.70	297.54	270.99	272.52	273.60	272.16
S3	1026.50	285.52	284.40	287.78	285.97	262.94	263.81	264.60	263.52
S4	1025.60	295.56	294.08	296.82	295.56	273.06	273.53	274.72	273.82
S5	1026.50	340.20	338.40	340.99	339.84	318.15	318.82	320.40	319.14
S6	1022.90	340.56	339.08	341.28	340.20	319.50	320.18	321.30	320.22
S7	1021.64	360.54	359.64	361.58	360.00	339.75	340.11	341.10	340.20
S8	1022.45	266.62	265.14	267.52	265.50	247.28	247.50	247.50	247.50
S9	1022.00	268.20	266.40	268.65	266.94	249.12	249.34	249.34	249.34
S10	1024.70	329.76	327.82	330.48	328.95	309.82	309.96	310.59	310.05
S11	829.85	284.85	283.86	284.49	283.05	271.98	271.58	272.02	271.13
S12	851.45	306.68	305.46	306.90	306.22	294.98	294.53	295.20	294.53
S13	882.95	329.85	328.50	329.85	328.71	316.58	316.80	317.70	316.35
S14	826.25	246.38	245.25	245.70	245.02	236.70	236.70	236.92	237.15
S15	825.35	206.78	205.20	205.87	205.65	198.22	198.00	197.32	198.00
S16	825.80	207.22	205.85	206.55	205.65	197.55	197.10	196.20	196.65
S17	853.25	316.35	314.55	315.90	315.22	306.81	306.45	306.45	306.22
S18	832.55	288.45	287.55	288.68	287.55	279.90	279.45	279.90	279.90
S19	896.36	288.00	286.65	287.55	287.10	278.55	277.65	278.10	277.65
S20	904.28	347.94	346.50	347.40	346.95	340.65	340.20	340.20	339.75
S21	910.40	285.75	283.72	285.08	284.40	278.10	277.42	276.84	276.62
S22	908.60	289.35	287.55	288.68	288.00	281.70	281.25	280.89	280.58
S23	903.92	287.78	286.42	287.10	286.20	280.58	280.35	279.90	279.45
S24	904.28	288.22	286.65	287.55	286.65	280.80	280.35	279.90	279.90
S25	818.15	332.10	332.10	333.90	331.20	323.10	323.10	323.55	323.10
S26	906.80	287.55	285.75	287.10	286.20	279.90	279.45	279.00	279.00
S27	905.90	222.30	221.40	221.85	220.95	213.30	212.85	212.40	212.40
S28	905.90	362.70	361.80	362.48	361.35	356.40	356.62	355.95	355.95
S29	906.35	364.05	362.70	363.60	362.70	357.30	356.85	356.40	356.40
S30	1024.25	368.10	367.20	367.65	365.85	356.22	354.60	355.50	356.22
S31	1024.25	273.24	271.80	272.70	270.90	258.70	257.40	255.60	257.40
S32	923.72	241.65	240.30	239.85	238.95	226.80	226.35	225.45	225.45
S33	927.97	230.85	228.60	229.95	229.50	217.98	216.36	216.36	216.90
S34	1024.25	299.70	299.70	301.05	300.15	278.55	279.45	279.90	279.45
S35	1024.25	300.60	300.60	302.58	301.50	278.10	279.90	279.90	279.00
S36	1023.35	317.25	318.17	319.95	319.05	296.55	298.35	298.35	297.45
S37	904.64	208.80	208.80	210.60	209.70	189.90	191.70	190.35	190.35
S38	907.25	268.65	268.65	270.45	270.45	251.55	253.35	252.90	252.00
S39	822.65	270.00	270.00	271.35	270.90	255.15	256.50	255.15	255.15
S40	1146.20	296.10	296.10	297.90	296.55	268.92	272.70	272.70	270.90
S41	1146.20	269.64	270.00	272.25	270.00	240.30	242.55	242.55	241.20
S42	1146.65	271.35	270.90	273.15	271.35	240.48	243.45	242.55	241.65
S43	1147.10	273.15	273.15	275.85	274.50	242.55	245.25	244.35	243.45
S44	1146.20	270.00	270.00	272.25	270.00	239.40	242.10	241.20	240.30
S45	1022.00	350.55	350.55	352.35	351.00	329.85	333.45	330.75	331.20
S46	1272.20	305.28	305.28	307.35	305.28	270.90	274.50	273.15	271.35
S47	1272.20	297.72	297.72	299.52	297.45	263.70	268.20	265.05	264.15
S48	1272.20	222.75	224.10	226.80	225.00	196.65	198.90	197.55	196.20
S49	1272.20	220.05	220.95	224.55	222.30	193.50	195.75	194.40	193.05

in the vapor space of the test chamber and the silver lining of the test condenser, thermocouples 7 through 10 being located closer to the inlet and thermocouples 11 through 14 being located closer to the outlet of the condensing tube (see Section V-B).

Appendix B. Calibrations

1. Electrical Meters

The electrical meters described in Section V-C-4 were calibrated in place against Weston A.C. standard meters which were accurate to within $\pm 1/4\%$ of full scale. The power, voltage, and current measurements reported in Appendix A are the corrected values.

A distortion meter was used to check the wave shape of the voltage generated by the power supply. The voltage was found to be sinusoidal, with distortion being negligible.

2. Heat Losses

To check the boiling heat velocity calculated from the temperatures measured in the copper cylinder (see Appendix C), an energy balance was made around the copper cylinder for each boiling run. Since the heat loss from the copper cylinder must be accounted for in the energy balance, heat loss calibrations were carried out. The results are presented in Fig. B-1, where Q_{loss} is the heat loss below the boiling plate and T_{below} is the average surface temperature of the unheated portions of the copper cylinder and of the alundum core (the average surface temperature below the boiling plate).

The heat loss calibrations were carried out by supplying power to the heater with the test chamber evacuated. The total heat loss from the apparatus at each steady state was equal to the power supplied to the heater.

The temperature profiles in the copper cylinder and the alundum core were determined from the measured temperatures in the copper

Fig. B-1. Results of Heat Loss Calibrations

cylinder. T_{below} was then calculated as the average surface temperature of the unheated portions of the copper cylinder and the alundum core.

The theoretical heat loss from each section of the apparatus was calculated. The theoretical heat loss below the boiling plate, Q_{below} , was calculated from the average surface temperatures of the unheated portions of the copper cylinder and the alundum core and from the calculated thermal resistances around each section:

$$Q_{\text{below}} = \left(\frac{T_{\text{un}} - T_{\text{Ar}}}{\frac{0.48}{k_{\text{fib}}} + \frac{2.78}{h_{\text{r+c}}}} \right) + \left(\frac{T_{\text{bot}} - T_{\text{Ar}}}{\frac{3.40}{k_{\text{fib}}} + \frac{20.4}{h_{\text{r+c}}}} \right) + \left(\frac{T_{\text{cr}} - T_{\text{Ar}}}{\frac{0.28}{k_{\text{fib}}} + \frac{1.69}{h_{\text{r+c}}}} \right) \quad (\text{B-1})$$

where T_{un} , T_{Ar} , T_{bot} , T_{cr} , k_{fib} , and $h_{\text{r+c}}$ represent the average surface temperature of the unheated section of the copper cylinder above the core, °F; the temperature of the argon in the closed chamber, °F; the temperature at the bottom of the copper cylinder, °F; the average surface temperature of the alundum core, °F; the thermal conductivity of the Fiberfrax insulation, Btu/ft-hr-°F; and the sum of the radiative and convective heat transfer coefficients for each section, Btu/sq ft-hr-°F, respectively.

