HEAT AND MASS TRANSFER IN THE ABSENCE OF AIR

BY

JAMES JOSEPH HOWLEY

A THESIS

PRESENTED IN PARTIAL FULFILLMENT OF

THE REQUIREMENTS FOR THE DEGREE

OF

MASTER OF SCIENCE IN CHEMICAL ENGINEERING

AT

NEWARK COLLEGE OF ENGINEERING

Newark, New Jersey
1967

APPROVAL OF THESIS

HEAT AND MASS TRANSFER IN THE ABSENCE OF AIR

BY

JAMES JOSEPH HOWLEY

FOR

DEPARTMENT OF CHEMICAL ENGINEERING

NEWARK COLLEGE OF ENGINEERING

BY

FACULTY COMMITTEE

APPROVED: ____________________________

____

____

NEWARK, NEW JERSEY

JUNE, 1967

ACKNOWLEDGEMENT

The author wishes to express his sincere gratitude to Dr. Robert Casciano of the Newark College of Engineering for his guidance during the course of this work.

Appreciation is also extended to Mr. Wallace Grubman of National Starch and Chemical Corporation for the fabrication of the heat exchanger and the supply of other equipment used in this experiment.

TABLE OF CONTENTS

	PAGE
TITLE PAGE	i
APPROVAL PAGE	ii
ACKNOWLEDGEMENT	iii
LIST OF FIGURES	v
LIST OF GRAPHS	vi
LIST OF TABLES	vii
ABSTRACT	1
INTRODUCTION	3
THEORY AND METHOD OF CALCULATION	9
PROCEDURE	21
A. PREPARATION	21
B. DETERMINING THE SYSTEM HEAT GAIN	21
C. DEVELOPING THE ICE FILM	22
D. ACCURACY OF THE DATA	23
DESCRIPTION OF THE APPARATUS	24
DISCUSSION	31
CONCLUSION	33
RECOMMENDATIONS	34
NOMENCLATURE	35
APPENDIX	36
REFERENCES	59

LIST OF FIGURES

		PAGE
FIGURE 1	FLOW SHEET	8
FIGURE 2	HEAT EXCHANGER ASSEMBLY	27
FIGURE 3	HEAT EXCHANGER DETAILS	28
FIGURE 4	ICE COATING DEVELOPED	29
FIGURE 5	ICE COATING DEVELOPED	30
FIGURE 6	LIQUID CHILLER	58

LIST OF GRAPHS

		PAGE
GRAPH 1	WILSON PLOT, RUN 1	40
GRAPH 2	WILSON PLOT, RUN 2	41
GRAPH 3	WILSON PLOT, RUN 3	42
GRAPH 4	WILSON PLOT, RUN 4	43
GRAPH 5	WILSON PLOT, RUN 5	44
GRAPH 6	WILSON PLOT, RUN 6	45
GRAPH 7	ROTAMETER CALIBRATION	46
GRAPH 8	TEMPERATURE DATA, RUN #1	47
GRAPH 9	TEMPERATURE DATA, RUN #2	48
GRAPH 10	TEMPERATURE DATA, RUN #3	49
GRAPH 11	TEMPERATURE DATA, RUN #4	50
GRAPH 12	TEMPERATURE DATA, RUN #5	51
GRAPH 13	TEMPERATURE DATA, RUN #6	52
GRAPH 14	THERMAL CONDUCTIVITY OF FROST	53
GRAPH 15	SPECIFIC HT. OF ETH. GLYCOL SOLUTIONS	54
GRAPH 16	THERMAL CONDUCT. OF ETH. GLYCOL SOLUTION	55
GRAPH 17	VISCOSITY OF ETHYLENE GLYCOL SOLUTIONS	56
GRAPH 18	SPECIFIC GRAV. OF ETH. GLYCOL SOLUTIONS	57

LIST OF TABLES

		PAGE
TABLE 1	SYSTEM DATA	17
TABLE 2	TEST RESULTS	19
TABLE 3	SYSTEM HEAT GAINS	37
TABLE 4	WATER EVAPORATION RATE	38
TABLE 5	THERMOCOUPLE CALIBRATION	39

ABSTRACT

The effects of heat and mass transfer across a frosted heat exchanger operating in the absence of air was evaluated. It is felt that this work could be applied to processes such as desalination of water or the purification of materials by reverse sublimation.

The system consisted of a single ½" O.D. stainless steel tube mounted within a 3" O.D. plexiglas tube with the ends suitably flanged. A vacuum was maintained in the annular space. Refrigerated ethylene glycol passed through the inner tube in laminar flow. A series of runs with substantially air-free water vapor added to the annular space deposited an ice film on the metal tube. Small, fine-grained droplike sites of ice were observed at the start of each run. These soon joined together and formed a continuous layer of ice. The experiments, carried out at reduced pressures, for a 2½ to 4 hour period developed sufficient ice which could be measured, removed from the metal tube and weighed.

Five runs were made under vacuums of 1.3 mm. to 4.0 mm. Hg and 1 run at atmospheric pressure. From the concept of resistances in series, overall coefficients of heat transfer were calculated and compared to values found in the literature from frosted heat exchanger

experiments. The present experiments showed that improvement in heat transfer rates can be expected when air is evacuated from the system. For example, at 1.5 mm., the heat transfer coefficient was 6.72 compared to 0.662 at atmospheric pressure.

Wilson plots were also used to evaluate the surface coefficients. With this method it is possible to obtain an overall resistance of the system at infinite ethylene glycol velocity. From this is subtracted the know resistances of the metal and ice film. The remainder is the surface film resistance. This method gave good agreement with surface coefficients as calculated above.

INTRODUCTION

Sublimation covers the physical changes encountered by a substance in passing from a solid phase to a gas phase and back to a solid phase. The temperature at which the vapor pressure of the solid equals the total pressure of the gas phase in contact with it is defined as the sublimation point. To describe the condensation from gas to solid, the terms desublimation or reverse sublimation have been used.

The effect of a frost or ice film is important in the design of a reverse-sublimation process. Heat transfer rates will control the condensing system capacity. Material purification may be performed by causing the condensible portion of a vapor to freeze out on a refrigerated surface. This principle may be applied to water purification or other material which can be deposited as a solid from its vapor. This experiment was run to determine what effect the absence of air will have during a reverse-sublimation process. For convenience, the water-vapor to ice sublimation system was used.

A literature search showed the following references which are related to this experiment:

1. M.M. Chen and W. Rohsenow[1] considered heat, mass and momentum transfer with frost on the inside surface of a heat-exchanger tube. It was found that surface roughness of the frost contributed to higher coefficients of heat transfer during the initial phases of the experiment. As frosting progressed the degree of roughness may level off and remain fairly constant, and the insulating effect of the frost layer will then reduce the total heat transfer to lower values. Heat transfer results are evaluated in terms of the effective heat transfer coefficient, based on the tube wall to gas temperature difference.

$$h_e = \frac{q/A}{\Delta t}$$

For the low vapor concentrations encountered, the mass flux does not contribute significantly to the heat transfer to be included in the energy balance. In this work, the apparatus used consisted of two concentric copper tubes about 36" long in length and 5/16" and 11/16" inside diameters. Freezing took place in the inner tube. The experiment was run at pressures of 20 to 30 pounds per sq. in. Air was the moisture carrier and refrigerated air as the coolant. Runs took about 60 minutes.

The heat transfer coefficients increased at the beginning of a run as a rule. The authors attribute this to the roughness developed by the frost and present their theoretical study. With a moist air flow having a Re number of 22,100 and Δt of 20.7 $^{\circ}$F, the effective heat transfer coefficient as defined above averaged about 38 Btu/ft hr $^{\circ}$F.

2. W.L. Bryan[2] gives a report on the transfer of heat and mass by a bare surface coil. Apparatus consisted of a bank of staggered coils, six rows deep. Air was blown across the tube bank while water was circulated through the tubes. Here, a high water flow rate was used, and the refrigerant temperature could be maintained substantially constant throughout the coils. The heat and mass transfer were determined by the temperature and humidity measurements of the air before and after the coil. The coefficient of mass transfer was checked by measuring the rate of condensate from the coil. No frosting occured during these runs. Experimentally determined coil surface temperatures provided a direct evaluation of the heat and mass transfer coefficients.

Coefficients of heat transfer were correlated statistically as a function of the coil face velocity giving the equation:

$$h = 0.616\ V^{\frac{1}{2}}$$

Tabulated heat transfer coefficients show an h of 6.5 at a face velocity of 130 feet per minute to an h of 12.4 at a face velocity of 376 feet per minute.

3. K.O. Beatty Jr., E.B. Finch and E.M. Schoenborn[3] studied the effects of frosting on heat transfer. Results are presented of measurement on a four foot long surface and on a four inch long unit showing some of the effects of air velocity, air condition and cold surface temperature. The larger test apparatus is a vertical double-pipe unit. Conditioned air flowed upward through an annulus formed by a 4" diameter lucite tube and a 2" diameter brass cylinder arranged concentrically. A bore of 0.3125" was made in the brass cylinder and cold methanol flowed countercurrently through this passage. Air at a predetermined velocity, temperature, and humidity was passed upward through the annulus after the brass tube had been cooled to the desired temperature. All the data were taken within a Reynolds nimber range of 1,000 to 8,000. The smaller unit was used in the same general manner.

An effective heat transfer coefficient for the smaller test unit is defined as:

