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The effect of diameter on the film boiling behavior of liquid nitrogen

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THE EFFECT OF DIAMETER ON THE FILM BOILING BEHAVIOR OF LIQUID NITROGEN

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ABSTRACT

A film boiling heat transfer study was conducted for liquid nitrogen at atmospheric pressure. The heat transfer surfaces used were cylinders with outside diameters of 0.450, 0.650, 0.850, and 1.000 inches. The temperature differences covered were from approximately $150^{o}F$ to approximately $600^{o}F$.

The data indicate that the heat flux will decrease to a minimum then increase slightly as the diameter of the heat transfer element is increased at constant temperature difference. The heat transfer coefficient is expected to approach a flat plate correlation as the diameter approaches infinity.

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CHAPTER I

INTRODUCTION

This investigation is concerned with the film boiling of liquid nitrogen from horizontal cylinders of differing diameters. The typical boiling curve as shown in Figure 1 was first predicted by Nukiyama(1)* in 1934. The curve is a plot of the logarithm of the heat flux from the heating surface as a function of the logarithm of the temperature difference between the surface and the surrounding saturated liquid.

This curve is divided into four regions. As the temperature of the heat transfer surface is raised slightly above the saturation temperature of the liquid, convection currents circulate the liquid and evaporation occurs at the liquid surface. This behavior characterizes the first region on the figure and is identified as the convection region.

As the temperature of the heat transfer surface is further increased, bubbles begin to form on the heater surface at discrete nucleation sites. The point where the bubbles initially form is the beginning of the nucleate boiling region. The number of nucleation sites will increase as the temperature of the heater surface is

*Numbers in parentheses refer to listing in Bibliography

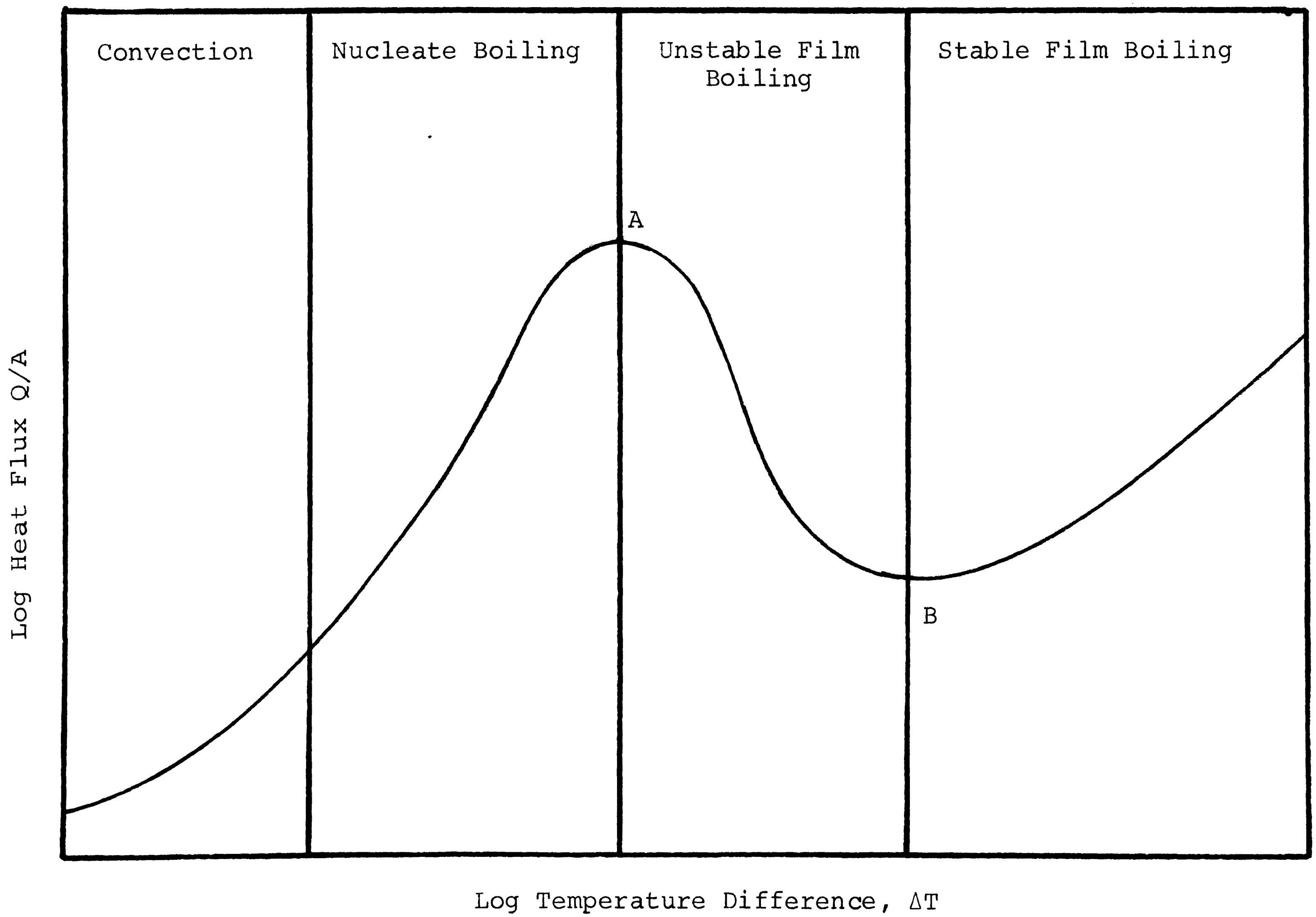

Figure 1. A Typical Boiling Heat Transfer Curve

increased, and the heat flux will increase to a maximum (Point A). Point A is known as the burnout point and is characterized by the formation of a vapor film over the heat transfer surface. At this point in the boiling curve, the very high heat transfer rate in the nucleate boiling region decreases because of the vapor film formation. Point A therefore describes the beginning of the third region called the unstable film boiling region. In this region the film is continuously forming and collapsing as the temperature difference increases and the heat transfer rate decreases because of the partial vapor film. This condition continues with increased temperature difference until point B is reached. This point is known as the Liedenfrost Point, or minimum heat flux point. The Liedenfrost Point describes the minimum conditions for maintaining stable film boiling. Point B therefore initiates the fourth region of the boiling heat transfer curve (Figure 1) which is known as the stable film boiling region.

It is this stable film boiling region which is investigated in this study. Specifically, the primary purpose of this study is to investigate the effect of diameter of cylindrical heat transfer surfaces on the film boiling heat transfer characteristics of liquid nitrogen.

CHAPTER II

PREVIOUS WORK

There have been numerous investigations concerning the film boiling of cryogens. The most significant of these works as they pertain to the effect of varying diameters upon film boiling heat flux will be discussed here. For a more complete summary of film boiling heat transfer in general, the reader is referred to the paper by Clements and Colver[2].

Bromley[3] was the first investigator to suggest a method for predicting heat transfer coefficients for film boiling. His analysis was similar to Nusselt's[4] development for condensation. He assumed a mechanism in which the vapor film is in dynamic equilibrium with the surrounding liquid. As the vapor rises under the action of bouyant forces, vapor is added to the film from the surrounding liquid. His final equation for horizontal cylindrical heat transfer elements is given below.

$$h = (\text{constant}) \left| \frac{k_v^3 \rho_v (\rho_\ell - \rho_v) g \lambda}{D \; \Delta T \; \mu_v} \right|^{1/4} \qquad (1)$$

The value of the constant was found experimentally to be 0.62.

Bromley[5] later suggested that for better results λ should be replaced by λ'', where λ'' is given below.

$$\lambda'' = \lambda \left| 1.0 + \frac{0.34 C_p \Delta T}{\lambda} \right|^2 \qquad (2)$$

Banchero, Barker and Boll[6] in their study of liquid oxygen boiling from cylindrical heat transfer surfaces suggest that the diameter range where Bromley's equation is valid is between diameters of 0.069 and 0.127 inches. These authors found that the following equation correlated their data for a range of diameters of 0.025 to 0.75 inches.

$$h = a\left(\frac{1}{D} + C\right) \left| \frac{k_v^3 \rho_v (\rho_\ell - \rho_v) g \lambda}{\Delta T \ \mu_v} \right|^{1/4} \qquad (3)$$

In the above equation a is a temperature and fluid dependent constant, and C is equal to 36.5 inches^{-1}. In their study of film boiling oxygen, they found that the heat flux at a constant temperature difference decreased with increasing diameter up to a diameter of 0.75 inches. Their diameter correlation suggests an asymptotic approach to the flat plate solution.

The concept of Taylor instability has been used in recent years by several authors to develop correlations for film boiling on flat horizontal surfaces.