In order to calculate the heat loss from the test chamber, it was assumed that the wall of the test chamber acted as an infinite cylindrical fin. The following equation was then derived for the heat loss from the test chamber wall, Q_{fin} :

$$Q_{\text{fin}} = \frac{0.0079 k_w m \left[\frac{T_w - T_{\text{Ar}}}{\tanh 0.78 m} - \frac{T_{\text{top}} - T_{\text{Ar}}}{\sinh 0.78 m} \right]}{\left[1 + \frac{0.069 m}{\tanh 0.78 m} \right]} \quad (\text{B-2})$$

where

$$m = \left[\frac{790}{k_w} \left(\frac{1}{\frac{0.69}{k_{fib}} + \frac{4.0}{h_{r+c}}} \right) \right]^{1/2} \quad (B-3)$$

and k_w and T_{top} represent the thermal conductivity of the stainless steel 316 test chamber wall, Btu/ft-hr-°F, and the temperature at the top of the test chamber, °F.

The heat transferred by radiation, Q_{rad} , from the boiling plate to the top of the test chamber was calculated. The equations and assumptions involved in the calculation are discussed by Madsen (M1). Touloukian's (T1) values for the emissivity of stainless steel 316 were used in the computations for the interchange factor. The theoretical heat loss above the boiling plate, Q_{above} , was then:

$$Q_{above} = Q_{fin} + Q_{rad} \quad (B-4)$$

The actual heat loss below the boiling plate, Q_{loss} , is:

$$Q_{loss} = \frac{Q_{in}}{\left[1 + \frac{Q_{above}}{Q_{below}} \right]} \quad (B-5)$$

where Q_{in} represents the power supplied to the heater, Btu/hr. This method of calculating Q_{loss} has the important advantage that the ratio of the theoretical heat losses can be predicted with much greater accuracy than the heat losses themselves, since uncertainties in the thickness and thermal conductivity of the Fiberfrax insulation and in the radiative and convective heat transfer coefficients from the surface of the insulation tend to cancel out when the ratio is taken.

For each boiling run, T_{below} was calculated from the measured temperatures in the copper cylinder, and Q_{loss} was obtained from Fig. B-1. The values of Q_{loss} have been estimated to be accurate within $\pm 10\%$. Since Q_{loss} generally represented only 10 to 15% of the power supplied to the heater during boiling experiments, the uncertainty in the energy balance around the copper cylinder that resulted from the uncertainty in Q_{loss} was generally within $\pm 2\%$.

3. Thermocouples

Before the heat transfer experiments with selenium were started, relative calibrations of the thermocouples in-place were carried out. Temperature differences between the fixed thermocouple in the vapor space of the test chamber and the eight thermocouples in the test condenser were measured under isothermal conditions. The temperature differences between the two moveable thermocouples in the boiling pool and the fixed thermocouple in the vapor space were also measured.

The depth of the selenium boiling pool during the calibrations was 0.5 in. With the bellows stretched to its maximum length, the junction of one moveable thermocouple was located very close to the surface of the boiling pool, while the junction of the other moveable thermocouple was actually in the vapor space, approximately 0.75 in. above the surface of the boiling pool. The power to the heater was adjusted so that the condensing ring was close to the bottom of the auxiliary condenser. Temperature gradients in the boiling pool were therefore minimized.

To ensure isothermal conditions around the test condenser, the calibrations were carried out without coolant-flow through the test condenser and with the wall of the test chamber very heavily insulated. Estimates

of heat losses from the test condenser showed that, under these conditions, the vapor space and the condensing tube must have been practically isothermal. The results of the relative calibrations are summarized in Table B-1,

Table B-1. Relative Calibrations of Thermocouples

Run	T_v (°F)	$(T_v - T_j)_{av}$ (°F)	$(T_l - T_v)$ (°F)
CAL1.	851.00	4.32	2.88
CAL2.	1025.60	4.86	2.88
CAL3.	1214.60	5.04	1.80

where T_j represents one of the eight measured temperatures in the test condenser, $(T_v - T_j)_{av}$ represents the average temperature difference between the fixed thermocouple in the vapor space and the eight thermocouples in the test condenser, and T_l represents the temperature of the moveable thermocouple whose junction was 0.75 in. above the liquid surface. For each of the calibration runs, the eight measured values of $(T_v - T_j)$ agreed very closely, the standard deviation of the eight values being less than 0.4°F for each run.

Measured temperature profiles in the boiling pool during subsequent heat transfer experiments indicated that the calibrations for the two moveable thermocouples were practically identical. The values of $(T_l - T_v)$ in Table B-1 were therefore also used to correct measured temperature differences between the fixed thermocouple and the moveable thermocouple that was closer to the boiling plate (see Fig. V-1).

Since the 14 thermocouples in the test condenser and the copper cylinder were made from the same spool of thermocouple wire, the calibrations of the 14 thermocouples were practically equal. The standard deviation of the eight values of $(T_v - T_j)$ for each calibration run was less than 0.4°F . The calibrations of the 14 thermocouples in boiling water at 212°F were found to be within 0.4°F of each other, the standard deviation of the 14 values being less than 0.2°F . The values of $(T_v - T_j)_{av}$ shown in Table B-1 were therefore used as corrections for the measured temperature differences between the fixed thermocouple in the vapor space and each of the six thermocouples in the copper cylinder.

All thermocouples were also calibrated at 212°F against a standard thermometer in boiling water. In all cases, thermocouple corrections fell within the special limits of error ($\pm 2^\circ\text{F}$ at 212°F) for chromel-alumel thermocouples. The value of $(T_v - T_j)_{av}$ at 212°F was found to be 2.16°F , while the value of $(T_g - T_v)$ was found to be 0.18°F .

Appendix C. Calculations

1. Boiling Data

The boiling heat velocities and temperature differences reported in Section VII were obtained from the measured temperature profiles in the copper cylinder. The locations of the thermocouple junctions in the copper cylinder are shown in Fig. C-1. The thermocouples were located on radii of the copper cylinder, junctions 1, 2, and 3 being 0.25 in. from the center line and junctions 4, 5, and 6 being 0.75 in. from the center line. Junctions 1 and 4, 2 and 5, and 3 and 6 were located 0.10 in., 0.85 in., and 1.60 in. respectively from the top of the copper cylinder.

Due to conduction of heat along the wall of the test chamber above the boiling plate, the uniformity of heat velocity in the copper cylinder was uncertain. Since the uniformity of the heat velocity was greatest near the center line of the copper cylinder, temperatures T_1 , T_2 , and T_3 were employed in the computations for the boiling heat velocities and temperature differences reported in Section VII. By assuming a parabolic temperature profile in the axial direction, the following equations were derived for calculating the boiling heat velocity (q_b) and the temperature at the top of the copper cylinder (T_t) at the 0.25 in. radial position:

$$T - T_v = \alpha + \beta y + \gamma y^2, \quad (C-1)$$

$$\beta = \frac{1.8375(T_2 - T_1) - 0.7125(T_3 - T_2)}{0.75(1.1250)}, \quad (C-2)$$

$$\alpha = \frac{0.7225(T_1 - T_v) - 0.0100(T_2 - T_v) - \beta(0.06375)}{0.7125}, \quad (C-3)$$

Fig. C-1. Locations of Thermocouple Junctions in Copper Cylinder

$$q_b = \left(k_{Cu} \frac{\partial T}{\partial y} \right) \Big|_{y=0} = k_{Cu} \beta, \quad (C-4)$$

$$(T_t - T_v) = (T - T_v) \Big|_{y=0} = \alpha, \quad (C-5)$$

where k_{Cu} is the thermal conductivity of copper and y is the distance from the top of the copper cylinder. The heat velocity at the boiling surface was assumed to be equal to the heat velocity at the top of the copper cylinder. It should be noted that the temperature differences between the thermocouples in the copper cylinder and the vapor space were measured during the heat transfer experiments, whereas the actual temperatures in the copper cylinder were not measured directly.