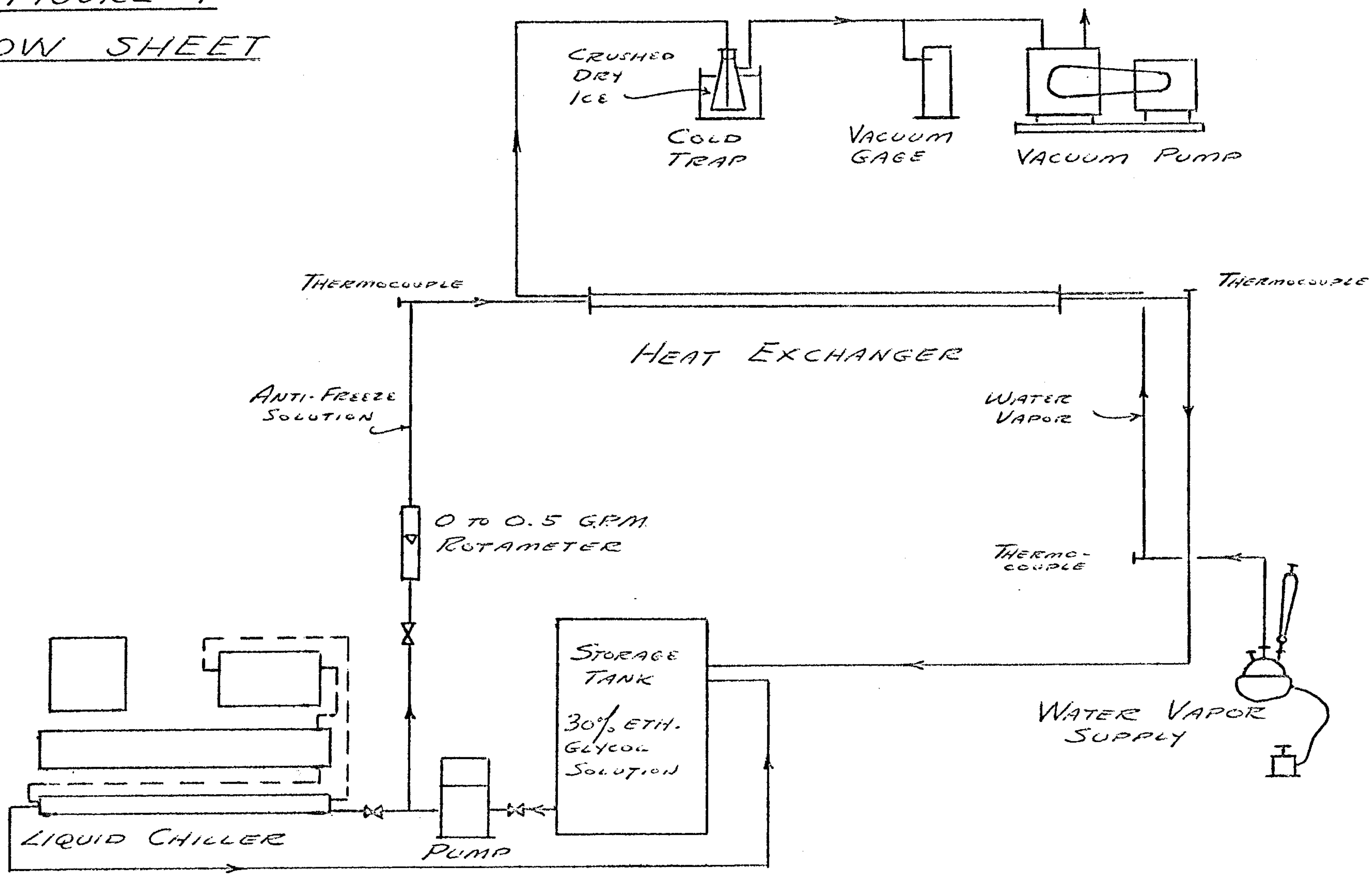
$$h_e = \frac{q/A}{t_a - t_s}$$

where t_a is the air bulk temperature and t_s is the metal temperature. The results show a h_e of 4 to 6 at Reynolds numbers of 3500. Variations in the air humidity and surface temperature was seen to affect the heat transfer coefficient even though the Reynolds number remained at 3500.

4. A.L. London and R.A. Seban[4] have reported on the rate of ice formation and present a general approximate method of analyzing the problem of freezing in liquids bounded by surfaces of various geometries. A solution to the freezing problem is presented with applications to ice formation at spherical, cylindrical, and plane boundaries. Application of the method is illustrated by examples in ice manufacture and quick freezing of food products.

FIGURE 1

FLOW SHEET



THEORY AND METHOD OF CALCULATION

A. Experimental Data

Consider experiment #1 with equipment arranged as illustrated on the Flow Sheet, Figure #1. The system was evacuated and the general procedure followed which gave data as follows:

Vacuum: 1.3 mm. Hg

Length of run: 2.5 hours

Quantity of ice formed: 78.8 grams

Length of tube covered with ice: 4.83 feet

Ethylene glycol rate: 196.2 pounds per hour

Temperature data: see Table #3 and Graph #8

Heat exchanger dimensions: see Figure #2

B. Heat Transfer Coefficient

Heat is transferred from the water vapor to the ethylene glycol solution. After the initial deposit of ice, the heat flow will pass through several resistances in series, namely h_o the ice/vapor interface, the ice layer, metal tube wall, and h_i the ethylene glycol surface film. In actual operation, the ice layer is continually increasing and the total system resistance increases accordingly. For simplification let us assume a stepwise condition where at the end of a time interval the total heat transfer may be calculated as follows:

1. Heat of sublimation, vapor$\rightarrow$ice:

Q = (lbs. ice/hour) (heat of sublimation)

= (0.0694) (1,219)

= 84.5 Btu/hr.

2. Average thickness of ice:

Avg. vol. ice/ft. tube = (lbs. ice/ft.)/(density of ice)

= (0.0357)/(57.5)

= 0.000625 cu. ft.

$$\text{Vol. ice ring} = \pi/4\ (D^2 - d^2)$$

$$0.000625 = \pi/4\ (D^2 - 0.0416^2)$$

D = 0.0504 ft.

D = 0.605 in.

3. Ethylene glycol surface coefficient:

Reynolds No. = DG/μ

D = .0357 ft.

G = (196.2 lbs./hr.)/(.001009 ft^2)

μ = (5.90 cp.) (2.42 lb./ft. hr./cp.)

Re No. = 488

From McAdams[5], the following empirical equation holds for laminar flow:

$$h_i = \frac{(1.62)\ (k)}{d} \left[\frac{(4)\ (W)\ (Cp)}{(\pi)\ (k)\ (L)} \right]^{1/3}$$

$k = .274$ Btu/hr/sq.ft./^oF/ft.

$d = .0358$ ft.

$W = 196.2$ lbs./hr.

$Cp = .857$ Btu/lb./^oF

$L = 4.83$ ft.

$h_i = 67.5$ Btu/hr. ft.^{2} ^oF

4. Finding Δ t's across each resistance:

a. Ethylene glycol:

$\Delta t = Q_i / (h_i)\ (A_i)$

$Q_i = 84.5$ Btu/hr.

$h_i = 67.5$ Btu/hr. ft.^{2} ^oF

$A_i = (0.1125)\ (4.83)$ sq. ft.

$\Delta t = 2.3$ ^oF

b. Metal wall:

$\Delta t = (q)\ (\Delta x)/(k)\ (A_m)$

$q = 84.5$ Btu/hr.

$\Delta x = .00291$ ft.

$k = 8.5$ Btu/hr./sq.ft./^oF/ft.

$A_m = (0.1217)\ (4.83)$ sq. ft.

$\Delta t = 0.0493$ ^oF

c. Ice film:

$\Delta t = (q)(\Delta x)/(k)(A_o)$

$q = 84.5$ Btu/hr.

$\Delta x = (.053''/12)$ ft.

$k = 1.3$ Btu/hr./sq. ft./^oF/ft.

$A_o = (.553/12)(\pi)(4.83)$ sq. ft.

$\Delta t = 0.411^oF$

d. Ice/vapor interface:

$\Delta t = t$ (avg. bulk vapor) - t (outside ice surface)

$= 46.7 - (19.6 + 2.76)$

$= 24.3\ ^oF$

5. Ice/vapor surface coefficient:

$h_o = Q/A_o\ \Delta t$

$Q = 84.5$ Btu/hr.

$A_o = (.605/12)(\pi)(4.83)$ sq. ft.

$\Delta t = 24.3\ ^oF$

$h_o = 4.54$ Btu/hr. ft.$^2\ ^oF$

6. U (total), based on inside surface of tube:

$1/U_i A_i = 1/h_i A_i + \Delta x\,(\text{tube})/k A_m +$

$\Delta x\,(\text{ice})/k A_m + 1/h_o A_o$

$1/U_i = 1/h_i + \Delta x \text{ (tube)}/(k)\ (d_m/d_i) + \Delta x \text{ (ice)}/\ (k)$
$(d_m/d_i) + 1/\ (h_o)\ (d_o/d_i)$

$= 1/67.5 + 0.0292/\ (8.5)\ (.465/.43) +$
$0.0441/\ (1.3)\ (.553)/(.43) + 1/(4.54)(.605/.43)$

$= 0.0148 + 0.000318 + 0.00264 + 0.1565$

$1/U_i = 0.1746$

$U_i = 5.73$ Btu/hr. ft.^{2} $^{\circ}$F

7. U_i , check calculation

Δt (overall) = t (bulk vapor) - t (bulk glycol)

$= 47.6 - 19.6$

$= 27.1\ ^{\circ}$F

$Q = U_i\ A_i\ \Delta t$

$U_i = Q/A_i\ \Delta t$

$= (84.5)/(.1125)\ (4.83)\ (27.1)$

$= 5.66$ Btu/hr. ft.^{2} $^{\circ}$F

8. Avg. U (total)

U (total) = $(5.73) + (5.66)/2$

$= 5.70$ Btu/hr. ft.^{2} $^{\circ}$F

C. <u>Wilson Plot Technique</u>

Individual surface coefficients are difficult to obtain from direct experimentation, but they may be found by difference through the use of a Wilson plot.

For a tube with an ice film:

$$\frac{1}{U} \text{ total} = R(\text{fluid}) + R(\text{tube}) + R(\text{ice}) + R(\text{vapor/ice})$$

Wilson has shown that the sum of the first three resistances is approximately constant. Since the water side resistance is an inverse function of the water velocity through the tube, and neglecting changes in temperature due to increased flow, the water-side resistance could be taken as a function of the water velocity alone. If a plot of $1/U$ vs $1/V^{\frac{1}{3}}$ is plotted on rectangular coordinates, a water resistance line is plotted through the origin. $1/U$ was obtained using an empirical equation for laminar flow:

$$h_i = \frac{(1.62)\,(k)}{d} \left[\frac{(4)\,(W)\,(Cp)}{(\pi)\,(k)\,(L)}\right]^{1/3} \qquad (1)$$

A line parallel to this at the operating condition of $1/U$ and $1/V^{\frac{1}{3}}$ will give an intercept on the vertical axis which is the sum of R (tube) + R (ice) +(vapor/ ice film)at infinite water velocity. From this intercept is subtracted R (tube) And R (ice). The remainder is the vapor ice film coefficient.