This concept states that if a liquid lies above a vapor, waves will form at the interface between the liquid and vapor and rupture will occur allowing vapor to rise to the surface as bubbles. The wavelength of the smallest wave which can form at the interface is given by the following equation.

$$\lambda_c = 2\pi \left| \frac{g_c \sigma}{g(\rho_\ell - \rho_v)} \right|^{1/2} \tag{4}$$

Breen and Westwater[7] extended the Taylor instability theory to film boiling on horizontal cylinders. They conducted experiments with a large variation in diameter and with two different fluids, isopropanol and Freon 113. The range of diameters used was from 0.185 to 1.895 inches. Their data showed that as the diameter was increased the heat transfer coefficient began decreasing rapidly to a minimum and then increased slowly to a flat plateau. The minimum occurred at a diameter of approximately 0.65 inches. Their final correlation equation is given below.

$$h \left| \frac{\lambda_c \mu_v \Delta T}{k_v^3 \rho_v (\rho_\ell - \rho_v) g \lambda} \right|^{1/4} = 0.59 + 0.069 \left(\frac{\lambda_c}{D}\right) \tag{5}$$

Flanigan[8] suggested that for substances which follow the law of corresponding states to a high degree of accuracy (CO, O_2, A, N_2, Xe, Kr, and CH_4) the usual functional relationship,

$$h = f(T, D, \text{Physical properties}) \tag{6}$$

should be replaced by equation 7 given below.

$$h = f(T_r, D, P_r) \tag{7}$$

where,

$$T_r = \frac{T_{saturation} + \frac{\Delta T}{2}}{T_c} \tag{8}$$

This transformation to the use of reduced temperature and reduced pressure is possible because the physical properties of corresponding states vapors and liquids are functions of these properties. Flanigan determined the functional relationship for equation 7 using a least squares fitting technique with the data of Park(9), Banchero, Barker and Boll(6), Sciance(10), and Flanigan(8). His final equation is

$$h = \alpha_2 \left(\frac{1}{D} + C\right) P_r^{1/4} \tag{9}$$

where,

$$\alpha_2 = 8.49 - 8.24T_r + 2.97T_r^2 - 0.267T_r^3 \tag{10}$$

Flanigan and Park(11) present data for the film boiling of liquid nitrogen and argon using cylinders of

0.55, 0.75, and 0.95 inch diameters. They suggest that over this range of diameters, at a given temperature difference, the heat flux decreases to a minimum and then increases. They also suggest that the value of the minimum heat flux is strongly dependent upon the system pressure. At low pressure levels, the minimum was well defined; whereas, the heat flux dependence upon diameter decreased as the system pressure approached the critical pressure.

Frederking[12] studied film boiling of nitrogen from small wires and concluded that the heat transfer coefficient would decrease with increasing diameter. His range of diameters was from 0.0134 to 0.051 millimeters, well below the range of diameters where the minimum occurred in the studies of Flanigan and Park[11], and Breen and Westwater[7].

It should be noted that the only investigators to note the heat flux passing through a minimum as the diameter is increased are Breen and Westwater[7] and Flanigan and Park[11]. This is probably due to the fact that most investigators have studied a limited range of diameters. It should also be noted that these two investigations do not agree as to where this minimum occurs.

Breen and Westwater[7] postulated that the minimum in heat transfer coefficient occurred when the cylinder was equal to "the most dangerous wavelength," denoted by λ_D. λ_D is defined by the following equation.

$$\lambda_D = \sqrt{3}\ \lambda_c \tag{11}$$

λ_c = the critical wavelength (equation 4 above)

In the Flanigan and Park study(11), "the most dangerous wavelength" varied from 0.00187 feet at 655 psig to 0.0291 feet at 56 psig for argon and from 0.00306 feet at 453 psig to 0.0332 feet at atmospheric pressure for nitrogen. The diameters of the heat transfer elements used in their study were 0.0458, 0.0625, and 0.0792 feet, which are well above "the most dangerous wavelength".

The objective of the present investigation is to provide film boiling heat transfer data for design purposes and to attempt to define more accurately the minimum heat flux as a function of the diameter of the heat transfer surface.

CHAPTER III

DESCRIPTION OF EXPERIMENTAL EQUIPMENT

The equipment used in this investigation consisted of four cylindrical heat transfer elements of differing diameters, a direct current power source, a voltmeter, an ammeter, a millivolt meter, a thermocouple selector switch, and an insulated metal dewar.

The heat transfer element design consisted of three primary components: a copper cylindrical sleeve, an inner core which served as a source for heat generation, and thermal insulating end plates (See Figure 2).

Four copper cylinders three inches long were machined to outside diameters of 1.000, 0.850, 0.650, and 0.450 inches and inside diameters of 0.500, 0.500, 0.350, and 0.250 inches, respectively. In each end of each cylinder, three thermocouple wells 0.052 inches in diameter and one inch in depth were drilled 90 degrees apart. Solder was then melted into the thermocouple wells until they were full. While the solder was molten, 25 gauge glass coated copper-constantan thermocouples were sunk into the wells. Beads had been welded onto the thermocouples with an Argon Thermocouple Welder.

The inner heat generation elements consisted of a coil of number 26 gauge tungsten wire and a method of securing this coil within the outer copper cylinder. For

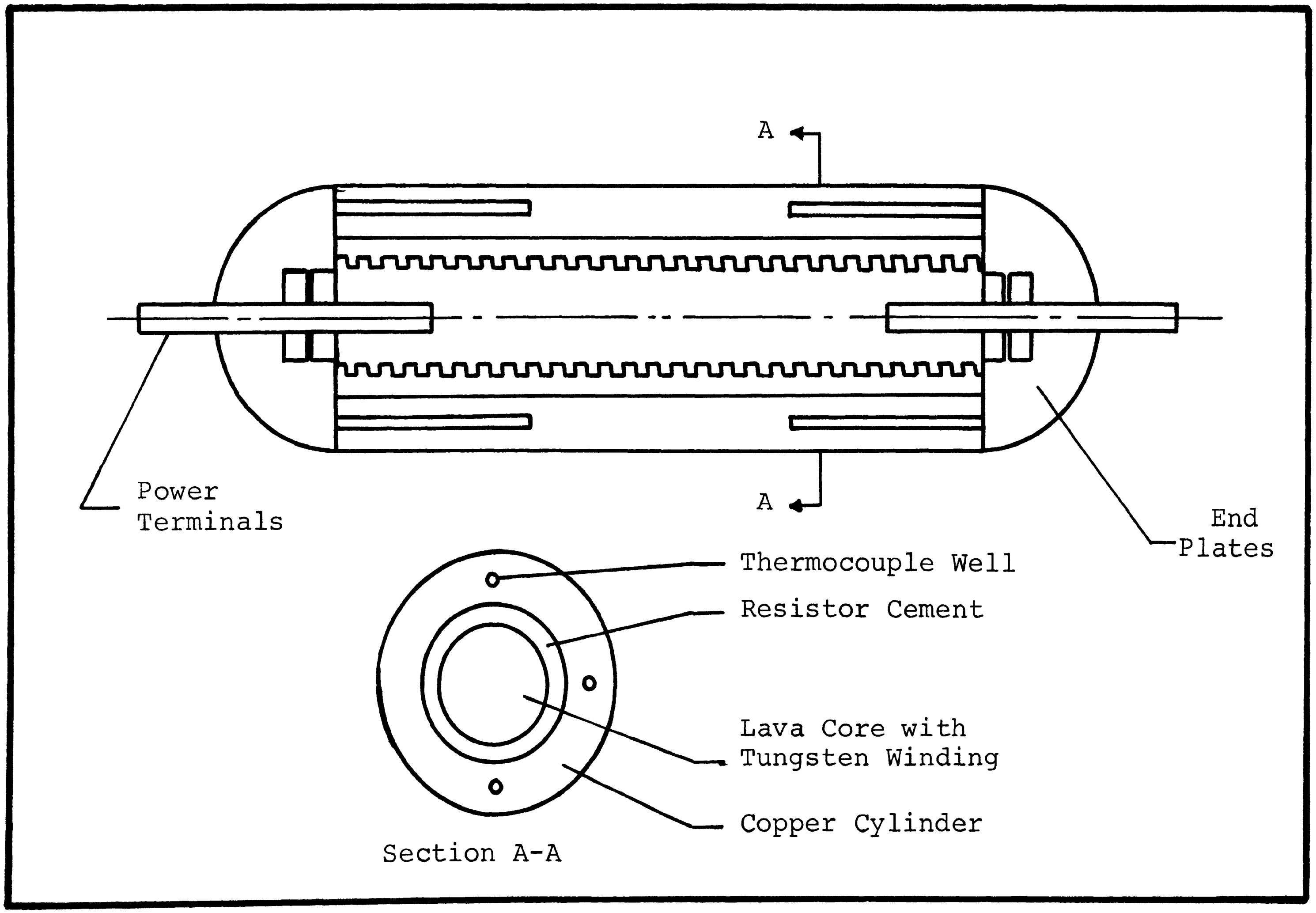

Figure 2. Heat Transfer Element

the 1.000, 0.850, and 0.650 inch heaters, lavatite cores three inches long were machined to diameters of 0.400, 0.400, and 0.250 inches, respectively. These cores were threaded 12 lefthanded threads per inch, such that the grooves were deep enough to fully embed the tungsten wire. After the lavatite cores had been heat treated in an oven at 1500°F for twenty-four hours, the tungsten wire was wrapped into the grooves and secured at each end of the core by 4-40 machine screws which were screwed into the ends of the lavatite cylinder. These screws also served as power terminals for the heaters. After the wire was wrapped on the cores, the cores were coated with Sauereisen Electrical Resistor Cement, Number 7 paste, and dried in an oven for six hours at 180°F. The cores were then cemented into the copper cylinders with the same cement and again dried under the same conditions.

A lavatite cylinder could not be used in the core for the 0.450 inch heater because its small size caused failure due to shear because of the torsional stress of the coiled tungsten wire. For this core, an aluminum wrapping guide was machined to 0.150 inches in diameter and threaded 12 threads per inch. The tungsten wire was then wrapped on this guide and then removed from the guide. The coil thus produced was fitted with 4-40 terminal screws and dipped into the cement paste to form the core. The same drying and fitting procedure was followed for this heater as for the other three larger heaters.

The thermal insulating end plates were formed by molding the same cement paste in an approximate hemispherical shape at each end of the heater. The end plates were oven dried also at 180^{o}F and waterproofed by coating with a dilute solution of acetic acid.

An improved design for a cylindrical heat transfer element is provided in Appendix C of this investigation.