Thermocouples 4, 5, and 6 were used to check the reliability of thermocouples 1, 2, and 3. As expected, T_4 , T_5 , and T_6 were consistently lower than the corresponding temperatures T_1 , T_2 , and T_3 , usually by 1 to 3°F (in the case of the boiling selenium experiments). The boiling heat velocities computed by Eq. C-4 were an average of 7% higher than heat velocities determined from thermocouples 4, 5, and 6, using an equation analogous to Eq. C-4. The standard deviation between the values of q_b and the heat velocities determined from thermocouples 4, 5, and 6 was 11%.

The temperature drop across the stainless steel 316 boiling plate (ΔT_{ss}) was calculated by the following equation:

$$\Delta T_{ss} = \frac{q_b X_{ss}}{k_{ss}}, \quad (C-6)$$

where X_{ss} and k_{ss} represent the thickness and thermal conductivity of

the stainless steel 316 boiling plate, respectively. The thermal conductivity data of Watson (W1) for k_{ss} are represented to within $\pm 0.25\%$ by:

$$k_{ss} = 0.00503(T - 572) + 10.5 . \quad (C-7)$$

Watson's data have been estimated to have an accuracy of $\pm 2\%$.

The temperature drop across the brazed joint (ΔT_{br}) was calculated:

$$\Delta T_{br} = \frac{q_b X_{br}}{k_{br}} , \quad (C-8)$$

where X_{br} is the thickness of the brazed joint and k_{br} is the thermal conductivity of the brazing alloy. The thickness of the brazed joint was estimated by measuring the distance between the top of the boiling plate and the bottom of the copper cylinder with a 7 to 8 in. micrometer, both before and after the brazing was carried out. The thermal conductivity of the brazing alloy, estimated by Ewing (E5), is:

$$k_{br} = 16.0 + 0.0149(T - 68) . \quad (C-9)$$

The calculated value of ΔT_{br} never exceeded 1% of the boiling temperature difference. The uncertainty in the boiling temperature difference resulting from the uncertainty in k_{br} was therefore negligible.

The boiling temperature difference ($T_w - T_v$) and the boiling heat transfer coefficient (h_b) were then calculated:

$$(T_w - T_v) = (T_t - T_v) - \Delta T_{ss} - \Delta T_{br} + \Delta T_{cal} , \quad (C-10)$$

$$h_b = q_b / (T_w - T_v) , \quad (C-11)$$

where ΔT_{cal} represents the calibration for the temperature differences between the thermocouples in the copper cylinder and the vapor space (see Appendix B).

It should be noted that the boiling heat velocities and temperature differences calculated by Eqs. C-4 and C-10 are actually local values for the 0.25 in. radial position.

The average boiling heat velocity ($\bar{q}_b$) was determined from an energy balance around the copper cylinder:

$$\bar{q}_b = (Q_{in} - Q_{loss} - Q_{wall})/A_b, \quad (C-12)$$

where Q_{in} , Q_{loss} , Q_{wall} , and A_b represent the power supplied to the heater, the heat loss below the boiling plate, the heat conducted along the wall of the test chamber above the boiling plate, and the area of the boiling plate, respectively.

Q_{in} was determined from the following equation:

$$Q_{in} = (\text{power})(3.415)(0.984), \quad (C-13)$$

where 0.984 corrects for power dissipation in the lead wires.

Q_{loss} was determined from the heat loss calibrations, described in Appendix B.

Q_{wall} was estimated by assuming that the wall of the test chamber acted as an infinite cylindrical fin (B12). The heat conduction equation for the wall could then be written:

$$\frac{d^2(T_f - T_v)}{dx^2} = \frac{\pi D_f h_f}{k_f S_f} (T_f - T_v), \quad (C-14)$$

where X , T_f , D_i , h_f , k_f , and S represent the distance from the base of the wall, the temperature of the wall (or "fin"), the inside diameter of the wall, the heat transfer coefficient from the wall to the liquid, the thermal conductivity of the wall, and the cross sectional area of the wall, respectively.

The boiling heat velocity was expressed as a function of pressure and boiling temperature difference:

$$q_b = a(T_w - T_v)^b (P)^c, \quad (C-15)$$

where the constants a , b , and c were determined by a least squares analysis of the boiling heat transfer data (see Section VII-B-2). By replacing h_f in Eq. C-14 with the h_b derived from Eq. C-15 and by utilizing the boundary conditions that at $X = \infty$, $(T_f - T_v) = 0$, and $dT_f/dX = 0$, the following equation for Q_{wall} was derived:

$$Q_{wall} = \left[\frac{2\pi D_i a k_f S P^c}{b+1} \right]^{\frac{1}{b+1}} (T_w - T_v)^{\frac{b+1}{2}}. \quad (C-16)$$

By substituting the calculated values of Q_{in} , Q_{loss} , and Q_{wall} into Eq. C-12, the average boiling heat velocity was determined.

The values of $\bar{q}_b$ and q_b were generally in close agreement, the values of q_b being an average of 4% higher than the values of $\bar{q}_b$; the standard deviation between the values of q_b and $\bar{q}_b$ was 6%. The close agreement between the values of q_b , $\bar{q}_b$, and the heat velocities determined from thermocouples 4, 5, and 6 confirmed the reliability of the boiling heat velocities calculated by Eq. C-4 and reported in Section VII.

The calculations were performed on an IBM 360 computer.

a. Sample Calculations - run S43

The following measurements were recorded during run S43:

$$\text{power} = 1677 \text{ watts},$$

$$P_m = 299.8 \text{ Torr},$$

$$T_v = 1147.10^\circ\text{F},$$

$$T_1 - T_v = 193.95^\circ\text{F},$$

$$T_2 - T_v = 222.75^\circ\text{F},$$

$$T_3 - T_v = 249.75^\circ\text{F},$$

$$T_4 - T_v = 192.15^\circ\text{F},$$

$$T_5 - T_v = 220.50^\circ\text{F},$$

$$T_6 - T_v = 248.40^\circ\text{F}.$$

The values of q_b and $(T_t - T_v)$ were calculated:

$$\beta = \frac{1.8375(222.75 - 193.95) - 0.7125(249.75 - 222.75)}{0.75(1.1250)}$$

$$= 39.92^\circ\text{F/in.} = 479.04^\circ\text{F/ft};$$

$$(T_t - T_v) = \frac{0.7225(193.95) - 0.0100(222.75) - 39.92(0.06375)}{0.7125}$$

$$= 189.98^\circ\text{F};$$

$$T_t = 189.98 + 1147.10 = 1337.08^\circ\text{F};$$

$$\text{at } 1337.08^\circ\text{F}, k_{\text{Cu}} = 205.7 \text{ Btu/ft-hr-}^\circ\text{F};$$

$$q_b = (205.7)(479.04) = 98,556 \text{ Btu/sq ft-hr}.$$

k_{ss} and ΔT_{ss} were determined by an iterative procedure. The final values were calculated:

$$T_{ss} = 1337.08 + \frac{58.64}{2} = 1366.40^\circ\text{F},$$

where T_{ss} represents the average temperature of the boiling plate;

$$k_{ss} = 0.00503(1366.40 - 572) + 10.5 = 14.49 \text{ Btu/ft-hr-}^{\circ}\text{F} ;$$

$$X_{ss} = 0.1035 \text{ in.} ;$$

$$\Delta T_{ss} = \frac{98,556(0.1035)}{14.49(12)} = 58.64^{\circ}\text{F} .$$

ΔT_{br} was calculated:

$$X_{br} = 0.004 \text{ in.} ;$$

$$k_{br} = 16.0 + 0.0149(1337.08 - 68) = 34.91 \text{ Btu/ft-hr-}^{\circ}\text{F} ;$$

$$\Delta T_{br} = \frac{98,556(0.004)}{34.91(12)} = 0.94^{\circ}\text{F} .$$

The values of $(T_w - T_v)$ and h_b were then calculated:

$$\Delta T_{cal} = 4.89^{\circ}\text{F} ;$$

$$(T_w - T_v) = 189.98 - 58.64 - 0.94 + 4.89 = 135.29^{\circ}\text{F} ;$$

$$h_b = 98,556/135.29 = 728.53 \text{ Btu/sq ft-hr-}^{\circ}\text{F} .$$

The values of Q_{in} , Q_{loss} , and Q_{wall} were calculated:

$$Q_{in} = (1677)(3.415)(0.984) = 5635.32 \text{ Btu/hr} ;$$

by the procedure described in Appendix B, T_{below} was computed to be 1647.9°F ; then,

$$Q_{loss} = 669.50 \text{ Btu/hr (see Fig. B-1)} ;$$

$$D_i = 2.87 \text{ in.} = 0.239 \text{ ft} ;$$

$$S = \frac{\pi(3.00^2 - 2.87^2)}{(4)(144)} = 0.00413 \text{ sq ft} ;$$

from the least squares fit of the boiling heat transfer data (see Section VII-B-2),

$$a = 0.1521 ;$$

$$b = 1.981 ;$$

$$c = 0.6422 ;$$

$$T_w = 1147.10 + 135.29 = 1282.39^\circ\text{F} ;$$

$$k_f = 0.00503(1282.39 - 572) + 10.5 = 14.05 \text{ Btu/ft-hr-}^\circ\text{F} ;$$

$$Q_{\text{wall}} = \left[\frac{(2\pi)(0.239)(0.1521)(14.05)(0.00413)(299.8)^{0.6422}}{1.981 + 1} \right]^{1/2} \\ \times (135.28)^{\frac{2.981}{2}} = 661.87 \text{ Btu/hr} .$$

$\bar{q}_b$ was then calculated:

$$A_b = \frac{\pi(2.87)^2}{(4)(144)} = 0.04492 \text{ sq ft} ;$$

$$\bar{q}_b = (5635.32 - 669.50 - 661.87)/0.04492 \\ = 95,803 \text{ Btu/sq ft-hr} .$$

2. Condensation Data

The condensing heat flux was determined by measuring the sensible heat gained by the air coolant as it passed through the test condenser. The heat conducted through the test chamber wall to the air coolant was accounted for analytically. Heat losses in the air coolant line and in the expansion chamber can be shown to be negligible in all cases. The condensing temperature difference was obtained by extrapolating the eight measured temperatures in the silver layer of the condensing tube to the

surface of the condensing tube. The test condenser and the condensing section of the apparatus are shown in detail in Fig. V-2. A simplified schematic diagram of the test condenser is presented in Fig. C-2.

The sensible heat gained by the air coolant (Q_{air}) was computed by the following equation:

$$Q_{air} = M_{air} c_{p,air} (T_{out} - T_{in}) , \quad (C-17)$$

where M_{air} , $c_{p,air}$, T_{out} , and T_{in} represent the mass flow rate, heat capacity, outlet temperature, and inlet temperature of the air coolant, respectively. At the temperature and pressure conditions of the air coolant, the enthalpy change due to the compressibility of air ($[\partial H / \partial P]_T \Delta P$) was computed to be negligible in all cases.

Since the air coolant became saturated with water vapor in the wet test meter, a humidity correction was required to account for the volume of water vapor added to the air coolant in the wet test meter. The corrected M_{air} and $c_{p,air}$ were calculated by the following equations:

$$V_{dry} = \left[\frac{V_m}{1 + 1.62 H_g} \right] , \quad (C-18)$$

$$M_{dry} = V_{dry} \rho_{dry} , \quad (C-19)$$

$$M_{H_2O} = M_{dry} H_{in} , \quad (C-20)$$

$$M_{air} = M_{dry} + M_{H_2O} , \quad (C-21)$$

$$c_{p,air} = (c_{p,dry} + c_{p,H_2O} \times H_{in}) \frac{M_{dry}}{M_{air}} , \quad (C-22)$$

where V_m is the volumetric flow rate measured by the wet test meter; H_g is

Fig. C-2. Schematic Diagram of Test Condenser

the humidity of the saturated air in the wet test meter; V_{dry} , M_{dry} , and ρ_{dry} are the volumetric flow rate, mass flow rate, and density of dry air; $M_{\text{H}_2\text{O}}$ is the mass flow rate of water vapor through the test condenser; H_{in} is the humidity of the air in the test condenser; and $c_{p,\text{dry}}$ and $c_{p,\text{H}_2\text{O}}$ are the heat capacities of dry air and water vapor, respectively. H_{in} was evaluated by assuming that the air was saturated at the conditions upstream from the pressure regulator, the air being at approximately 80 psig and 85°F. Since the mass of water vapor in the air entering the test condenser was generally less than 0.5% of the mass of dry air, uncertainties in the conditions upstream from the pressure regulator had negligible effect on the calculated values of M_{air} and $c_{p,\text{air}}$.

To calculate the heat transferred to the air-coolant through the wall of the test chamber, it was assumed that the section of wall around both ends of the test condenser acted as infinite circular fins, perpendicular to the axis of the condensing tube. The following equation was then derived to calculate the ratio of heat transferred through the "fin" at the inlet of the test condenser ($Q_{\text{end } 1}$) to the condensing heat flux (Q_c);

$$\frac{Q_{\text{end } 1}}{Q_c} = \frac{(k_f)(L_f)(T_v - T_{bl})(\mu)[K_1(\mu r_o)]}{(h_c)(L_c)(T_v - T_o)[(K_0' + (\mu r_o'))]} \quad (C-23)$$

where k_f , L_f , L_c , T_{bl} , and r_o represent the thermal conductivity of the wall, the wall thickness, the average length of the test condenser, the temperature at the base of the "fin", and the outside radius of the condensing tube, respectively; K_1 and K_0 are Bessel functions of orders 1 and 0, respectively; and

$$\mu = \left[\frac{h_f}{k_f L_f} \right]^{\frac{1}{2}} \quad (C-24)$$

The value of h_f , the heat transfer coefficient between the "fin" and the vapor, was assumed to be equal to h_c . An equation similar to Eq. C-23 was written for the "fin" at the outlet of the test condenser to calculate $Q_{\text{end } 2}/Q_c$.

Equation C-23 contains two unknowns, $Q_{\text{end } 1}$ and T_{b1} . An additional relationship between $Q_{\text{end } 1}$ and T_{b1} was derived by considering the heat transferred between the condensing tube and the air coolant:

$$\frac{Q_{\text{end } 1}}{Q_c} = \frac{(L_f)(T_{b1} - T_{\text{in}})}{(L_c)(T_o - T_{\text{av}})} \quad (C-25)$$

where T_{av} is the average air temperature in the test condenser. In deriving Eq. C-25, it is assumed that axial conduction in the test condenser is negligible and that the heat transfer coefficients of the air coolant at the ends and along the length of the condensing tube are equal. An equation similar to Eq. C-25 was written for the "fin" at the outlet of the test condenser.

The condensing heat flux was calculated by the following equation:

$$Q_c = \frac{Q_{\text{air}}}{\left[\frac{Q_{\text{end } 1}}{Q_c} + \frac{Q_{\text{end } 2}}{Q_c} + 1 \right]} \quad (C-26)$$

This method of calculating Q_c has the advantage that the ratios $Q_{\text{end } 1}/Q_c$ and $Q_{\text{end } 2}/Q_c$ can be calculated more accurately than the actual values of $Q_{\text{end } 1}$ and $Q_{\text{end } 2}$, due primarily to the uncertainty in the heat transfer coefficient of the air coolant. The condensing heat velocity was then

calculated:

$$q_c = Q_c / A_c , \quad (C-27)$$

where A_c , the surface area of the condensing tube, was computed to be 0.03742 sq ft. It should be noted that q_c is the average condensing heat velocity over the surface of the test condenser.