1. Establishing a water resistance line:

a. Assume a ethylene glycol velocity of 1 ft./sec.

through the heat exchanger tube. The flow rate will be:

$$\text{lbs./hr.} = (.001009 \text{ sq. ft.})\ (1 \text{ ft/sec.})\ (3600 \text{ sec./hr.})$$
$$= (65.8 \text{ lbs./cu.ft.})$$
$$= 239 \text{ lbs./hr.}$$

b. Determine h_i from formula (1):

$$h_i = \frac{(1.62)\ (.274)}{.0358} \left[\frac{(4)\ (239)\ (.859)}{(\pi)\ (.274)\ (4.83)} \right]^{\frac{1}{3}}$$

$$= 72.1$$

$$\frac{1}{h_i} = \frac{1}{U_i} = 0.01386$$

c. Summary of points for Wilson plot:

Velocity	$1/V^{\frac{1}{3}}$	$1/U_i$
1.0 (assumed)	1.0	.01386
0.22 "	1.66	.0229
2.0 "	0.794	.0110
0.82 (actual)	1.07	.1755

d. On rectangular coordinates, plot the water resistance line using data from (c). Draw the best line through the points to the origin.

e. Plot the run point, draw a line parallel to the water resistance line. The intercept on the vertical axis is the sum of all resistances at infinite water velocity. We know the resistance of the metal wall and of the ice film. These are subtracted from the intercept value, the remainder will be the value of the ice/vapor interface:

vertical intercept = .1608

metal resistance = .00032

ice resistance = .00264

ice/vapor resistance = .1608 - (.00032 + .00264)

= .1550

$h_{o,i}$ = 1/.1550 6.45 Btu/hr. ft.^{2} ^{o}F

(at infinite velocity)

TABLE #1 SYSTEM DATA

Run No.	1	2	3
system vacuum, mm. Hg	1.3	1.5	2.0
time of run, hrs.	2.5	2.25	3.17
wgt. water charged, gms.	150.0	150.0	150.0
wgt. water recovered, gms.	60.5	57.9	61.2
wgt. ice formed, gms.	78.9	79.9	71.0
wgt. cold trap ice gms.	8.5	5.8	8.1
system losses, gms.	2.2	6.5	4.7
length of ice cover, ft.	4.83	4.5	3.0
heat of ice formation, Btu/hr.	84.5	95.1	75.0
eth. glycol flow rate, lbs/hr.	196.2	129.0	111.0

TABLE #1 SYSTEM DATA

Run No.	4	5	6
system vacuum, mm. Hg	2.5	4.0	atmos. press.
time of run, hrs.	4.17	3.0	2.5
wgt. water charged, gms.	163.5	150.0	atmos. press.
wgt. water recovered, gms.	70.0	113.8	-
wgt. ice formed, gms.	81.0	30.7	16.7
wgt. cold trap ice, gms.	6.3	3.7	-
system losses, gms.	6.2	1.8	-
length of ice cover, ft.	2.0	1.21	5.0
heat of ice formation, Btu/hr.	52.0	27.5	44.7
eth. glycol flow rate, lbs./hr.	196.2	196.2	196.2

TABLE 2

EXPERIMENTAL RESULTS

Run No.	1	2	3
system vacuum, mm. Hg	1.3	1.5	2.0
avg. thickness of ice, in.	0.053	0.057	0.070
length of tube ice cover, ft.	4.83	4.5	3.0
wgt. ice formed, gms.	78.8	79.9	71.0
Δt, bulk eth. glyc. to bulk vapor, ^oF	27.1	27.9	27.6
Δt, bulk eth. glyc. to surf. of ice, ^oF	2.76	3.61	4.24
Reynolds no. eth. glyc. flow	488	320	275
Reynolds no. annulus, approx.	4.9	5.5	3.5
eth. glyc. surf. coeff., h_i	67.5	60.1	65.5
ice/vapor surf. coeff., $h_{o,i}$	6.39	7.74	9.49
overall heat trans. coeff., U_i	5.70	6.72	8.04
ice/vap. surf. coeff. Wilson plot, $h_{o,i}$	6.45	7.94	9.47

TABLE 2

EXPERIMENTAL RESULTS

Run No.	4	5	6
system vacuum, mm. Hg	2.5	4.0	atmosph. pressure
avg. thickness of ice, in.	0.125	0.078	0.019
length of tube ice cover, ft.	2.0	1.21	5.0
wgt. ice formed, gms.	81.0	30.7	16.7
Δt, bulk eth glyc. to bulk vapor, ^{o}F	26.8	26.1	49.2
Δt, bulk eth. glyc. to surf. of ice, ^{o}F	3.77	2.70	0.64
Reynolds no. eth. glyc. flow	488	488	488
Reynolds no. annulus, approx.	3.1	1.6	nat. conv.
eth. glyc. surf. coeff., h_i	90.6	107.3	66.7
ice/vapor surf. coeff., $h_{o,i}$	10.06	8.64	0.649
overall heat trans. coeff., U_i	8.61	7.73	0.641
ice/vapor.surf. coeff. Wilson plot, $h_{o,i}$	10.3	8.83	0.662

EXPERIMENTAL PROCEDURE

A. Preparation

Air leakage should be kept to a minimum for best results. It was essential that the following steps were taken in setting up the experiment:

1. The ends of the Lucite tube had to be ground flat and square.

2. Fresh rubber gaskets were provided for each run.

3. The apparatus was checked for air leaks before any water was added to the system. First a vacuum test was used. If leaks could not be eliminated, then a soap test with 5 to 10 pounds air pressure was made to detect leaks.

B. Determining the System Heat Gain

The annular space was evacualed and the cold ethylene glycol was passed through the heat exchanger to establish the heat gain of the system. This was done for a period of 60 to 90 minutes to cool down the end flanges and other parts of the system. Thermocouple readings were taken 12 to 15 minutes apart to record the heat pick-up by the glycol. This was usually in the range of 0.25^{o}F to 0.70^{o}F depending on the glycol flow and room temperature. This procedure was followed just prior to adding water to the system. System heat gains are shown in Table 3.

C. Developing the Ice Film

After equilibrium had been reached, water was added from the dropping funnel into the 250 ml. flask in a rapid, but dropwise manner until 50 grams had been added. The flask had previously been scribed to indicate the 50 gram level. Additional water was added dropwise so that the 50 gram level was maintained as closely as possible. This level was checked every 12 to 15 minutes and adjustments were made in the water addition. The graduated dropping funnel was read periodically to record an approximate water addition rate. This data is summarized in Table 4.

The chiller was thermostatically controlled with a cycle range from 18° to $20^{\circ}F$. Therefore, it was the practice to take thermocouple readings at the mid-point of the temperature cycle. This established the 12 to 15 minute frequency of system readings. Ethylene glycol flow rates and vacuum conditions were varied to develop data for analysis. These are summarized in Table 1. Runs lasted from 2.25 to 4.17 hours. They were continued until 70 to 80 grams of ice formed on the tube. At the conclusion of a run the end flange and Lucite tube were removed and the ice film was chipped or slid off the metal tube very carefully and weighed. Likewise, a record was kept of the cold trap ice and water remaining in the flask. Aluminum foil and paper towels previously tared were spread under the metal heat exchanger tube to catch any drops of water falling from the melting ice. Water balances showed that the average water losses as 4.3%. The information collected permitted the calculation of heat transfer coefficients. Experimental results are given in Table 2.

D. Accuracy of the Data

All thermocouples were standardized at 32°F in a ice bath and checked with an A.S.T.M. thermometer. This data was obtained from 27 sets of thermocouple readings and is summarized in Table 5. The standard deviation of the potentiometer readings is 2.5 millivolts. It can be estimated that 66% of all readings were within ± 2.5 millivolts and 95% of the readings were within ± 5 millivolts. In terms of temperature 66% of the readings are accurate to ± 0.1°F and 95% of the readings are included in a ± 0.2°F spread.

The Zimmerli vacuum gage was new and filled with clean mercury as instructed by the manufacturer. The manufacturer's data sheet claims ± 0.1 mm. Hg accuracy.

The water balances made on the runs showed a water recovery of 95.7%.

DESCRIPTION OF APPARATUS

The flow sheet in Figure 1 illustrates the equipment used in the investigation. The 6'-0" long heat exchanger was constructed of a ½" o.d., 304 stainless steel inner tube mounted concentrically within a 3" o.d. Lucite tube. The general experimental procedure was to make measurements on the rate of heat transfer between refrigerated ethylene glycol flowing through the inner tube and water vapor in the annulus. The heat exchanger assembly is shown in Figure 2 and details in Figure 3. Ends of the annular space were closed off with 6" o.d., 304 stainless steel flanges. The inner tube was welded to one flange and slipped through a 3/4" nipple which was welded to the other flange. A rubber hose seal as shown in Figure 3 was used to make the end connection. The flanges were also machined with a circular groove which received the Lucite tube. A 1/8" thick white neoprene gasket was used between the flange and Lucite tube. The whole unit was held together with 3/8" o.d. tie rods.

This design permitted the unit to be taken apart for removal and weighing of the ice films which collected on the surface of the inner tube, and at the same time gave a relatively air tight heat exchanger. A centrifugal pump circulated the ethylene glycol through a chiller unit and the heat exchanger. The flow was metered through a calibrated rotameter. A calibration curve is shown on Graph 7. Water vapor was generated in a 250 c.c. flask fitted with a dropping funnel and heating mantle. Rubber vacuum hose carried the vapor to the annular space of the heat exchanger. The vacuum pump which was connected to the other end of the annular space helped pull the water vapor through the exchanger. A cold trap made with a vacuum filtering flask in a crushed dry ice bath trap- any water vapor which did not freeze out in the heat exchanger.

Iron-constantan thermocouples were located as shown in Figure 2. Two thermocouples were installed at each measuring point and the average of the readings were used in calculations. Millivolt readings were read by a poten- tiometer and conversions to temperature were made using the National Bureau of Standards Thermocouple Tables as published in circular 561.

Major equipment specifications are as follows:

Heat Exchanger

Inner Tube:	Material	- 304 stainless steel
	Length	- 6'-0"
	O.D.	- 0.500"
	I.D.	- 0.430"

Jacket Tube:	Material	- Lucite clear acrylic
	Length	- 6'-0"
	O.D.	- 3.00"
	I.D.	- 2.50"

Liquid Chiller

Chiller model: Worthington LCH-10

Compressor model: 23JF5-10

Nominal tonnage: 9.4

Refrigerant: R-22

Vacuum Pump

Model: Welch Scientific Co. Disto-Pump Model 1399

Motor: 1/3 H.P., 115 volts, 1725 R.P.M.

Capacity: 32 liters per minute free air

Vacuum Gage

Model: Zimmerli No. JV-3330

Range: 0 to 100 mm. Hg in 1 mm.graduations

Accuracy: - 0.1 mm. Hg

Potentiometer

Model: Leeds & Northrup Co. Catalog #8686

Range: 10.1 mv. to + 100.1 mv.