The power for the experiments was provided by a Trygon Electronics, Model C36-50 DC Power Supply, rated at 0 - 36 volts and 0 - 50 amperes. System potential was measured at the heater element by a COHU Series 510 digital voltmeter, and current was measured by a Westinghouse Type PX-161 DC ammeter.

The boiling vessel was an insulated aluminum cylindrical dewar 8 inches inside diameter and 9.6 inches in depth.

The thermocouple leads from the heating surfaces were lead to a liquid nitrogen reference function and to a Leeds and Northrup rotary thermocouple switch. This switch was used in conjunction with a Digitec Model 454, 115 Volt, 60 cycle, 0.25 Ampere, 40 MV full scale millivolt meter accurate to $\pm$ 0.005 millivolts.

The above described equipment was utilized in the configuration illustrated in Figure 3.

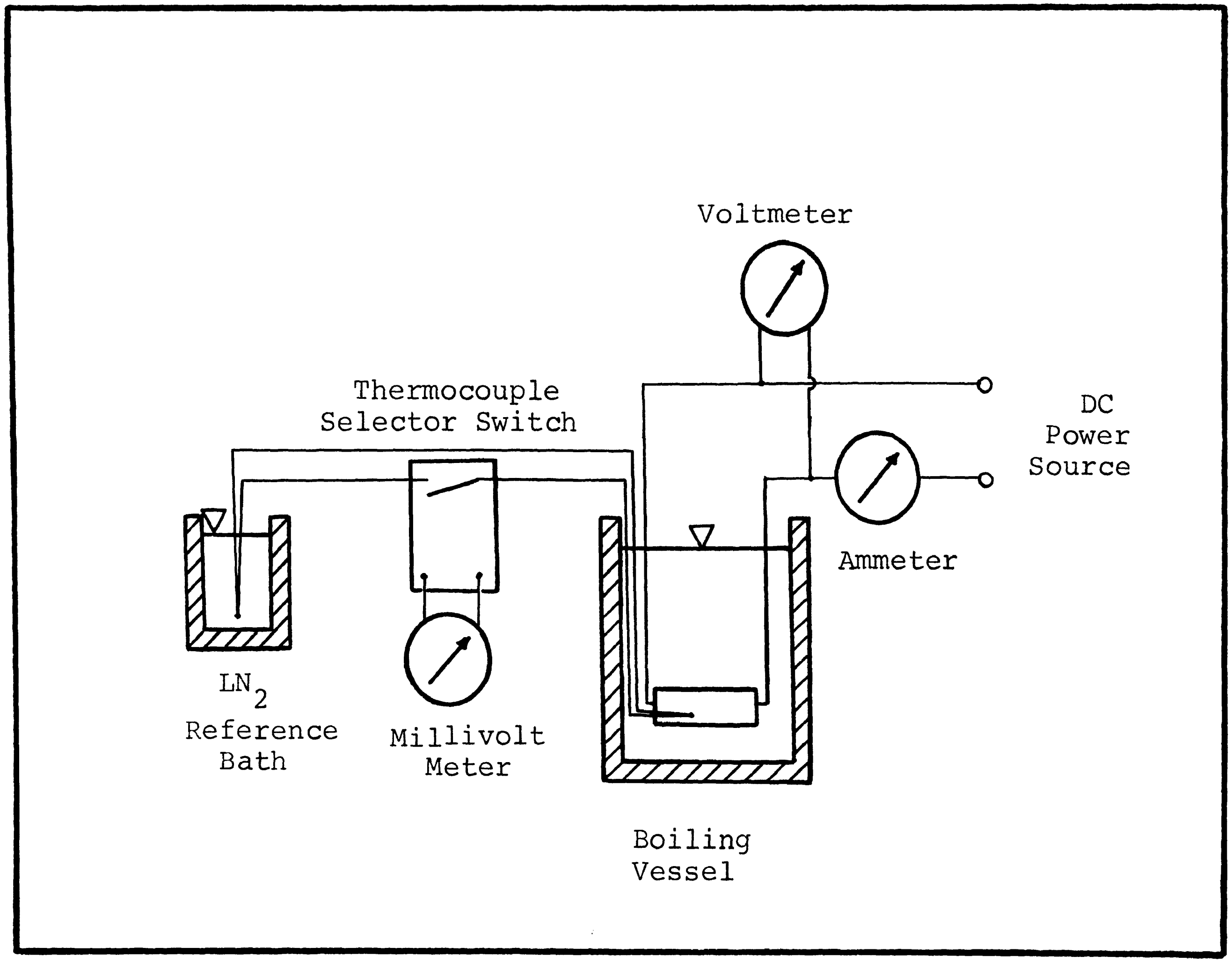

Figure 3. Experimental Equipment Set-Up

CHAPTER IV

EXPERIMENTAL PROCEDURE

Experimental data were taken at atmospheric pressure for each heater in the horizontal position. To begin each run for a given heater, the dewar was filled with liquid nitrogen while the heater was suspended in its horizontal position within the dewar. The system was then allowed to reach equilibrium. At this steady state condition, the "zero" of the millivolt meter was verified.

Power was then turned on and slowly increased through the nucleate boiling region. As the burnout point was reached and passed, evidenced by a large rapid increase in temperature difference, the power was reduced and the system was allowed to reach steady state at a point in the lower portion of the film boiling curve. The time required to reach a steady state condition for the initial data point was not less than two hours for all heaters and all data runs. The power supply was operated in the "voltage adjust" mode. The condition of steady state was defined when, over a five minute period of time, there was neglible change in either the system current or the temperature difference measurements.

After the first data point was taken, the power was increased slightly and the system was again allowed to reach steady state. For this data point, as well as for each succeeding point where small power changes were involved, a minimum of thirty minutes was allowed to elapse even though in some cases a steady state condition may have appeared to occur earlier.

In a similar fashion, additional data points were taken under increased power conditions until such time as a temperature difference of approximately 700^{o}F was attained. At this time, power was reduced, and additional data points were taken as the power was being reduced. The reproducibility of the data verified the validity of each experimental run. This condition also supports the thesis that hysterisis is not apparent in the film boiling region. Data points were taken under reduced power conditions until such time as the temperature difference would not sustain film boiling conditions. This latter condition was apparent when there was observed a sudden rapid decrease in temperature difference.

Data were taken in this fashion for each of the four cylindrical heaters of diameters of 0.450, 0.650, 0.850, and 1.000 inches, respectively. At least two runs were conducted for each heater, and in no case were the two runs for a given heater conducted on the same day.

CHAPTER V
RESULTS

The experimental data obtained during the runs were converted by conventional means. "Conversion Tables for Thermocouples" published by Leeds and Northrup Company were used to convert the electromotive force generated in the thermocouples directly into the difference in temperature between the heat transfer surface and the saturation temperature of the boiling liquid nitrogen. This difference in temperature is hereafter referred to as Temperature Difference or Delta Temperature (ΔT). The steady state potential and current across the heat transfer element were converted directly into heat flow using the conversion factor from watts to British Thermal Units per hour.

Data points thus obtained were plotted for each heater diameter on rectilinear plots and are presented in Figures 4,5,6, and 7. These data were fitted using a least squares routine which selects the local best polynomial to fit the experimental data by minimizing the standard deviation.

In this case, the first degree polynomial was selected for each set of data. For the 0.450 inch heater, the average deviation is 2.67 percent and the maximum