The average condensing temperature difference ($T_v - T_o$) was calculated by the following equations:

$$T_v - T_o = (T_v - T_j)_{av} - \Delta T_{ss} - \Delta T_{Ag} - \Delta T_{cal}, \quad (C-28)$$

$$\Delta T_{ss} = \frac{q_c (\ln \frac{0.628}{0.574})(r_o)}{k_{ss}}, \quad (C-29)$$

$$\Delta T_{Ag} = \frac{q_c (\ln \frac{0.574}{0.462})(r_o)}{k_{Ag}}, \quad (C-30)$$

where $(T_v - T_j)_{av}$ is the average temperature difference between the vapor and the eight thermocouples in the silver layer of the test condenser; ΔT_{ss} and ΔT_{Ag} are the temperature drops across the thin-walled stainless steel 316 tube and the silver layer, respectively; k_{ss} and k_{Ag} are the thermal conductivities of stainless steel 316 and silver, respectively; and ΔT_{cal} is the calibration for the average temperature difference between the vapor and the eight thermocouples in the test condenser (see Appendix B). The condensing heat transfer coefficient (h_c) was then calculated:

$$h_c = q_c / (T_v - T_o) . \quad (C-31)$$

Equations C-23 through C-31 were solved by an iterative technique, the calculations being performed on an IBM 360 computer.

a. Sample Calculations - run S43

The following measurements were recorded during run S43:

$$P_m = 299.8 \text{ Torr},$$

$$T_v = 1147.10^\circ\text{F},$$

$$(T_{\text{out}} - T_{\text{in}})_1 = 135.90^\circ\text{F},$$

$$(T_{\text{out}} - T_{\text{in}})_2 = 137.70^\circ\text{F},$$

$$(T_{\text{out}} - T_{\text{in}})_3 = 138.15^\circ\text{F},$$

$$T_v - T_7 = 273.15^\circ\text{F},$$

$$T_v - T_8 = 273.15^\circ\text{F},$$

$$T_v - T_9 = 275.85^\circ\text{F},$$

$$T_v - T_{10} = 274.50^\circ\text{F},$$

$$T_v - T_{11} = 242.55^\circ\text{F},$$

$$T_v - T_{12} = 245.25^\circ\text{F},$$

$$T_v - T_{13} = 244.35^\circ\text{F},$$

$$T_v - T_{14} = 243.45^\circ\text{F},$$

$$T_{\text{in}} = 81.50^\circ\text{F},$$

$$V_m = 11.70 \text{ cfm},$$

$$T_{\text{wtm}} = 72.0^\circ\text{F}.$$

The average values of $(T_{\text{out}} - T_{\text{in}})$ and $(T_v - T_j)$ were calculated:

$$(T_{\text{out}} - T_{\text{in}}) = \frac{135.90 + 137.70 + 138.15}{3} = 137.25^\circ\text{F};$$

$$(T_v - T_j)_{\text{av}} = \frac{1}{8}(273.15 + 273.15 + 275.85 + 274.50 + 242.55 + 245.25 + 244.35 + 243.45) = 259.03^\circ\text{F}.$$

The values of M_{air} and $c_{p,\text{air}}$ were calculated:

$$\text{at } 72^{\circ}\text{F}, H_g = 0.0171 \text{ lb } H_2O/\text{lb dry air} ;$$

$$V_{\text{dry}} = \frac{11.70}{1 + 1.62(0.0171)} = 11.39 \text{ cfm} ;$$

$$\text{at } 72^{\circ}\text{F}, \rho_{\text{dry}} = 0.0752 \text{ lb/cu ft} ;$$

$$m_{\text{dry}} = 11.39(0.0752)(60) = 51.37 \text{ lb/hr} ;$$

at a temperature of 81.5°F and a pressure of 80 psig,

$$H_{\text{in}} = 0.00405 \text{ lb } H_2O/\text{lb dry air} ;$$

$$M_{H_2O} = 51.37(0.00405) = 0.21 \text{ lb/hr} ;$$

$$M_{\text{air}} = 51.37 + 0.21 = 51.58 \text{ lb/hr} ;$$

$$T_{\text{av}} = 81.50 + \frac{137.25}{2} = 150.13^{\circ}\text{F} ;$$

$$\text{at } 150.13^{\circ}\text{F}, c_{p,\text{dry}} = 0.241 \text{ Btu/lb-}^{\circ}\text{F}$$

$$\text{and } c_{p,H_2O} = 0.455 \text{ Btu/lb-}^{\circ}\text{F} ;$$

$$c_{p,\text{air}} = [0.241 + 0.455(0.00405)] \frac{51.37}{51.58} = 0.241 \text{ Btu/lb-}^{\circ}\text{F} .$$

Q_{air} was then calculated:

$$Q_{\text{air}} = 51.58(0.241)(137.25) = 1709.6 \text{ Btu/hr} .$$

The values of T_o , T_{bl} , Q_{end} and $1/Q_o$, q_o , and h_o were determined by an iterative procedure. It was found that:

$$T_o = 902.0^{\circ}\text{F} ,$$

$$T_{bl} = 1066.6^{\circ}\text{F} ,$$

$$h_o = 167.6 \text{ Btu/sq ft-hr-}^{\circ}\text{F} .$$

q_c was calculated:

$$k_f = 0.00503(1066.6 - 572) + 10.5 = 12.99 \text{ Btu/ft-hr-}^\circ\text{F} ;$$

$$L_f = 0.125 \text{ in.} ;$$

$$\mu = \left[\frac{167.6}{12.99(0.125/12)} \right]^{1/2} = 35.19 \text{ ft}^{-1} ;$$

$$r_o = 0.314 \text{ in.} ;$$

$$\mu r_o = \frac{(35.19)(0.314)}{12} = 0.921 ;$$

$$K_o(0.921) = 0.473 ;$$

$$K_l(0.921) = 0.692 ;$$

$$L_c = 2.732 \text{ in.} ;$$

$$\begin{aligned} \frac{Q_{\text{end } 1}}{Q_c} &= \frac{(12.99)(0.125)(1147.1 - 1066.6)(35.19)(0.692)}{(167.6)(2.732)(1147.1 - 902.0)(0.473)} \\ &= 0.0599 ; \end{aligned}$$

$$\frac{Q_{\text{end } 1}}{Q_c} = \frac{0.125(1066.6 - 81.50)}{2.732(902.0 - 150.1)} = 0.0599 ;$$

in like manner, $Q_{\text{end } 2}/Q_c$ was calculated to be 0.0522;

$$Q_c = \frac{1709.6}{(0.0599 + 0.0522 + 1)} = 1537.1 \text{ Btu/hr} ;$$

$$A_c = 0.03742 \text{ sq ft} ;$$

$$q_c = 1537.1/0.03742 = 41,078 \text{ Btu/sq ft-hr} .$$

$T_v - T_o$ and h_c were then calculated:

$$\Delta T_{ss} = \frac{(41,078)(\ln \frac{0.628}{0.574})(0.314)}{(12.14)(12)} = 7.97^\circ\text{F} ;$$

$$\Delta T_{Ag} = \frac{(41,078) \left(\ln \frac{0.574}{0.462} \right) (0.314)}{(227.6)(12)} = 1.03^{\circ}\text{F} ;$$

$$(T_v - T_j)_{av} = 259.03^{\circ}\text{F} ;$$

$$\Delta T_{cal} = 4.89^{\circ}\text{F} ;$$

$$T_v - T_o = 259.03 - 7.97 - 1.03 - 4.89 = 245.14^{\circ}\text{F} ;$$

$$h_c = 41,078/245.14 = 167.6 \text{ Btu/sq ft-hr-}^{\circ}\text{F} .$$

Appendix D. Electrical Analogue Study

Due to the high thermal conductivity of copper, it was expected that a relatively short length of copper above the heated surface (see Fig. V-1) would provide a uniform heat velocity to the boiling liquid. The temperature profiles that would occur in the boiling section were analyzed by employing a network of electrical resistors which represented the analogous thermal resistances in the copper cylinder, the stainless steel 316 boiling plate, and the boiling liquid film. The network of electrical resistors and the procedure employed by Hochman (H3) were used, with appropriate modifications, in the analogue study.