Limits of error: - (0.05% of reading + 3 v)

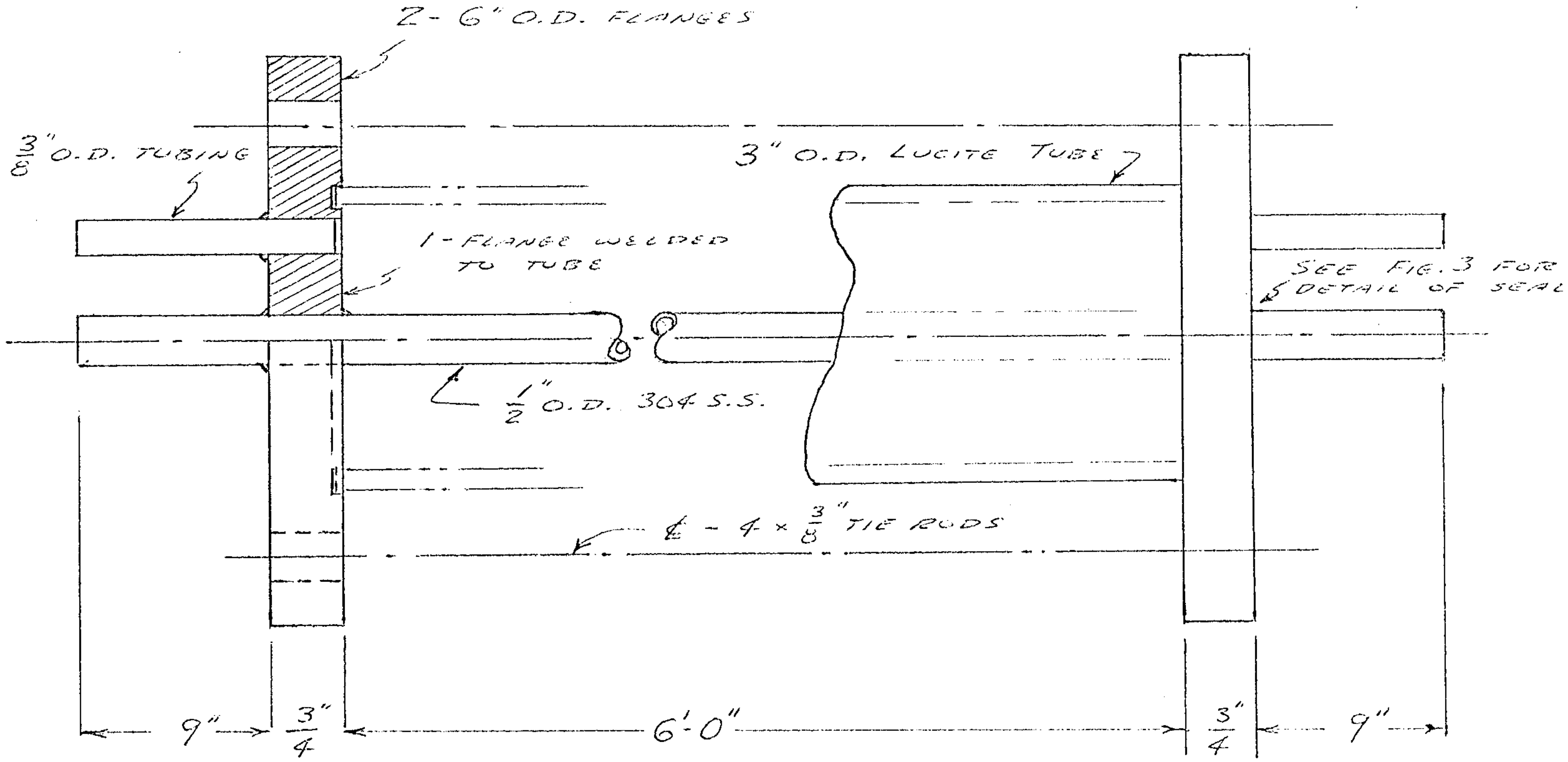


FIGURE 2

HEAT EXCHANGER ASSEMBLY

FIGURE 3
HEAT EXCHANGER DETAIL

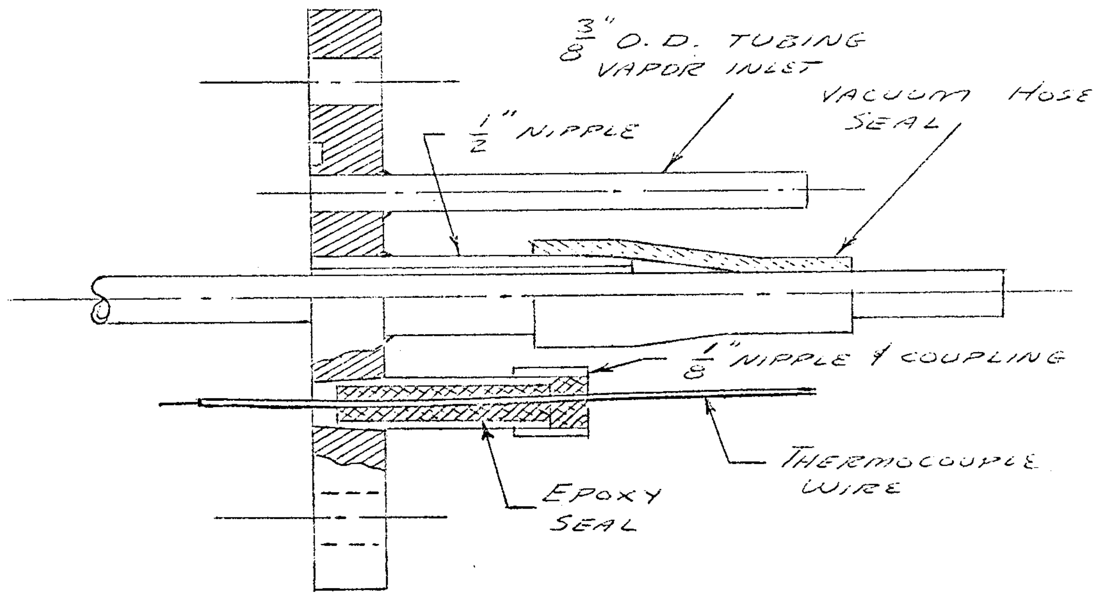


FIGURE 4
ICE COATINGS DEVELOPED

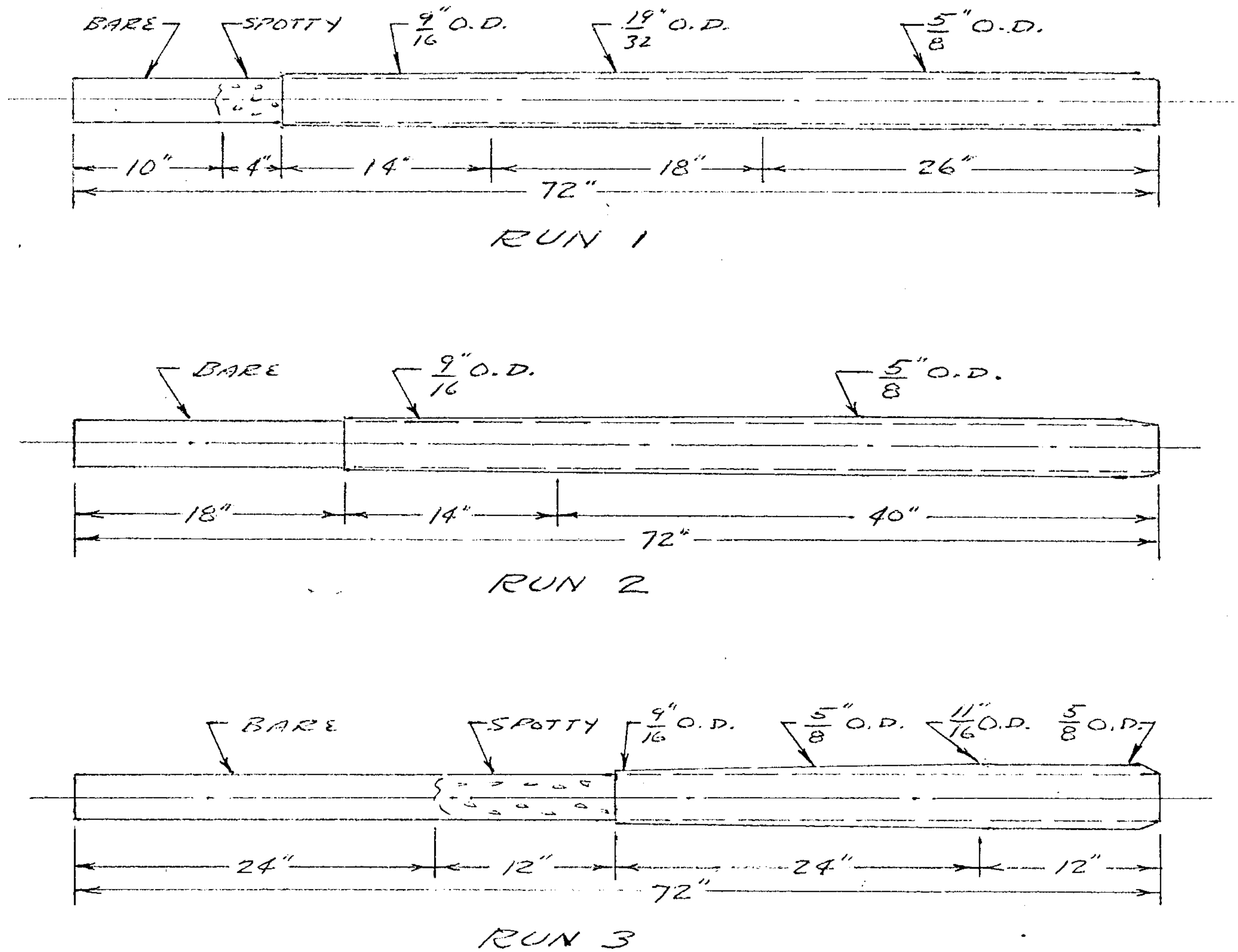
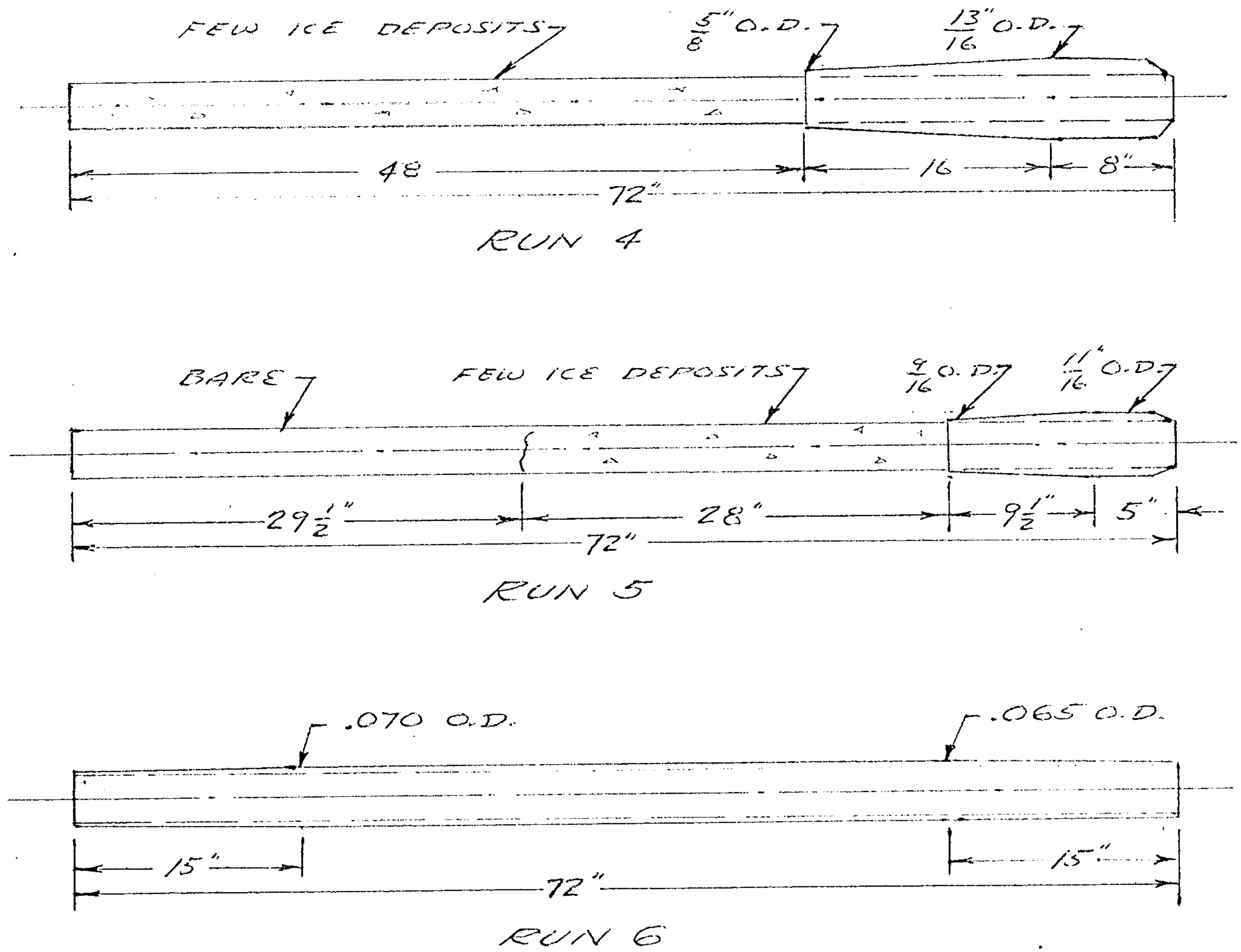