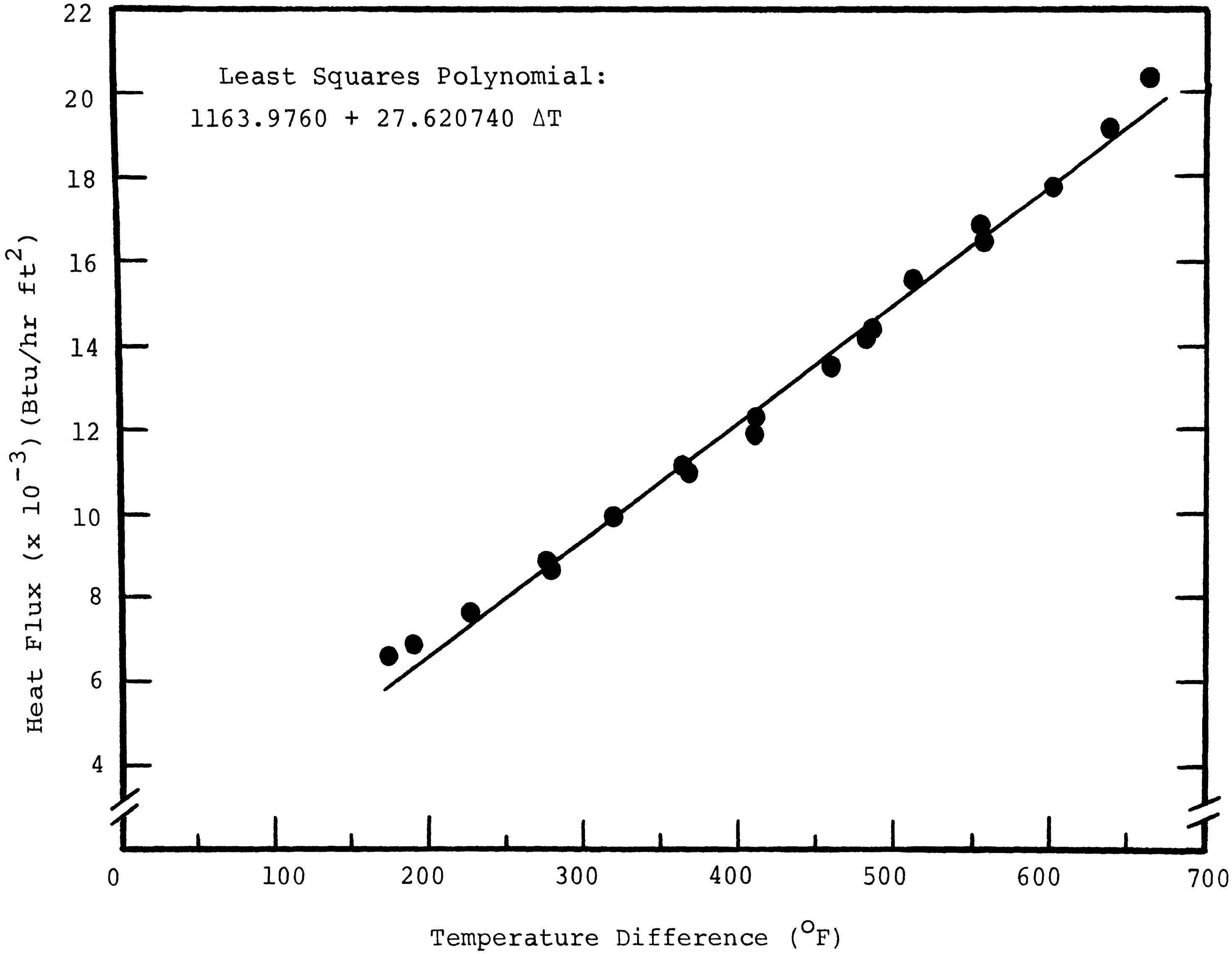

Figure 4. Film Boiling of Liquid Nitrogen, 0.450 inch Diameter Heater

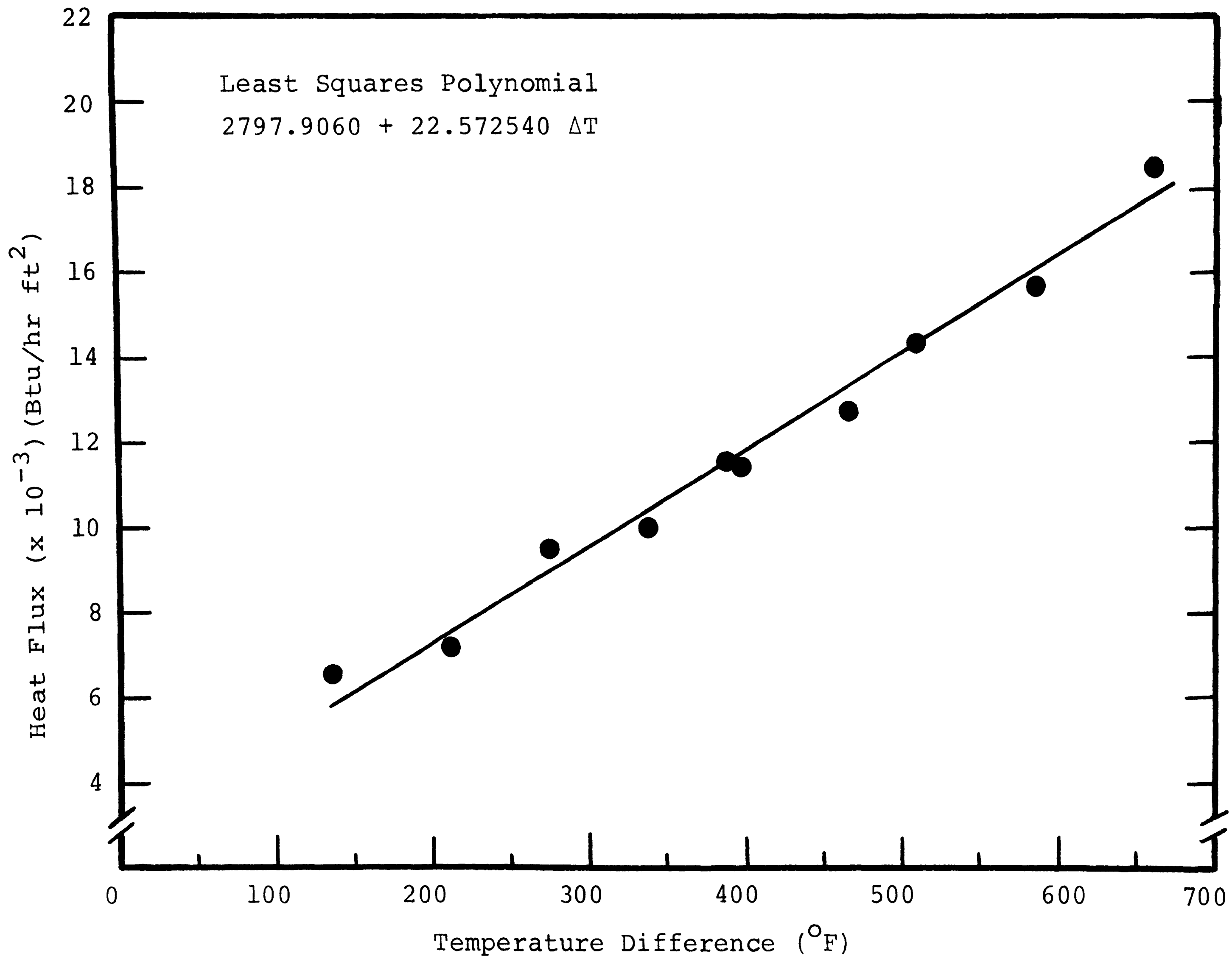

Figure 5. Film Boiling of Liquid Nitrogen, 0.650 inch Diameter Heater

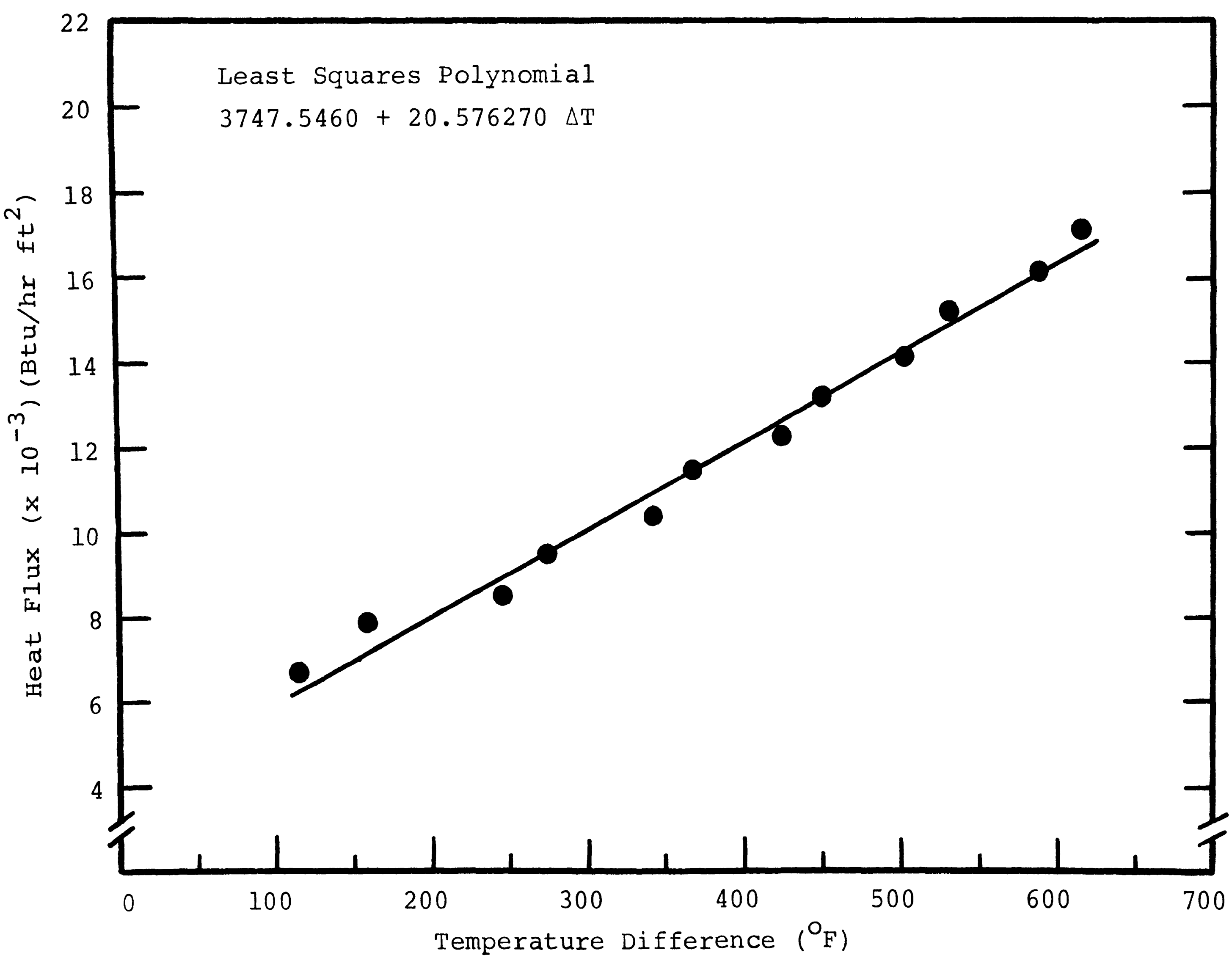

Figure 6. Film Boiling of Liquid Nitrogen, 0.850 inch Diameter Heater

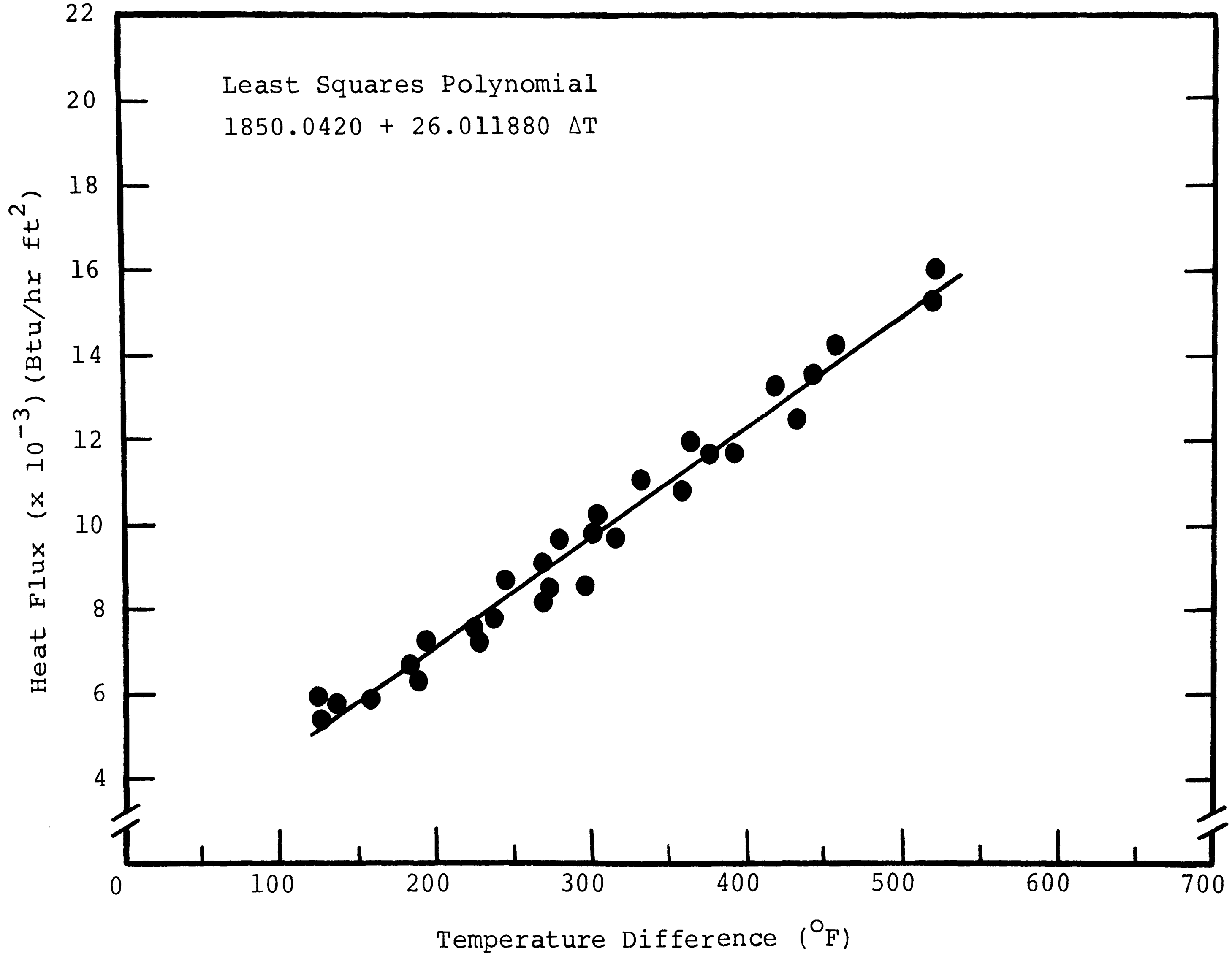

Figure 7. Film Boiling of Liquid Nitrogen, 1.000 inch Diameter Heater

deviation is 9.53 percent. The data for the 0.650 inch heater yielded an average deviation of 3.85 percent with a maximum deviation of 9.70 percent. The 0.850 inch heater yielded an average deviation of 3.06 percent, with a maximum deviation of 6.70 percent, and the 1.000 inch heater an average of 4.52 percent with a maximum of 13.01 percent.

CHAPTER VI
DISCUSSION OF RESULTS

The effect of diameter on the heat flux and heat transfer coefficient for film boiling of liquid nitrogen over the range of diameters used in this investigation is shown in Figure 8. It is noted that for temperature differences above 300°F the heat flux and heat transfer coefficient both appear to go through a minimum value. This trend was predicted by Flanigan and Park(11) for liquid nitrogen and was also observed by Breen and Westwater(7) for isopropanol and Freon 113. For Flanigan and Park this minimum occured at a diameter of 0.75 inches. Breen and Westwater observed a minimum heat