The electrical network that was used is shown in Fig. D-1. The 3 kilohm resistors represent vertical resistances to heat transfer and the other labeled resistors represent radial thermal resistances in the appropriate sections of the copper rod. To simulate the expected uniform generation of heat, the r_g resistors were chosen large enough so that the current input was essentially uniform. The r_b resistors represent the thermal resistances of the boiling selenium and of the stainless steel boiling plate.

By utilizing the similarity in the equations of thermal and electrical conduction, measured currents and voltage drops across the resistors in the electrical network were related to the analogous heat fluxes and temperature drops in the boiling section. The fundamental equation for the analogue study was derived by combining Ohm's Law for electrical conduction and Fourier's Law for thermal conduction:

$$\left(\frac{Q}{i}\right) = \left[\frac{R_e}{R_t}\right] \left[\frac{\Delta T}{\Delta E}\right], \quad (D-1)$$

Fig. D-1. Electrical Analogue Network (Resistances in Kilohms)

where Q , ΔT , and R_t represent heat fluxes, temperature drops, and thermal resistances in the boiling section, and i , ΔE , and R_e represent measured currents, voltage drops, and electrical resistances in the electrical network. Each ratio within a parenthesis refers to an analogous electrical and thermal situation. R_e was determined from the values of the resistors in the network. R_t was determined from the thermal conductivity and the dimensions of the copper cylinder. The ratio (R_e/R_t) was computed to be 220,000 ohm-Btu/hr-°F. By assuming a convenient value for $(\Delta T/\Delta E)$, the values of Q and ΔT corresponding to the measured values of i and ΔE were calculated.

In conducting the experiments with the electrical network, a wide range of boiling heat transfer coefficients were considered by varying the values of the r_b resistors. Possible nonuniformities in the heat velocity and temperature profiles in the copper cylinder due to conduction of heat along the wall of the test chamber were neglected. The results of the analogue study showed that the heat velocity would then have a variation of less than 2% across the radius of the boiling plate, as long as the unheated length of copper exceeded 1 in. For the dimensions shown in Fig. V-1, the heat velocity to the boiling liquid would have a variation of less than 1% across the radius.

Since the nonuniformities in the heat velocity due to wall conduction could be significant, the actual uniformity of the heat velocity during boiling experiments was uncertain. Nevertheless, the analogue study did show that the heater described in Sect. V would provide a highly uniform heat velocity near the center line of the copper cylinder, where the thermocouples used to calculate the boiling heat transfer data were located.

Appendix E. Physical Properties of Selenium

To correlate the heat transfer data obtained in this research program, several physical properties of selenium had to be either known or estimated. The physical property data relevant to this research are presented:

1. Vapor Pressure

Vapor pressure data have been reported by several investigators. Most of the reported data were quite consistent and yielded straight lines when $\log P$ was plotted against $1/T$, as shown in Fig. E-1. The straight line which the author has drawn through the experimental points can be represented by the following equation:

$$\log P = 8.27808 - \frac{5174.761}{T}, \quad (E-1)$$

where P and T have the units of Torr and $^{\circ}\text{K}$, respectively.

2. Liquid Heat Capacity

The heat capacity of liquid selenium has been determined by Monval (M10) to be $0.118 \text{ cal/g-}^{\circ}\text{K}$ from 217 to 300°C and by Rau (R1) to be $0.1058 \text{ cal/g-}^{\circ}\text{K}$ from 230 to 750°C . K.K. Kelley (K2) recommends that a value of $0.1065 \text{ cal/g-}^{\circ}\text{K}$ be employed as the heat capacity for the entire range of temperatures that correspond to the liquid phase; this was the value used by the author.

3. Liquid Density

The density of liquid selenium has been measured and reported by several investigators. A plot of liquid density vs. temperature is presented in Fig. E-2. It can be seen that the data are very consistent.

Fig. E-1. Vapor Pressure of Selenium

Fig. E-2. Liquid Density of Selenium

4. Vapor Density

Vapor densities have been measured by several investigators. Values of N , the average number of atoms per molecule in the vapor, have generally been reported by these investigators instead of the actual vapor density values. N was calculated from the perfect gas law,

$$N = \frac{RT\rho_v}{AP}, \quad (E-2)$$

where T , P , and ρ_v were determined experimentally, and A represents the atomic weight of selenium (78.96). A plot of N vs. temperature is shown in Fig. F-4. The vapor density curve, which was determined from Eq. E-2 by the author, is shown in Figs. E-3 and E-4.

5. Heat of Vaporization

The heat of vaporization was calculated by means of the Clausius-Clapeyron equation:

$$\lambda = -2.3026 \left[\frac{d \log P}{d(1/T)} \right] \frac{P}{T} (v_v - v_l), \quad (E-3)$$

where $d(\log P)/d(1/T)$ and P/T were obtained from the vapor pressure curve, and v_v and v_l were obtained from the vapor density and liquid density data. The heat of vaporization is plotted against temperature in Fig. E-5.

Due to the reliability of the Clausius-Clapeyron equation, the vapor pressure data, and the vapor density data, the calculated heats of vaporization were considered to be accurate within $\pm 5\%$.

Fig. E-3. Vapor Density of Selenium

Fig. E-4. Low-Range Vapor Density of Selenium

Fig. E-5. Heat of Vaporization of Selenium

6. Liquid Thermal Conductivity

The thermal conductivity of liquid selenium (k_ℓ) has been measured by Mogilevskii (M9) in the temperature range of 225 to 300°C; the value of k_ℓ at 300°C was found to be 6.5×10^{-4} cal/cm-sec-°C, with k_ℓ showing a slight decrease with temperature. The fact that Mogilevskii's thermal conductivity data for liquid sulfur are in good agreement with other well-accepted thermal conductivity data for liquid sulfur indicates that Mogilevskii's data are reliable.

Benguigui (B7) has also measured the thermal conductivity of liquid selenium and has confirmed the data of Mogilevskii. However, Benguigui found a slight increase in the thermal conductivity with temperature. Perron (P2) measured the thermal conductivity of pure liquid Te and of Se-Te liquid alloys; by extrapolation, he calculated the thermal conductivity of liquid selenium to be within 12% of the value obtained by Mogilevskii. Abdullaev (A1) measured k_ℓ to be approximately 5×10^{-3} cal/cm-sec-°C between 217 and 250°C, almost an order of magnitude higher than the data obtained by the other investigators. Abdullaev's data were considered to be unreliable.

While it is not certain whether the thermal conductivity of liquid selenium increases or decreases with temperature, the available data do show that the effect of temperature on the thermal conductivity is very small. A value of 6.5×10^{-4} cal/cm-sec-°C was therefore considered to be the most reliable value for the thermal conductivity of liquid selenium over the temperature range of interest in this investigation.

7. Surface Tension

The surface tension (σ) of selenium has been measured by several investigators. The data are shown in Fig. E-6. The straight line drawn through the data points was selected to minimize the uncertainty in σ over the temperature range from 500 to 700°C, which was the temperature range over which σ was evaluated in correlating the boiling heat transfer data. The uncertainty in σ was estimated to vary between $\pm 5\%$ and $\pm 15\%$ in this temperature range.

8. Liquid Viscosity

The viscosity (μ_l) of liquid selenium has been measured by several investigators. The data are shown in Fig. E-7, where $\log \mu_l$ is plotted against $1/T$. The uncertainty in μ_l was estimated to range between $\pm 10\%$ (at temperatures between 220 and 375°C) and $\pm 20\%$ (at a temperature of 725°C).