FIGURE 5
ICE COATNGS DEVELOPED



DISCUSSION

One of the most interesting points of the experiment was the physical manner in which the water vapor froze out on the walls of the metal tube. The initial condensation was composed of small, crystal clear droplets of ice. It appeared as a very fine dropwise type of condensation except that these were ice droplets instead of liquid condensate. Most were about 0.2" to 0.3" in diameter. In 15 to 30 minutes these particles lost their physical boundaries as additional water froze out of the vapor phase. A thin sheet of ice then formed. With experiments of 1.3 and 1.5 mm. pressure, 80% of the tube surface was covered with the ice film, whereas in runs at 2, 2.5 and 4 mm. the percentage of tube coated with ice was progressively smaller as more air was mixed with the water vapor. At 4 mm. system pressure, only 33% of the tube surface was ice covered. It was observed that the area of ice coverage was established very early in the experiment. As time went past the first 30 minutes the film grew thicker but did not spread over 100% of the tube surface which was available. It appeared that a diffusion process was occuring with the air forming an insulating blanket around the bare metal surface.

A human element was involved in the rate of water vapor generation from the 250 c.c. flask in that the operator had to judge what was the maximum allowable rate at which the vapor could be admitted to the heat exchanger. A few early trials showed that the ice film would melt if the system was pushed by too rapid boiling of the water in the flask. To avoid destruction of the ice film, the vapor temperature was kept to a maximum of approximately 48°F.

Another point which may be of general interest occured during the thermocouple standardization. It was found that changes in room temperature had an effect on thermocouple readings to the extent that a secondary junction was formed at a rotary switch terminal strip. Since the terminal strip was made of brass or similiar metal, the connections with the iron and constantan thermocouple leads created a secondary junction under conditions of changing room temperature. This was overcome by insulating the terminal strip area with 1" fiberglas blanket insulation which greatly improved the accuracy of thermocouple data. The insulation corrected a swing of approximately 1°F when measuring temperatures at 32°F.

CONCLUSION

1. In a reverse-sublimation system of water vapor, exclusion of all air results in improved heat transfer surface coefficients.

2. In the above system, the ice/vapor coefficient was the controlling resistance to heat transfer.

3. Frost was not seen at any time during vacuum runs. The films developed were clear ice.

4. Ice coatings above 2.0 mm. Hg were not as evenly distributed as were runs done at 1.3 mm. and 1.5 mm. Hg pressure. It is thought that enough air is present to involve the mechanics of diffusion of water vapor through air.

RECOMMENDATIONS

1. For effective heat and mass transfer rates when freezing out water vapor, it is essential to reduce air leakage to a minimum. The apparatus used in this experiment was not perfectly air tight. Vacuums of 1.3 mm Hg were obtained. It is believed that at an absolute vacuum, heat and mass transfer rates would be improved considerably.

2. Control of the flow of water vapor to the condensing or "freeze-out" section was done manually. In future work, some thought should be given to improve this operation.

3. In reading the literature, heat transfer rates over frost surfaces improved as the Reynolds number of the vapor phase was raised. It is suggested that the combination of 100% air removal plus circulation of the vapor to be condensed will improve surface heat transfer coefficients.

NOMENCLATURE

A	Heat flow area, ft.^{2}
Cp	Specific heat, Btu/lb. ^oF
D	Diameter of pipe or tube, ft.
G	Mass velocity, lb./(hr.)
h_i	Heat transfer coefficient, inside pipe surface, Btu/hr.)(ft.^{2}) (^oF)
$h_{o,i}$	Heat transfer coefficient, ice/vapor interface, based on inside pipe surface, Btu/(hr.)(ft.^{2})(^oF)
k	Thermal conductivity, Btu/(hr.)(ft.^{2})(^oF/ft.)
L	Length, ft.
Q	Heat flow, Btu/hr.
q	Heat flow, Btu/(hr.)(lin. ft.)
R	Thermal resistance, (hr.) (ft.^{2}) (^oF)/Btu
Re	Reynolds number, dimensionless
t	Temperature, ^oF
U	Overall coefficient of heat transfer, Btu/(hr.)(ft.^{2}) (^oF)
V	Velocity, ft./min.
W	Weight flow of fluid, lb./hr.
Δx	Thickness of cross-section, ft.
μ	Viscosity, lb./(ft.)(hr.)
λ	Latent heat, Btu/lb.

APPENDIX

	Page
System Heat Gains	37
Water Evaporation Rate	38
Thermocouple Calibration	39
Wilson Plots, all Runs	40
Rotameter Calibration	46
Temperature Data, all Runs	47
Thermal Conductivity of Frost	53
Physical Properties Data, Ethylene Glycol	54
Liquid Chiller	58
References	59

TABLE 3

SYSTEM HEAT GAINS

Run No.	Room Temp. °F	Flow of Eth. Glycol, lbs./hr.	Δt_{gain} °F
1	62.0	196.2	0.25
2	72.0	129.0	0.40
3	63.0	111.0	0.60
4	67.0	196.2	0.60
5	73.0	196.2	0.50
6	68.2	196.2	0.60

The system heat gain in each case was determined over a period of 60 to 90 minutes under vacuum at the respective flow rates without water vapor in the annulus.

TABLE 4

WATER EVAPORATION RATE

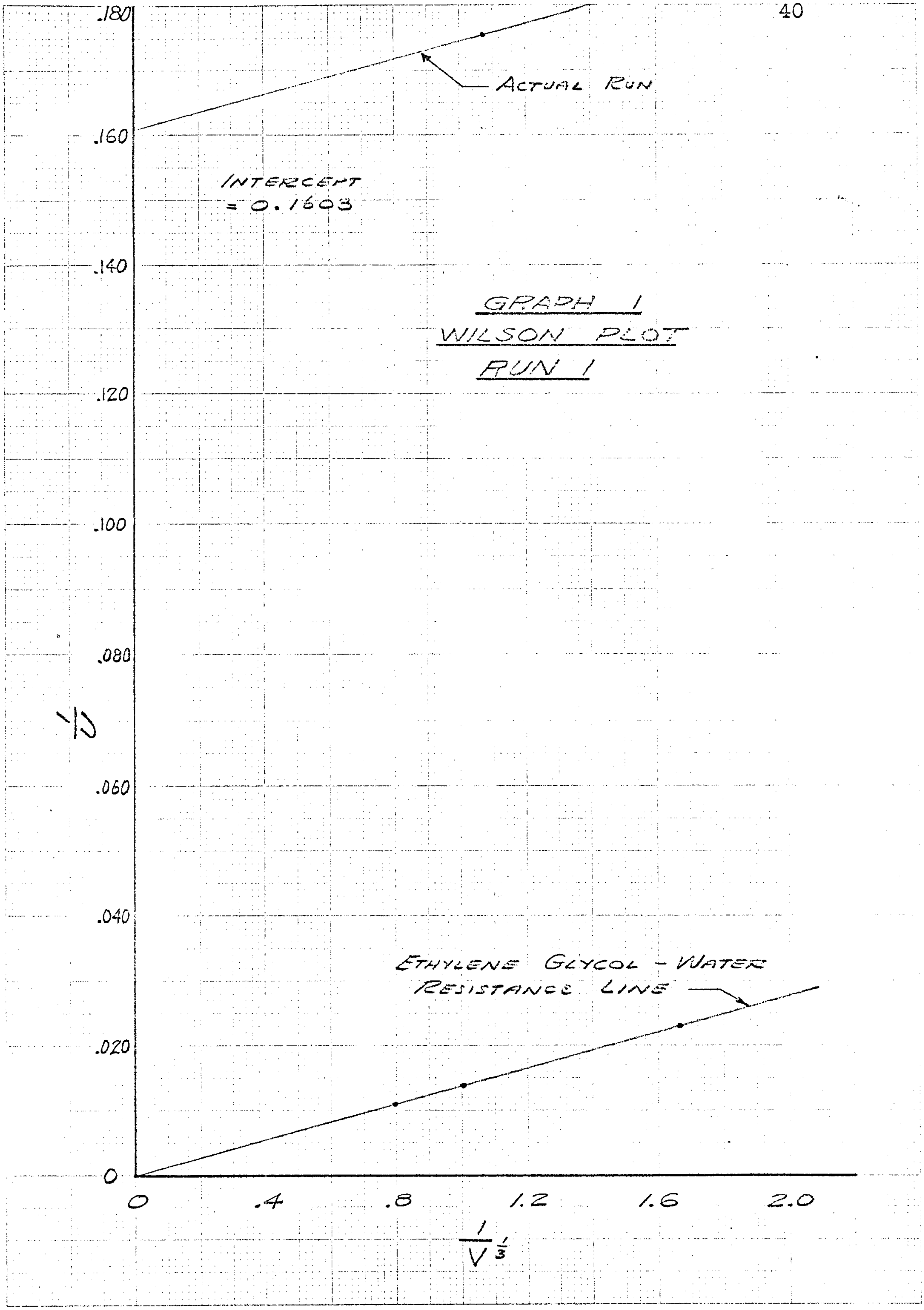
Run No.	Approximate Gms. Water Evaporated 1 hr.	2 hrs.	End of Run
1	37	70	78.8 (2.5 hrs.)
2	38	70	79.9 (2.25 hrs.)
3	20	50	76.0 (3.17 hrs.)
4	25	42	81.0 (4.17 hrs.)
5	not rec.	not rec.	30.7 (3.0 hrs.)
6	in air	in air	16.7 (2.5 hrs.)