flux and heat transfer coefficient for isopropanol and Freon 113 at a diameter of 0.65 inches which corresponds to "the most dangerous wavelength", as defined by equation 11. From the Flanigan and Park study(10), "the most dangerous wavelength" for liquid nitrogen varies from 0.03675 inches at 453 psig to 0.3985 inches at atmospheric pressure. The diameters used in that study were 0.55, 0.75 and 0.95 inches. Since their range of diameters was well above "the most dangerous wavelength", their observed minimum values at a diameter of 0.75 inches appear unrelated to "the most dangerous wavelength" for liquid nitrogen.

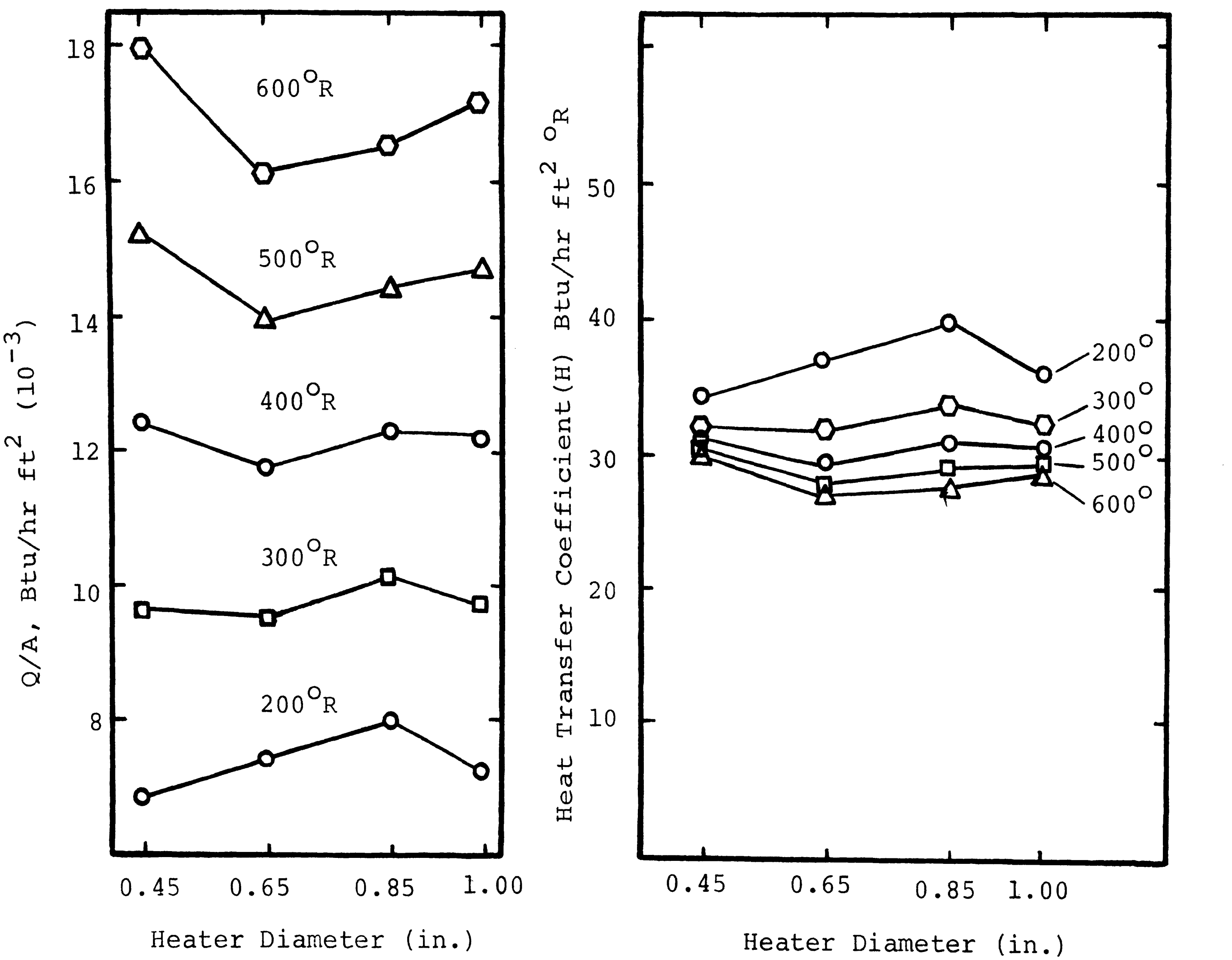

Figure 8. Nitrogen Film Boiling with Constant Delta Temperature

This investigation appears to substantiate a minimum heat flux and heat transfer coefficient in the vicinity of the diameter value of 0.75 inches. It is noted, however, that this minimum value did not occur for the lower temperature differences. At temperature differences of 300°F and 200°F maxima were observed. The explanation for this phenomenon is not readily apparent. A possible explanation is that at temperature differences near the Liedenfrost Point, other variables such as surface chemistry and roughness and boiling pool turbulence may become significant. Another explanation could be that the mechanism controlling film boiling changes as the temperature difference becomes small.