Fig. E-6. Surface Tension of Selenium

Fig. E-7. Viscosity of Liquid Selenium

Appendix F. Structure of Selenium

1. Solid

As a solid, selenium exists in several allotropic forms. The most stable allotrope, referred to as "grey", "metallic", and sometimes "hexagonal" selenium is a grey crystalline substance with a semimetallic appearance. It has a hexagonal structure consisting of long, parallel, helical chains of selenium atoms. Another crystalline form consists primarily of Se_8 rings and an amorphous form consists of long disordered chains. Upon heating, all the allotropes are readily converted to the more stable grey selenium. Due to its electrical properties, which make it suitable for use in rectifiers, the grey selenium is by far the most important of the various allotropes (B2, E2, G3).

2. Liquid

When grey selenium melts at 217°C , it yields a liquid with an extremely high viscosity. Due to the high viscosity of the liquid and the molecular nature of the various allotropes, it has been inferred that long chains of selenium atoms and Se_8 rings must be present in the liquid. A recent investigation of the liquid structure by means of x-ray analysis and infra-red spectroscopy has confirmed the presence of Se_8 rings and long chains in the liquid (K5). This is the general conclusion, though another study has concluded that Se_6 molecules constitute the predominant ring structure in the liquid (R2).

Gee (G2) and Eisenberg and Tobolsky (E1) have developed a theory of equilibrium between the rings (assumed to be Se_8) and the chains. By utilizing their theory and experimentally determining the average chain

length of liquid selenium at several temperatures between 220 and 320°C, Keezer (K1) was able to estimate the average chain length of liquid selenium over a wide range of temperatures. He measured the viscosity of pure and thallium-doped selenium as a function of temperature and thallium concentration. By determining the thallium concentration at which the viscosity of thallium-doped selenium equaled the viscosity of pure selenium, Keezer was able to estimate the average chain length at each temperature considered. In his analysis, Keezer assumed that thallium atoms terminate free chain ends until saturation of the chain ends occurs. The decrease in viscosity with increasing thallium concentration, after chain-end saturation, is then due to the breaking up of the polymer chains by the thallium atoms. The average chain length determined by Keezer is plotted versus temperature in Fig. F-1, where Z is the average number of Se_8 units per chain.

By relating the magnetic susceptibility to the average chain length, Massen (M4) was also able to estimate the average chain length for liquid selenium. His results generally agree within a factor of two to three with those of Keezer.

To estimate the weight percent of Se_8 rings in liquid selenium, Briegleb (B16) quenched liquid selenium at several temperatures. Se_8 rings are known to be soluble in CS_2 , while the long chain material is not. Thus, Briegleb determined the weight percent of rings as the percent of soluble material in each sample. The weight percent of Se_8 rings (ϕ) in liquid selenium is plotted versus the temperature in Fig. F-2.

Fig. F-1. Average Chain Length of Liquid Selenium (K1)

Fig. F-2. Weight Percent Seg Rings in Liquid Selenium (B16)

3. Vapor

The composition of saturated selenium vapor has been studied by several investigators. A mass-spectrographic analysis by Berkowitz and Chupka (B9, B10) indicated that all molecular species from Se_2 to Se_8 were present, Se_2 , Se_6 , and Se_5 appearing to be major components between 300 and 700°C.

By assuming that only Se_2 and Se_6 exist in the vapor and by utilizing their vapor density measurements, Preuner and Brockmoeller (P3) have estimated the partial pressures of Se_2 and Se_6 . Rau (R1) and Illarionov and Lapina (I2) assumed that the vapor consists of Se_2 , Se_4 , Se_6 , and Se_8 and computed the partial pressures consistent with their own total vapor density measurements. Brebrick (B14, B15) employed his optical density measurements of the vapor to obtain the partial pressure of Se_2 in the vapor.

A plot of the mole fraction of Se_2 in the saturated vapor (y_2) versus the temperature is presented in Fig. F-3. Since the results of the various investigators were in good agreement, the values of y_2 shown in Fig. F-3 were considered to be reliable. On the other hand, the mole fractions of all species other than Se_2 were very inconsistent. The mole fractions of each molecular species reported by these investigators are presented in Table F-1. The composition of saturated selenium vapor must therefore be considered uncertain for all components other than Se_2 .

The average number of atoms per molecule in the saturated vapor (N), determined from vapor density measurements, is plotted versus temperature in Fig. F-4 (see Appendix E).

Fig. F-3. Mole Fraction of Se_2 in Saturated Selenium Vapor

Table F-1. Mole Fractions of Molecular Species
in Saturated Selenium Vapor

Molecular Species	T (°C)	y (mole fraction)				
		Investigator				
		Berkowitz (B9, B10)	Brebrick (B14, B15)	Illarionov (I2)	Rau (R1)	Preuner (P3)
Se ₂	300	0.10	0.07	0.05	0.03	0.10
Se ₃	300	0.00	-	-	-	-
Se ₄	300	-	-	0.31	0.15	-
Se ₅	300	0.50	-	-	-	-
Se ₆	300	0.25	-	0.39	0.25	0.90
Se ₇	300	0.10	-	-	-	-
Se ₈	300	0.02	-	0.25	0.57	-
Se ₂	500	0.25	0.18	0.20	0.18	0.23
Se ₃	500	0.01	-	-	-	-
Se ₄	500	-	-	0.19	0.16	-
Se ₅	500	0.55	-	-	-	-
Se ₆	500	0.08	-	0.45	0.52	0.77
Se ₇	500	0.04	-	-	-	-
Se ₈	500	0.01	-	0.16	0.14	-
Se ₂	700	0.40	0.43	0.40	0.39	0.42
Se ₃	700	0.02	-	-	-	-
Se ₄	700	-	-	0.12	0.11	-
Se ₅	700	0.50	-	-	-	-
Se ₆	700	0.03	-	0.38	0.46	0.58
Se ₇	700	0.01	-	-	-	-
Se ₈	700	0.00	-	0.10	0.04	-

Fig. F-4. Average Number of Atoms per Molecule in Saturated Selenium Vapor

Appendix G. Analysis of Errors

The probable error in a variable F , which is a function of several variables ($X_1, X_2, \dots, X_n$) is:

$$\Delta F = \left[\left(\frac{\partial F}{\partial X_1} \Delta X_1 \right)^2 + \left(\frac{\partial F}{\partial X_2} \Delta X_2 \right)^2 + \dots + \left(\frac{\partial F}{\partial X_n} \Delta X_n \right)^2 \right]^{1/2}, \quad (G-1)$$

where Δ represents the probable error in each variable. The probable errors in the variables ($X_1, X_2, \dots, X_n$) are assumed to be independent.

1. Boiling Data

The boiling heat velocities (q_b) reported in Section VII were calculated by Eq. C-4. The close agreement between the reported q_b , obtained from the measured temperature profiles in the copper cylinder, and the heat velocities determined from an energy balance around the copper cylinder (see Appendix C) showed that significant systematic errors were not present in any of the quantities used to calculate q_b . The agreement between the boiling water data and the data of Lyon (16) and Vachon (VI) (see Section VII-A) also indicated that large systematic errors in the boiling heat transfer data obtained with selenium were unlikely.