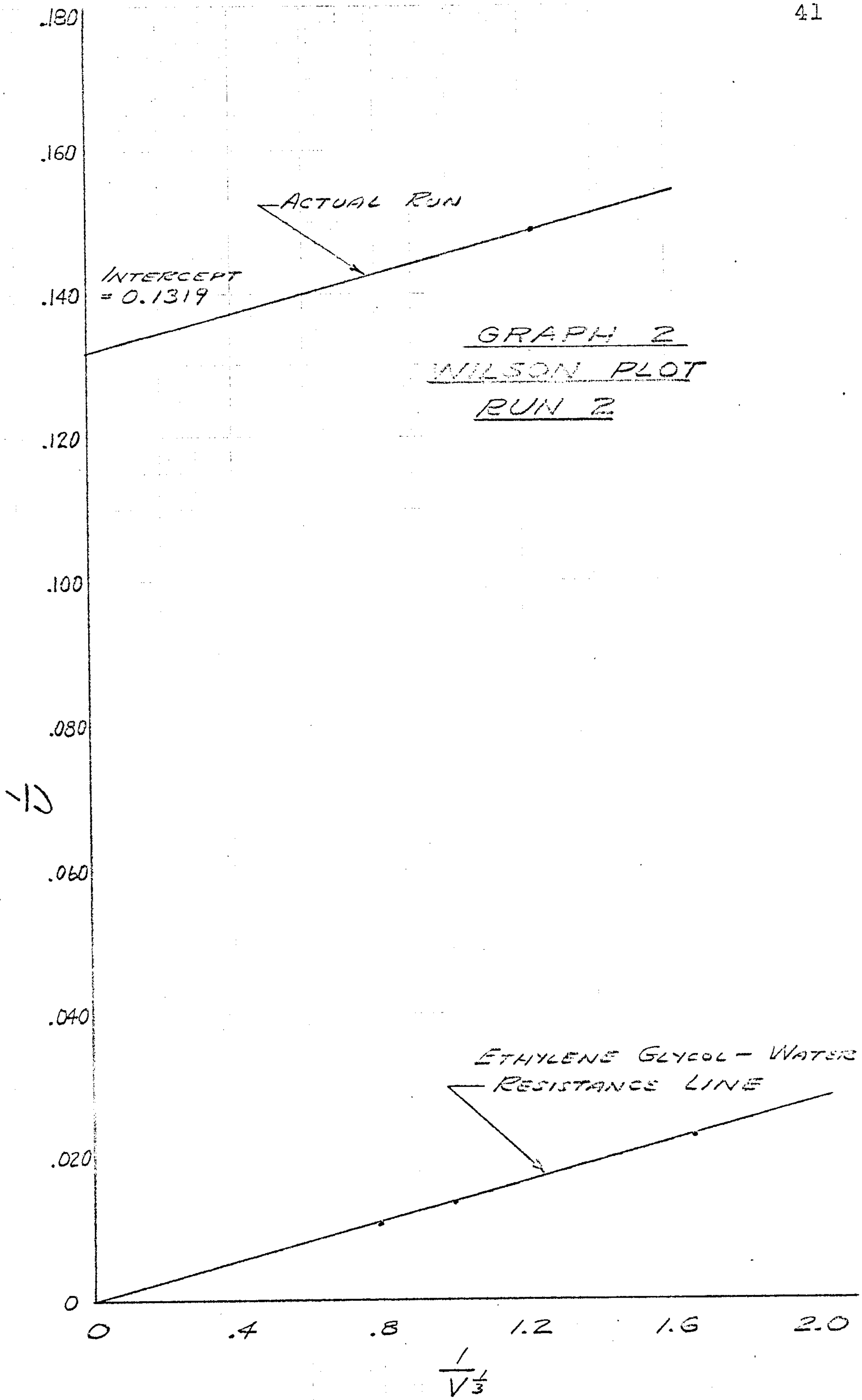
TABLE 5

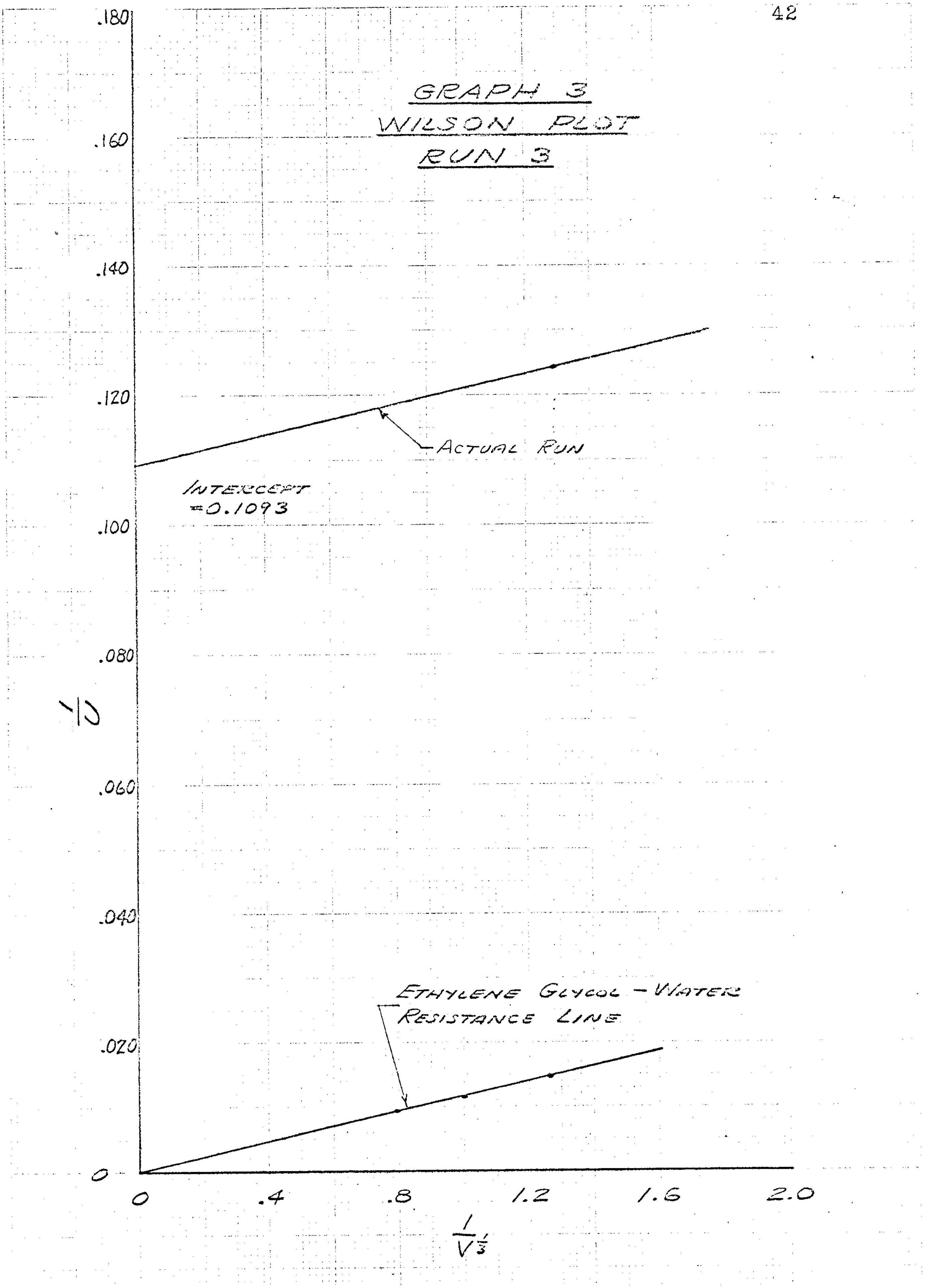
THERMOCOUPLE CALIBRATION

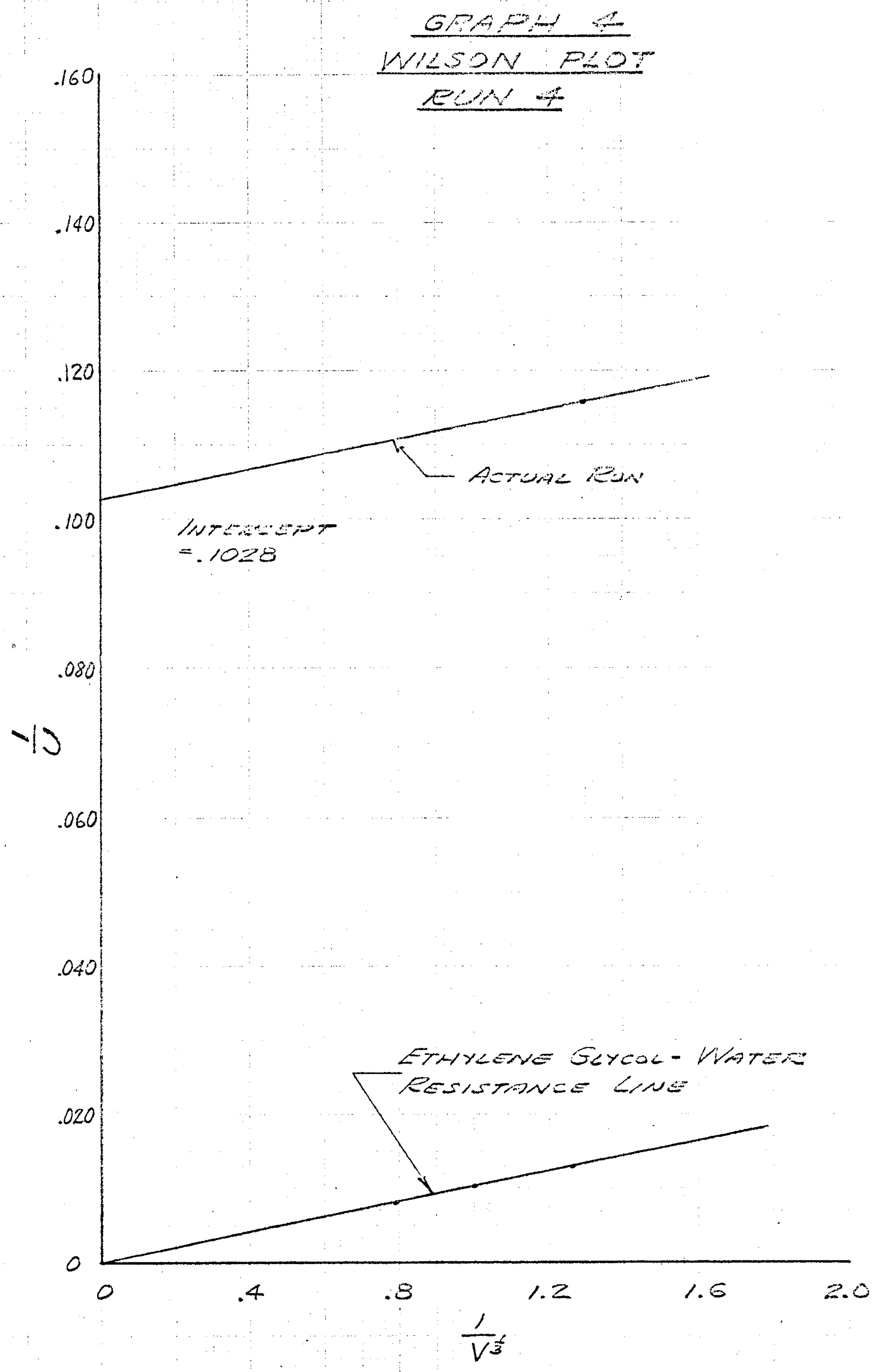
THERMOCOUPLE NO.	CORRECTION, mv. (add to T.C. reading)
0	+0.036 mv.
1	+0.035 mv.
2	+0.033 mv.
3	+0.029 mv.
4	+0.028 mv.
5	+0.024 mv.
7	+0.018 mv.
8	+0.019 mv.
9	+0.016 mv.
10	+0.012 mv.