The data of Flanigan[8], Park[9], Frederking[12], Flynn, Draper and Roos[13], Weil[14], and this investigation at a temperature difference of 400°F are represented in Figure 9. It should be noted from this figure that the heat transfer coefficient decreases sharply with increasing diameter until the range of diameters studied in this investigation is reached. The heat transfer coefficient then appears to increase slightly. The flat plate correlation of Frederking[15] is shown in the figure for comparison purposes. It is expected that as the diameter of the cylindrical heat transfer element is increased a theoretical equation for flat plates is approached.

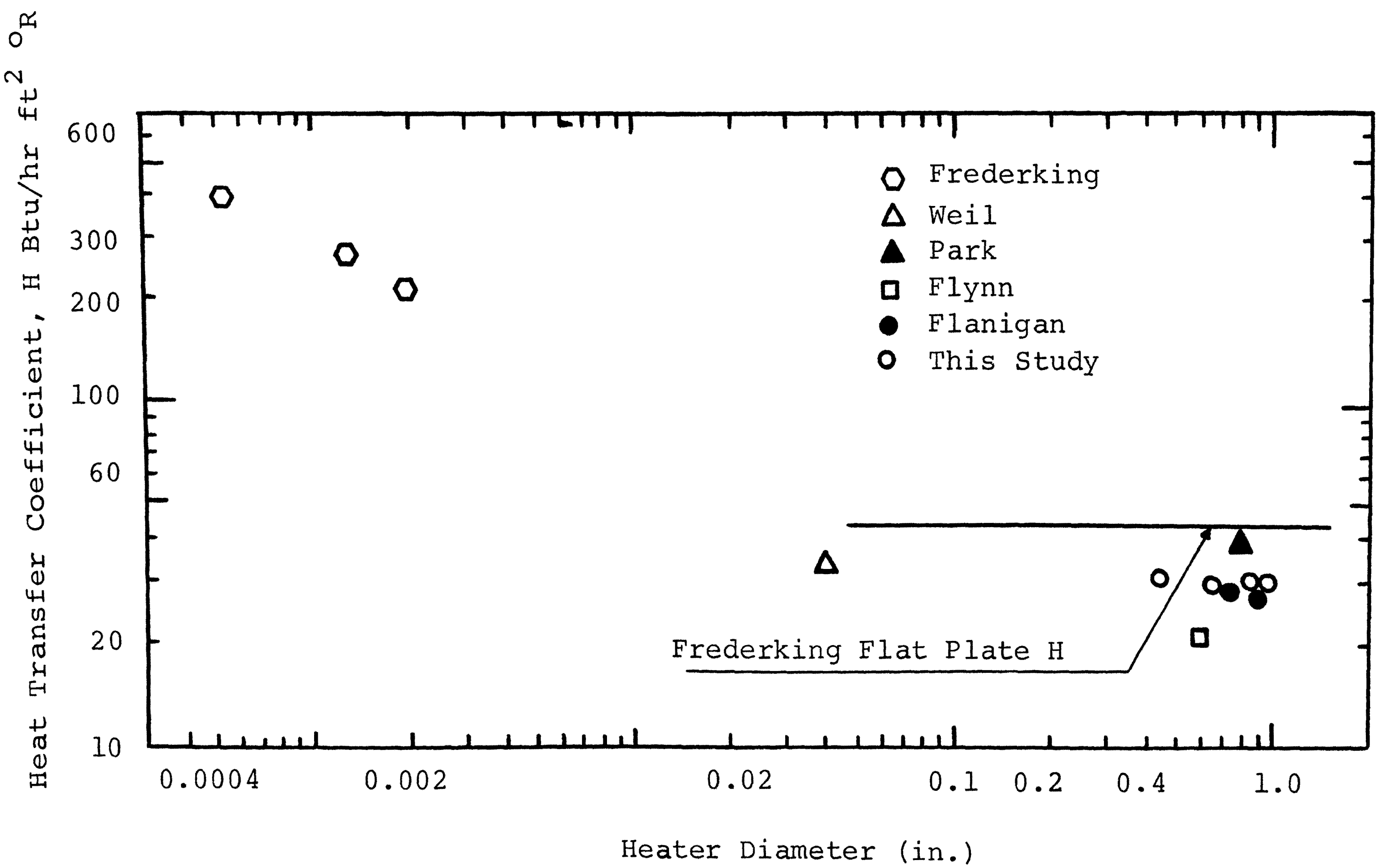

Figure 9. Film Boiling Data with Constant 400^oR Delta Temperature, Atmospheric Pressure

It should also be noted that there is very little data available in the diameter range of 0.002 inches to 0.4 inches. Further study is recommended in this diameter range.

The data of this investigation were compared with the correlation of Flanigan[8], Equation (9). This comparison is shown in Figures 10 and 11. For each heat transfer element diameter, the heat transfer coefficient curve predicted by the Flanigan equation is shown as well as the observed data from this investigation. For all data points, the average deviation between the observed and the predicted heat transfer coefficients is 6.08 percent. For the 0.450 inch heater, the average deviation is 5.97 percent, and for the 0.650 inch heater, the average is 7.00 percent. The 0.850 inch heater yielded an average deviation of 0.74 percent, while for the 1.000 inch heater, it was 10.60 percent. It is noted that for all heaters, the observed and predicted heat transfer coefficients agreed quite closely at the higher temperature differences. The largest errors occurred at temperature differences ranging from approximately 150^{o}F to approximately 250^{o}F.