Equation C-2 was derived from the following equation:

$$\beta = \frac{(y_3^2 - y_2^2)(T_2 - T_1) - (y_2^2 - y_1^2)(T_3 - T_2)}{(y_2 - y_1)(y_3^2 - y_2^2) - (y_3 - y_2)(y_2^2 - y_1^2)}, \quad (G-2)$$

where y_1, y_2 , and y_3 represent the distances of the thermocouple junctions from the top of the copper cylinder (see Fig. C-1). By applying Eq. G-1 to Eqs. G-2 and C-3 through C-11, the following equations were

derived for the probable errors in q_b , $(T_w - T_v)$, and h_b :

$$\frac{\Delta q_b}{q_b} = \left[\left[\frac{\Delta k_{Cu}}{k_{Cu}} \right]^2 + \left[\frac{4.2(T_2 - T_1)\Delta y_1}{\beta} \right]^2 + \left[\frac{2.34\Delta(T_2 - T_1)}{\beta} \right]^2 \right]^{1/2}, \quad (G-3)$$

$$\frac{\Delta(T_w - T_v)}{(T_w - T_v)} = \frac{1}{(T_w - T_v)} \left[\left[1.02\Delta(T_1 - T_v) \right]^2 + (T_t - T_v)^2 \left[\left[\frac{\Delta q_b}{q_b} \right]^2 + \left[\frac{\Delta X_{ss}}{X_{ss}} \right]^2 + \left[\frac{\Delta k_{ss}}{k_{ss}} \right]^2 \right] \right]^{1/2}, \quad (G-4)$$

$$\frac{\Delta h_b}{h_b} = \left[\left[\frac{\Delta q_b}{q_b} \right]^2 + \left[\frac{\Delta(T_w - T_v)}{(T_w - T_v)} \right]^2 \right]^{1/2}. \quad (G-5)$$

The values of $(\Delta k_{Cu}/k_{Cu})$, Δy_1 , $\Delta(T_2 - T_1)$, ΔX_{ss} , $(\Delta k_{ss}/k_{ss})$, and $\Delta(T_1 - T_v)$ were estimated to be 0.02, 0.01 in., 0.5°F, 0.002 in., 0.02, and 3°F, respectively. The estimation of $\Delta(T_2 - T_1)$ was based on the fact that all thermocouples in the copper cylinder and test condenser were made from the same spool of wire and consequently had nearly identical calibrations (see Appendix B). Probable errors in y_2 , y_1 , β , and $(T_2 - T_v)$ can be shown to have negligible effect on $\Delta(T_w - T_v)$.

In the range of q_b investigated in this research, $(\Delta q_b/q_b)$ ranged between 4.7% and 10.9%. Due to the relatively high $(T_w - T_v)$, $\Delta(T_w - T_v)/(T_w - T_v)$ ranged between 1.6% and 5.4%. $(\Delta h_b/h_b)$ ranged between 5.2% and 11.2%.

2. Condensation Data

The close agreement between the condensing heat transfer data obtained with Dowtherm A and predictions of the standard Nusselt equation (see

Section VII-A) indicated that large systematic errors in the condensing heat transfer data obtained with selenium were very unlikely.

By applying Eq. G-1 to Eqs. C-17 and C-26 through C-31, the following equations were derived for the probable errors in q_c , $(T_v - T_o)$, and h_c :

$$\frac{\Delta q_c}{q_c} = \left[\left[\frac{\Delta M_{air}}{M_{air}} \right]^2 + \left[\frac{\Delta(T_{out} - T_{in})}{(T_{out} - T_{in})} \right]^2 + \left[\frac{\Delta \epsilon}{\epsilon} \right]^2 \right]^{1/2}, \quad (G-6)$$

$$\frac{\Delta(T_v - T_o)}{(T_v - T_o)} = \frac{1}{(T_v - T_o)} \left[[\Delta(T_v - T_j)_{av}]^2 + \left[\frac{2.35 \times 10^{-3} \Delta q_c}{k_{ss}} \right]^2 + \left[\frac{1.365 q_c \Delta r_t}{k_{Ag}} \right]^2 \right]^{1/2}, \quad (G-7)$$

$$\frac{\Delta h_c}{h_c} = \left[\left[\frac{\Delta q_c}{q_c} \right]^2 + \left[\frac{\Delta(T_v - T_o)}{(T_v - T_o)} \right]^2 \right]^{1/2}, \quad (G-8)$$

where r_t represents the radial position of the thermocouple junctions in the test condenser, and

$$\epsilon = \frac{Q_{end\ 1}}{Q_c} + \frac{Q_{end\ 2}}{Q_c} + 1. \quad (G-9)$$

$(\Delta M_{air}/M_{air})$, $\Delta(T_{out} - T_{in})$, $\Delta(T_v - T_j)_{av}$, and Δr_t were estimated to be 0.02, 1°F, 2°F, and 0.01 in., respectively. Based on an estimated uncertainty of $\pm 50\%$ in the calculated values of $(Q_{end\ 1}/Q_c)$ and $(Q_{end\ 2}/Q_c)$, $(\Delta \epsilon/\epsilon)$ ranged between 0.04 and 0.05. Probable errors in $c_{p,air}$, A_c , k_{Ag} , and k_{ss} can be shown to have negligible effect on Δq_c , $\Delta(T_v - T_o)$, and Δh_c .

Most of the error in q_c resulted from the errors in $(Q_{end\ 1}/Q_c)$ and

$(Q_{\text{end } 2}/Q_c)$. $(\Delta q_c/q_c)$ ranged between 4.5 and 5.5%. Due to the high values of $(T_v - T_o)$, $\Delta(T_v - T_o)/(T_v - T_o)$ ranged between 0.59 and 1.1%. $(\Delta h_c/h_c)$ ranged between 4.6 and 5.6%.

Nomenclature

a	Thermal diffusivity, sq ft/hr
A	Area, sq ft
c_p	Heat capacity, Btu/lb-°F
D	Outside diameter of condensing tube, ft
E	Voltage, volts
g	Acceleration of Gravity
g_c	Newton's conversion factor
h	Heat transfer coefficient, Btu/sq ft-hr-°F
h_b	Boiling heat transfer coefficient, Btu/sq ft-hr-°F
h_c	Condensing heat transfer coefficient, Btu/sq ft-hr-°F
H	Height of condensing ring in auxiliary condenser, in.
I	Current, amperes
k	Thermal conductivity, Btu/ft-hr-°F
L	Distance above boiling plate, in.
M	Molecular weight
N	Average number of atoms per molecule in saturated selenium vapor
P	Pressure, Torr (mm Hg) or lb _f /sq ft
ΔP	Pressure difference corresponding to boiling superheat, lb _f /sq ft
Pr	Prandtl number
q	Heat velocity, Btu/sq ft-hr
q_b	Boiling heat velocity, Btu/sq ft-hr
q_c	Condensing heat velocity, Btu/sq ft-hr
Q	Heat flux, Btu/hr
Q_{air}	Sensible heat gained by air coolant, Btu/hr
Q_{in}	Power supplied to heater, Btu/hr

Q_{loss}	Heat loss below boiling plate, Btu/hr
Q_{wall}	Heat conducted along wall of test chamber, Btu/hr
r	Bubble radius, ft
r_o	Outside radius of condensing tube, ft
R	Perfect gas constant
t	Time, hr
T	Temperature, °F
ΔT	Boiling superheat, °F
ΔT_{cal}	Calibration (correction) for measured temperature differences, °F
T_{below}	Average surface temperature below boiling plate, °F
T_{in}	Coolant temperature at inlet of test condenser, °F
T_{out}	Coolant temperature at outlet of test condenser, °F
v	Specific volume, cu ft/lb
V	Volumetric flow rate, cfm
W	Mass flow rate of condensate, lb/hr
y	Distance from top of copper cylinder, ft
y_2	Mole fraction of dimer in saturated vapor
Z	Average number of Se_8 units per chain in saturated liquid
λ	Latent heat of vaporization, Btu/lb
ρ	Density, lb/cu ft
σ	Surface tension, lb _f /ft
σ_c	Condensation coefficient
μ	Viscosity, lb/ft-hr

Subscripts

air	Property of air coolant
b	Boiling

c	Condensing
j	Measured by one of the 8 thermocouples in test condenser
l	Liquid
m	Measured (experimentally determined)
Nu	Predicted by Nusselt equation for filmwise condensation on horizontal tubes
o	At surface of test condenser
pred	Predicted by semi-empirical correlating equations for nucleate pool boiling
r	Reduced
s	Saturation
sl	At surface of condensate film
t	At top of copper cylinder
v	Vapor
w	At boiling surface
1 through 6	Measured by the six thermocouples in copper cylinder
7 through 14	Measured by the eight thermocouples in test condenser

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