The above calibrations were done in an ice bath and checked at the ice point with a thermometer calibrated according to the A.S.T.M. This data represents the mean average of 27 trials.


ACTUAL RUN
INTERCEPT
= 0.1603
GRAPH 1
WILSON PLOT
RUN 1
ETHYLENE GLYCOL - WATER
RESISTANCE LINE
.180
.160
.140
.120
.100
.080
.060
.040
.020
0
1/U
0
.4
.8
1.2
1.6
2.0
1/V^{1/3}


.180
.160
.140
.120
.100
.080
.060
.040
.020
0
ACTUAL RUN
INTERCEPT = 0.1319
GRAPH 2
WILSON PLOT
RUN 2
ETHYLENE GLYCOL – WATER RESISTANCE LINE
0
.4
.8
1.2
1.6
2.0
$\frac{1}{V^{\frac{1}{3}}}$

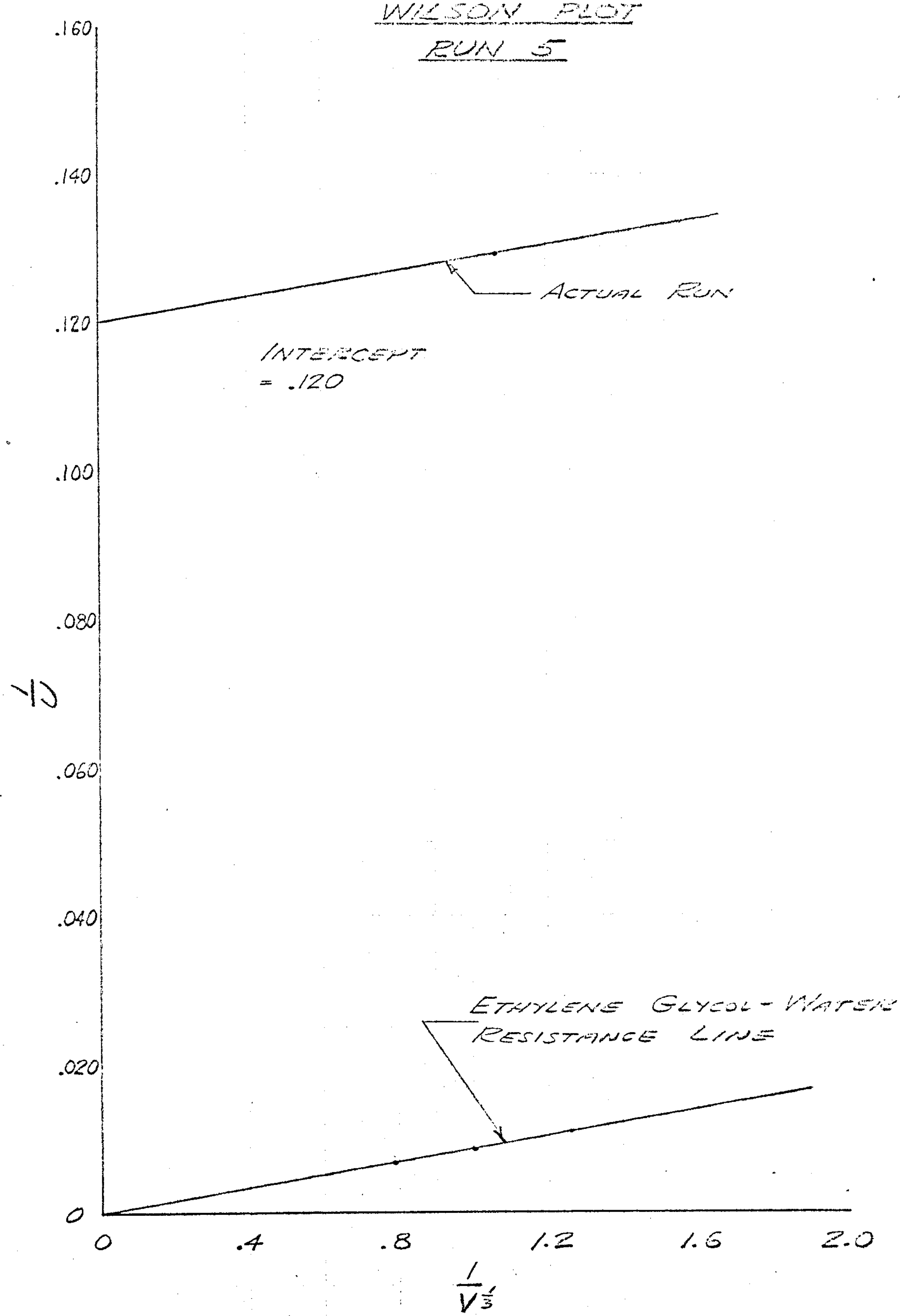

GRAPH 3
WILSON PLOT
RUN 3
ACTUAL RUN
INTERCEPT =0.1093
ETHYLENE GLYCOL - WATER RESISTANCE LINE
1/U
1/V^(1/3)
.180
.160
.140
.120
.100
.080
.060
.040
.020
0
0
.4
.8
1.2
1.6
2.0

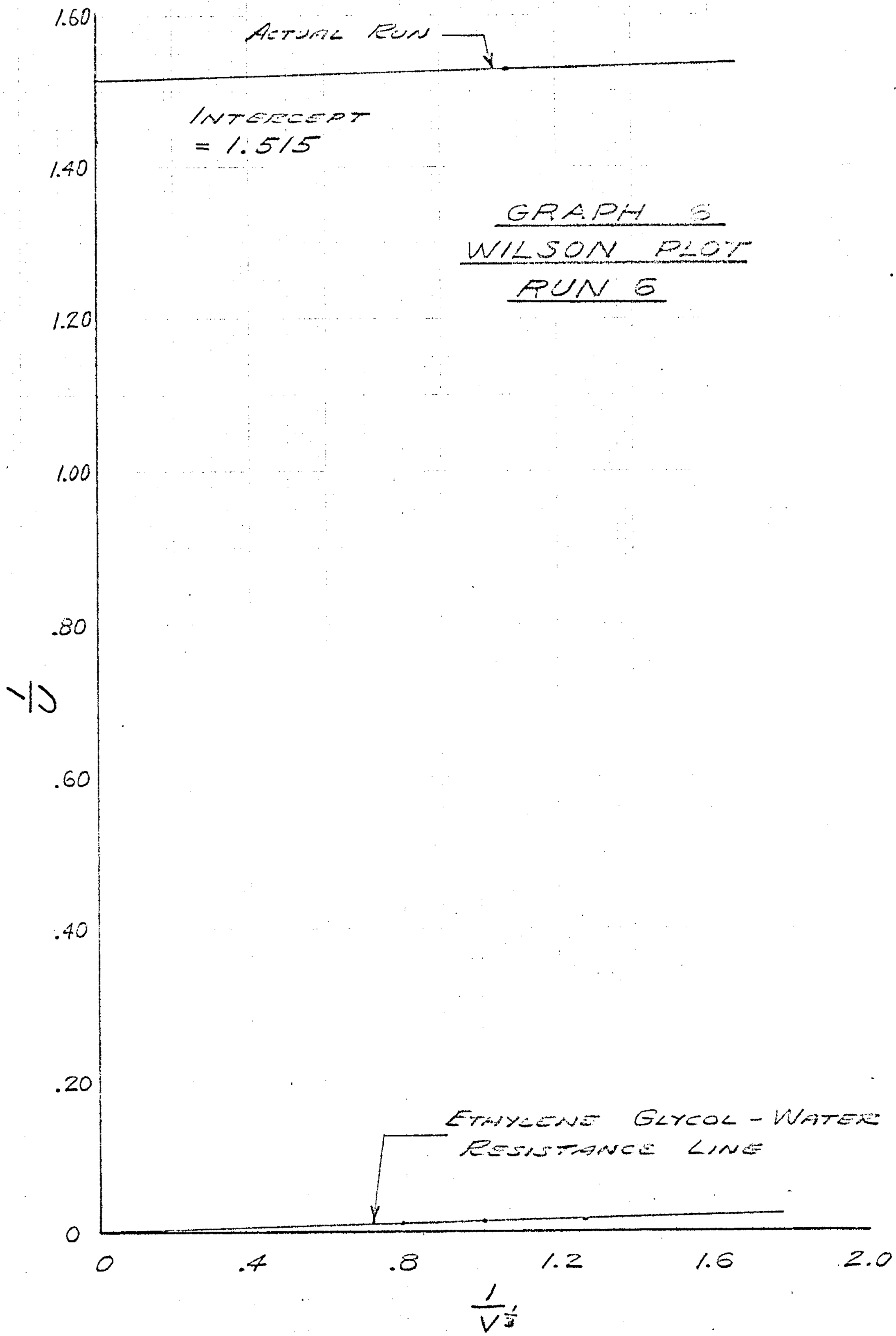

GRAPH 4
WILSON PLOT
RUN 4
.160
.140
.120
.100
.080
.060
.040
.020
0
$\frac{1}{U}$
ACTUAL RUN
INTERCEPT
=.1028
ETHYLENE GLYCOL- WATER
RESISTANCE LINE
0
.4
.8
1.2
1.6
2.0
$\frac{1}{V^{\frac{1}{3}}}$