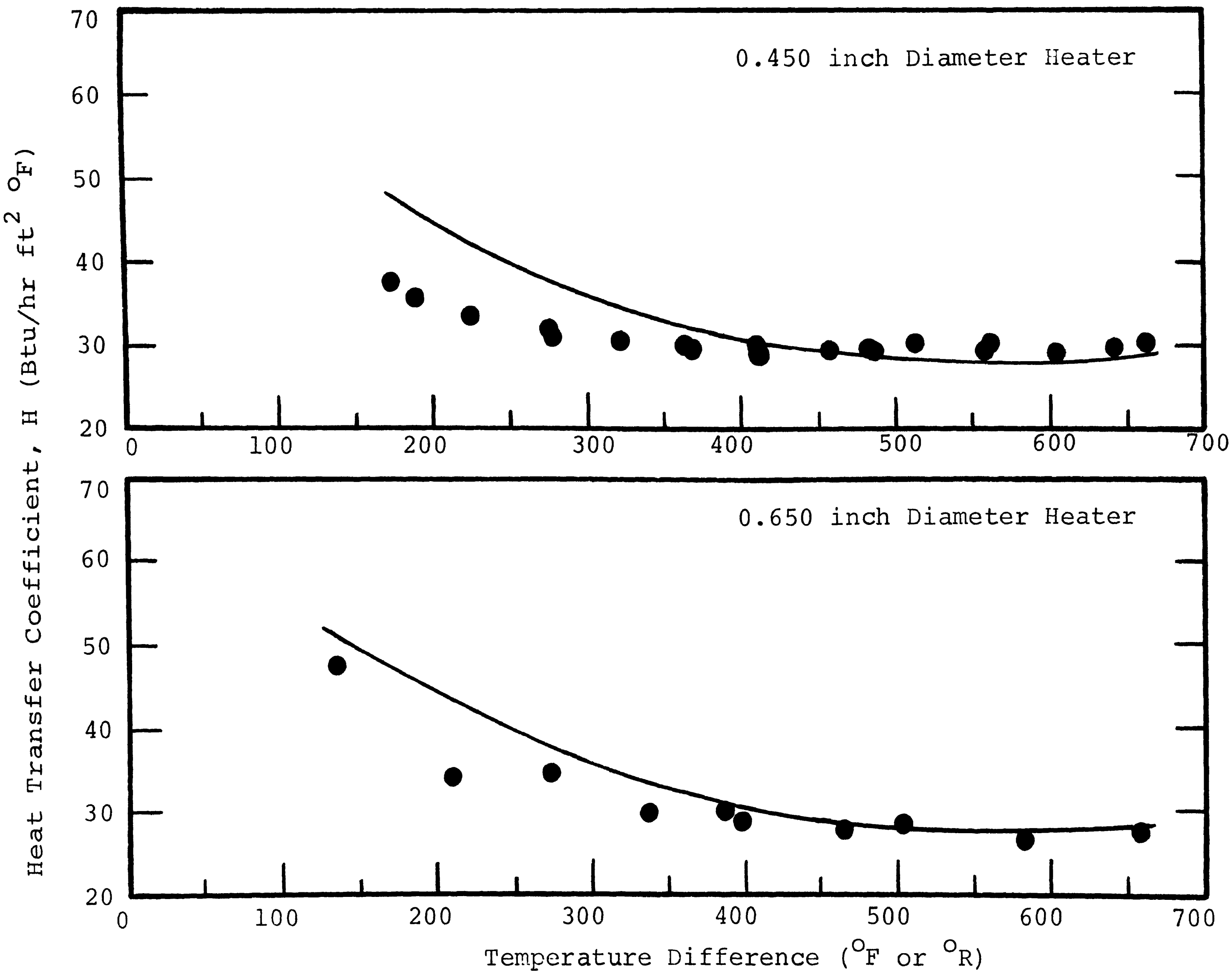

Figure 10. Comparison of Film Boiling Data with Flanigan's Correlation

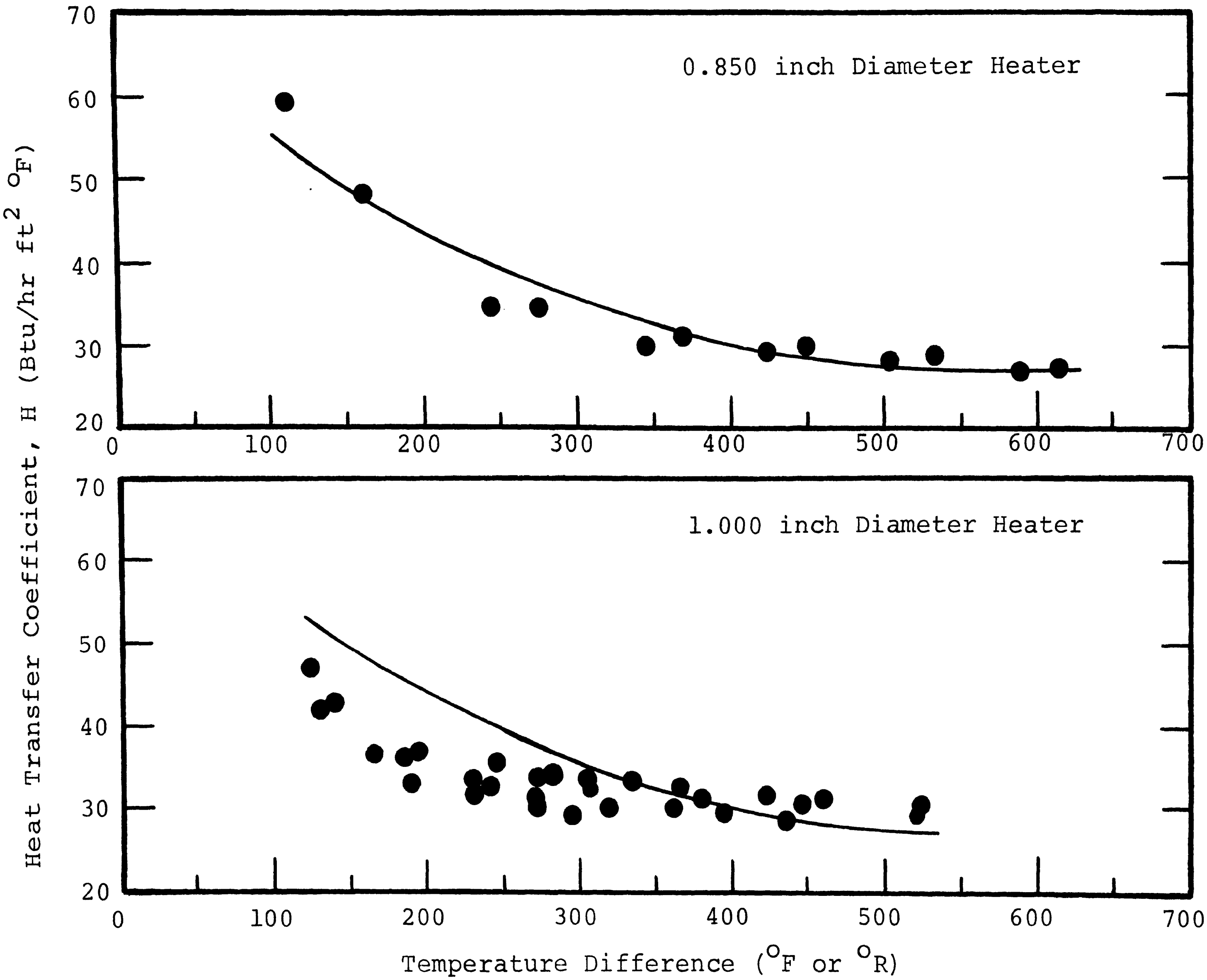

Figure 11. Comparison of Film Boiling Data with Flanigan's Correlation

CHAPTER VII
DISCUSSION OF ERROR IN MEASUREMENT

Park[9] discussed in detail the inherent error associated with nucleate and film boiling; therefore, error analysis will not be discussed in great detail here.

The current could be read accurately within $\pm$ 0.125 amps and the voltage could be read accurately within $\pm$ 0.005 volts. The product of these errors is less than 1 percent.

The thermocouples could be read accurately to $\pm$ 0.005 millivolts which corresponds to $\pm$ 0.25^{o}F. The thermocouples were calibrated by placing the reference junction in liquid nitrogen and the four heaters in a liquid nitrogen bath and in an ice water bath. The difference between the measured temperature difference and the known temperature difference was so small that no correction was required.

The magnitudes of heat lost from the ends of the cylinders can be calculated for the cylinders if the equation $q = -kA \frac{dT}{dx}$ is written in the form $q = -kA \frac{\Delta T}{\Delta x}$ and if temperature measurements are made axially along the cylinder. Park[9] did this in his investigation and found that the maximum heat loss was 4.8%. Banchero, Barker, and Boll[6] reported that the axial temperature gradient was virtually eliminated in copper cylinders if the length to diameter ratio exceeded 3.75. The length to diameter ratios used in this investigation were 6.67, 4.62, 3.53, and 3.00. Thus, the heat loss would be in the range equal to or less than that reported by Park.

CHAPTER VIII
CONCLUSIONS

1. Diameter has a definite effect on the film boiling behavior of liquid nitrogen from cylinders. It appears that at temperature differences well removed from the Liedenfrost Point the heat flux will decrease to a minimum then increase as the diameter of the heat transfer element is increased.

2. As the diameter of a cylindrical heat transfer element approaches infinity the heat transfer coefficient will be expected to approach the value predicted by a flat plate correlation.

3. More data are needed to evaluate diameter effect on film boiling of nitrogen, particularly at low temperature differences and for heat transfer elements with diameters less than 0.450 inches.

4. The film boiling correlation of Flanigan[8], Equation (9), predicted the values of heat transfer coefficients within an average accuracy of 6.08 percent. This accuracy appears quite good when compared to normal experimental accuracy.

NOMENCLATURE

A	-	Area, ft^2
a	-	constant in equation 3
C	-	constant in equation 3
C_p	-	heat capacity, Btu/lb$^{\circ}$F
D	-	diameter, ft.
E	-	potential, volts
g	-	acceleration due to gravity, ft/sec^2
g_c	-	gravitational constant, $\frac{lb_m ft}{lb_f sec^2}$
h	-	heat transfer coefficient, Btu/hr ft^2 $^{\circ}$F
I	-	current, amp
k	-	thermal conductivity, Btu/hr ft^2 $^{\circ}$R/ft
L	-	length, ft
P	-	pressure, psi
Q	-	rate of heat transfer, Btu/hr
T	-	temperature, $^{\circ}$R
ΔT	-	temperature difference, $^{\circ}$R or $^{\circ}$F

Greek Symbols

α_2	-	constant in equation 9, $\frac{\text{Btu in (psia)}}{\text{hr ft}^2\ ^{\circ}\text{F}}$
σ	-	surface tension, lb/ft

NOMENCLATURE (continued)

λ_c - $\left(\frac{g\sigma}{g_c(\rho_\ell-\rho_v)}\right)^{1/2}$, ft.

μ - viscosity, lb/ft hr

ρ - density, lb/ft^3

λ - latent heat of vaporization, Btu/lb_m

Subscripts

c refers to the critical point

v refers to the vapor

ℓ refers to the liquid

r refers to reduced property, (T/T_c, etc.)

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Appendix A

EXPERIMENTAL DATA

Nitrogen

Atmospheric Pressure

Heat Transfer Surface

Diameter: 0.450 inches
Length : 3.000 inches

Least Squares Polynomial Coefficients (First Degree)

A(0) = 1163.9760
A(1) = 27.620740

Average Deviation = 2.667879

No.	Delta T	Heat Flux	Calc.Heat Flux	% Deviation	H
1	412.870	11942.10	12567.75	-5.23905	28.92459
2	369.170	11005.03	11360.72	-3.23201	29.91023
3	278.400	8627.14	8853.59	-2.62480	30.98830
4	190.320	6846.45	6420.75	6.21775	35.97334
5	483.050	14230.28	14506.17	-1.93878	29.45921
6	559.150	16517.98	16608.10	-0.54559	29.54124
7	604.950	17782.39	17873.14	-0.51029	29.39482
8	643.630	19172.57	18941.51	1.20519	29.78819
9	666.650	20313.53	19577.33	3.62418	30.47107
10	461.230	13492.58	13903.48	-3.04536	29.25349
11	411.560	12304.44	12531.57	-1.84585	29.89708
12	366.230	11083.39	11279.51	-1.76950	30.26349
13	321.160	9918.68	10034.64	-1.16920	30.88393
14	277.780	8855.96	8836.46	0.22015	31.88120
15	226.890	7664.35	7430.84	3.04667	33.78003
16	174.270	6607.10	5977.44	9.53139	37.91357
17	487.600	14413.07	14631.85	-1.51788	29.55920
18	515.470	15615.46	15401.63	1.36936	30.29364
19	556.040	16865.73	16522.21	2.03681	30.33186

Nitrogen

Atmospheric Pressure

Heat Transfer Surface

Diameter: 0.650 inches
Length : 3.000 inches

Least Squares Polynomial Coefficients (First Degree)

A(0) = 2797.9060
A(1) = 22.572540

No.	Delta T	Heat Flux	Calc. Heat Flux	% Deviation	H
1	211.670	7162.25	7575.83	-5.77447	33.83687
2	340.060	10083.32	10473.92	-3.87366	29.65160
3	399.060	11491.71	11805.70	-2.73234	28.79694
4	508.120	14364.62	14267.46	0.67641	28.27014
5	467.050	12819.82	13340.41	-4.06076	27.44850
6	586.410	15668.69	16034.66	-2.33569	26.71968
7	662.520	18569.69	17752.66	4.39983	28.02888
8	388.210	11555.91	11560.78	-0.04222	29.76715
9	276.190	9493.50	9032.21	4.85900	34.37308
10	137.520	6536.31	5902.08	9.70313	47.52986

Average Deviation = 7.00274

Nitrogen

Atmospheric Pressure

Heat Transfer Surface

Diameter: 0.850 inches
Length : 3.000 inches

Least Squares Polynomial Coefficients (First Degree)

A(0) = 3947.5460
A(1) = 20.576270

No.	Delta T	Heat Flux	Calc.Heat Flux	% Deviation	H
1	590.380	16170.23	16095.36	0.46302	27.38954
2	505.580	14175.80	14350.50	-1.23232	28.03870
3	425.860	12285.69	12710.16	-3.45492	28.84914
4	343.910	10361.83	11024.13	-6.39169	30.12863
5	245.970	8505.49	9008.69	-5.91622	34.57936
6	113.950	6744.25	6292.21	6.70258	59.18604
7	161.630	7781.35	7273.29	6.52923	48.14297
8	274.120	9573.28	9587.92	-0.15289	34.92366
9	368.630	11451.10	11532.57	-0.71141	31.06396
10	450.840	13273.71	13224.15	0.37333	29.44217
11	533.550	15219.05	14926.01	1.92542	28.52412
12	616.750	17121.42	16637.96	2.82374	27.76073

Average Deviation = 3.056393

Nitrogen

Atmospheric Pressure

Heat Transfer Surface

Diameter: 1.000 inches
Length : 3.000 inches

Least Squares Polynomial Coefficients (First Degree)

A(0) = 1850.0420
A(1) = 26.011880

No.	Delta T	Heat Flux	Calc.Heat Flux	%Deviation	H
1	297.540	8591.09	9589.62	-11.62282	28.87372
2	272.470	8202.49	8937.50	-8.96083	30.10419
3	230.170	7227.84	7837.20	-8.43067	31.40218
4	191.120	6282.14	6821.43	-8.58454	32.87013
5	128.460	5373.95	5191.53	3.39456	41.83363
6	318.650	9702.15	10138.72	-4.49981	30.44766
7	362.200	10845.96	11271.54	-3.92391	29.94467
8	395.630	11705.17	12141.12	-3.72441	29.58617
9	436.570	12544.98	13206.05	-5.26960	28.73531
10	273.960	8506.64	8976.26	-5.52059	31.05066
11	227.280	7636.53	7762.02	-1.64331	33.59966
12	186.430	6754.99	6699.44	0.82237	36.23337
13	137.520	5848.67	5427.20	7.20636	42.52960
14	272.990	9167.49	8951.03	2.36118	33.58177
15	305.790	10296.53	9804.22	4.78134	33.67189

Heat Transfer Surface - Diameter 1.000 inches (continued)

No.	Delta T	Heat Flux	Calc. Heat Flux	% Deviation	H
16	367.380	11987.67	11406.28	4.84993	32.63019
17	461.130	14240.25	13844.89	2.77630	30.88121
18	526.620	16050.26	15548.42	3.12672	30.47787
19	423.480	13300.39	12865.55]	3.26935	31.40736
20	336.000	11066.28	10590.03	4.30365	32.93538
21	282.610	9669.28	9201.26	4.84027	34.21420
22	246.400	8684.99	8259.37	4.90061	35.24751
23	195.350	7265.55	6931.46	4.59822	37.19247
24	125.770	5887.79	5121.55	13.01396	46.81393
25	160.480	5836.94	6024.43	-3.21210	36.37177
26	239.100	7803.45	8069.48	-3.40920	32.63675
27	303.500	9806.47	9744.65	0.63040	32.31126
28	379.180	11767.76	11713.22	0.46343	31.03476
29	446.300	13583.01	13459.14	0.91190	30.43471
30	523.170	15356.51	15458.67	-0.66523	29.35283

Average Deviation = 4.523912

Appendix B

SAMPLE CALCULATIONS

A. Sample calculations for first data point for 0.450 inch heater data:

$$E = 5.48 \text{ volts}$$

$$I = 18.80 \text{ amps}$$

$$\Delta T = 412.87^{o}F$$

$$D = 0.450 \text{ inches}$$

$$L = 3.000 \text{ inches}$$

1. Area (ft^2)

$$A = \frac{\pi DL}{144} = \frac{\pi(0.45)(3.0)}{144}$$

$$A = 0.0295 \text{ ft}^2$$

2. Heat Flux (Btu/hr ft^2)

$$Q/A = \frac{3.413EI}{A} = \frac{(3.413)(5.48)(18.8)}{0.0295}$$

$$Q/A = 11{,}942.10 \text{ Btu/hr ft}^2$$

3. Heat Transfer Coefficient (Btu/hr ft^2 ^oF)

$$H = \frac{Q/A}{\Delta T} = \frac{11{,}942.10}{412.87}$$

$$H = 28.92 \text{ Btu/hr ft}^2\ {}^oF$$

B. Sample calculation for predicted heat transfer coefficient from Flanigan[8] correlation, equation 9. First data point, 0.450 inch heater.

Data:

$$\Delta T = 412.87\ ^{o}F$$

$$T_c = 227^{o}R$$

$$P_c = 33.3 \text{ atm}$$

$$T_{saturation} = 460-321 = 139^{o}R$$

$$T_r = \frac{T_{saturation} + \Delta T/2}{T_c}$$

$$T_r = \frac{139 + 412.87/2}{227} = 1.52$$

$$\alpha_2 = 8.49 - 8.24T_r + 2.97T_r^{2} - 0.267T_r^{3}$$

$$\alpha_2 = 1.89$$

$$P_r = \frac{P}{P_c}$$

$$P_r = 1/33.3 = 0.03$$

$$H = \alpha_2 (\frac{1}{D} + C) P_r^{1/4}$$

$$H = (1.89)(\frac{1}{0.450} + 36.5)(0.03)^{1/4}$$

$$H = 30.43 \text{ Btu/hr ft}^2\ ^{o}R$$

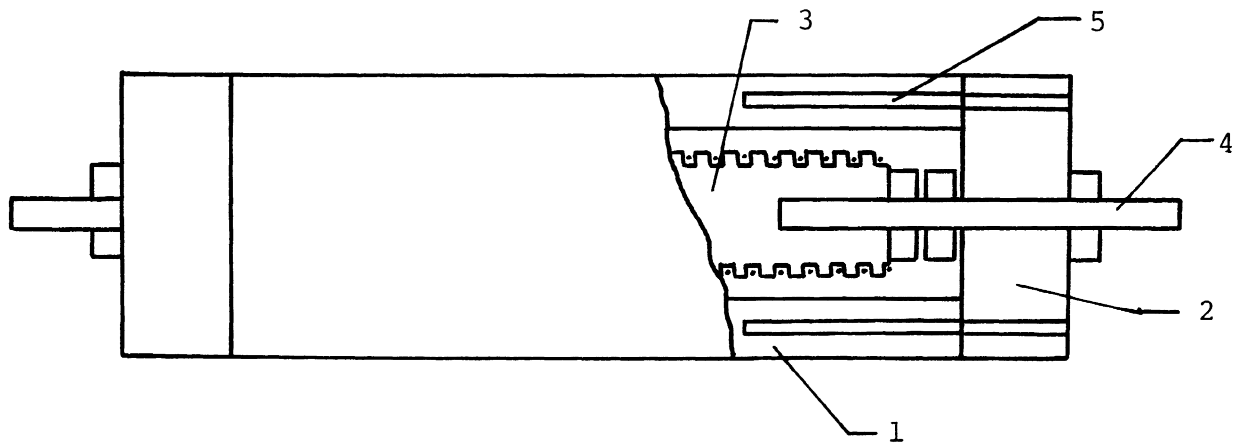
Appendix C

IMPROVED HEAT TRANSFER ELEMENT DESIGN

A cylindrical heat transfer element fabricated as shown in Figure C-1 is recommended for steady state film boiling heat transfer studies of cryogens. In this design the heat generation coil is recessed approximately one fourth inch into the copper cylinder such that the teflon end plates will fit flush against the cylinder and held firm by a 4-40 machine nut. A thermal insulating grease compound is used at the interfaces of the teflon and the copper cylinder,thermocouple wires and the 4-40 machine screw power terminals.

One distinct advantage of this design is that the resistor cement used to cement the heat generation core in place is completely isolated from the boiling fluid. Repeated exposure of the resistor cement to cryogenic fluid causes the cement to crack. The use of the teflon end plate therefore prolongs the life of a heat transfer element.

Another advantage of the use of teflon end plates is the reduction of heat losses from the ends. Although the end losses appear small when the molded cement end plates are used, they are reduced even further by the use of teflon which exhibits very good insulating characteristics.



1. Copper Cylinder
2. Teflon End Plate
3. Lava Core with Tungsten Winding
4. 4-40 Machine Bolts with Nuts
5. Thermocouple Wells for Measuring Surface Temperatures

Figure C-1. Improved Heat Transfer Element Design

VITA

William Joseph Wafer, son of Mrs. Willie Mae Wafer and the late Mr. Tom D. Wafer, was born in Monroe, Louisiana on November 16, 1935.

He attended Ouachita Parish High School of that city and was graduated in May, 1954. In July, 1954, he entered the United States Military Academy, West Point, New York and received his Bachelor of Science degree in June of 1958. He was commissioned in the United States Army at that time and has served as a regular army officer since that time. He has served various tours of duty within the Continental United States as well as in Vietnam, Korea, and Latin America. As of the time of this writing, he has attained the rank of Major. He entered the University of Missouri-Rolla in January 1970 as a candidate for the degree of Master of Science in Mechanical Engineering.

He is married to the former Miss Barbara-Ann Barger of Philadelphia, Pennsylvania and they have been blessed with three children.