GRAPH 5

WILSON PLOT

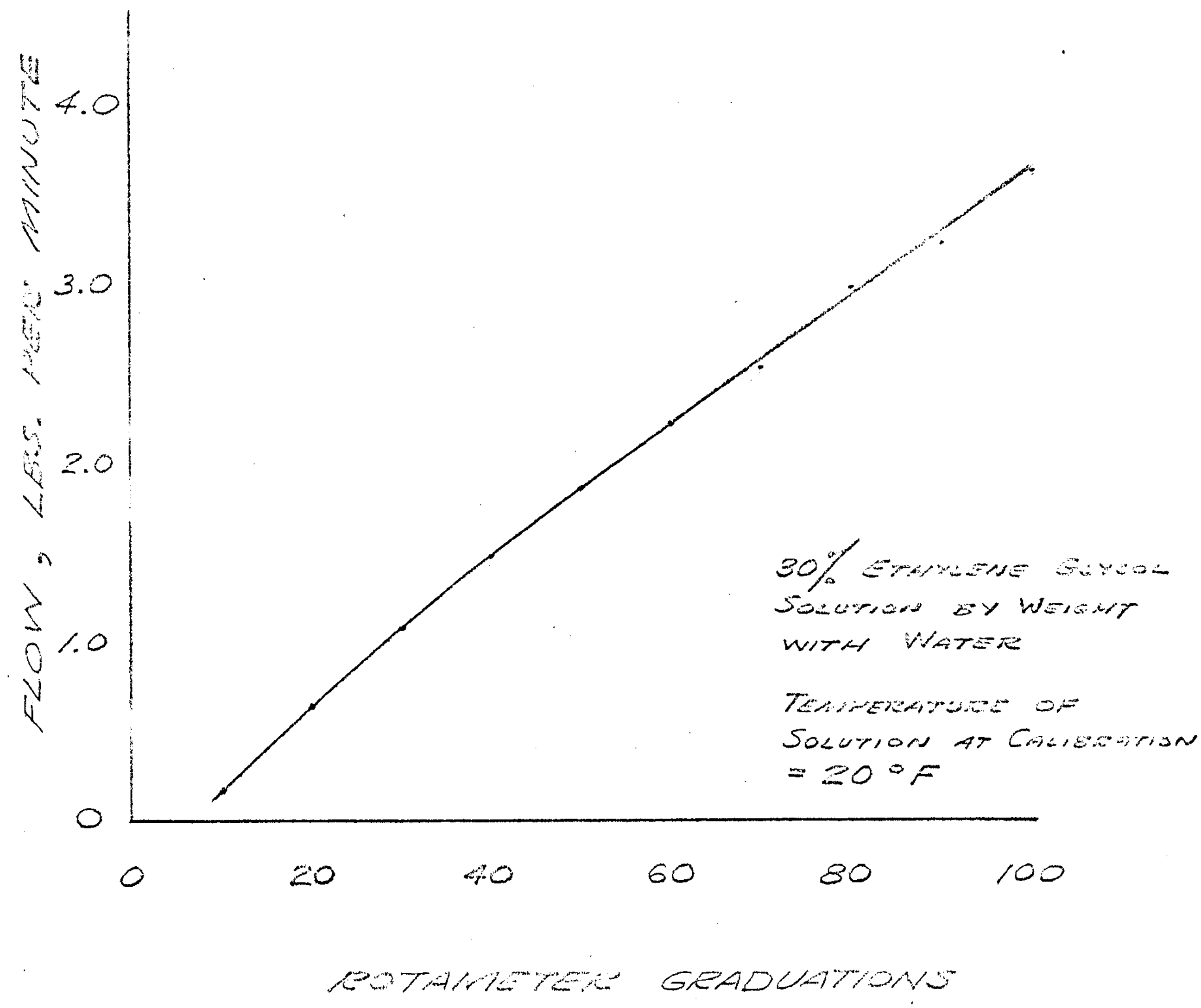
RUN 5



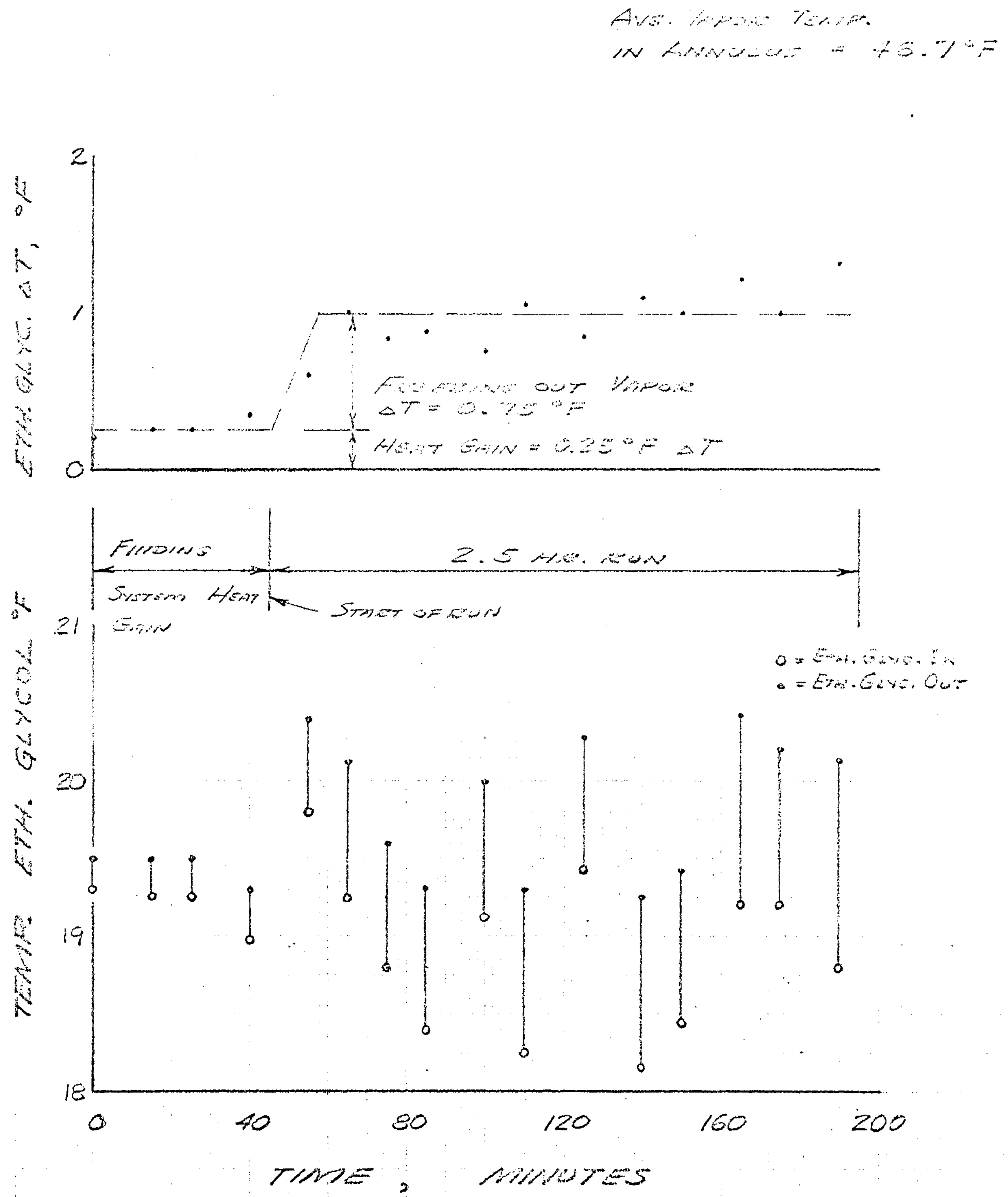

ACTUAL RUN
INTERCEPT
= 1.515
GRAPH 6
WILSON PLOT
RUN 6
1.60
1.40
1.20
1.00
.80
.60
.40
.20
0
1/U
ETHYLENE GLYCOL - WATER
RESISTANCE LINE
0
.4
.8
1.2
1.6
2.0
1/V^(1/3)

GRAPH # 7

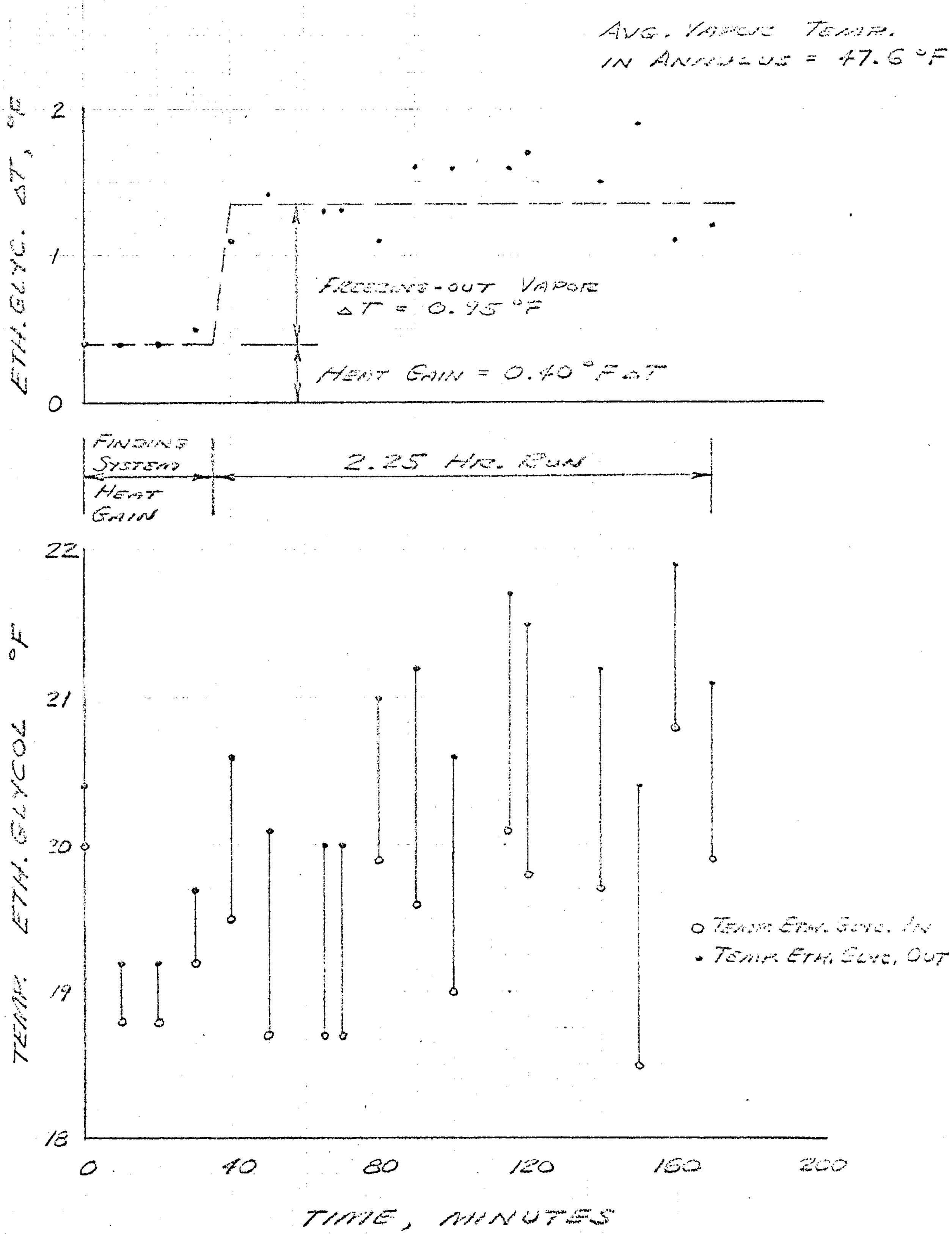
ROTAMETER CALIBRATION



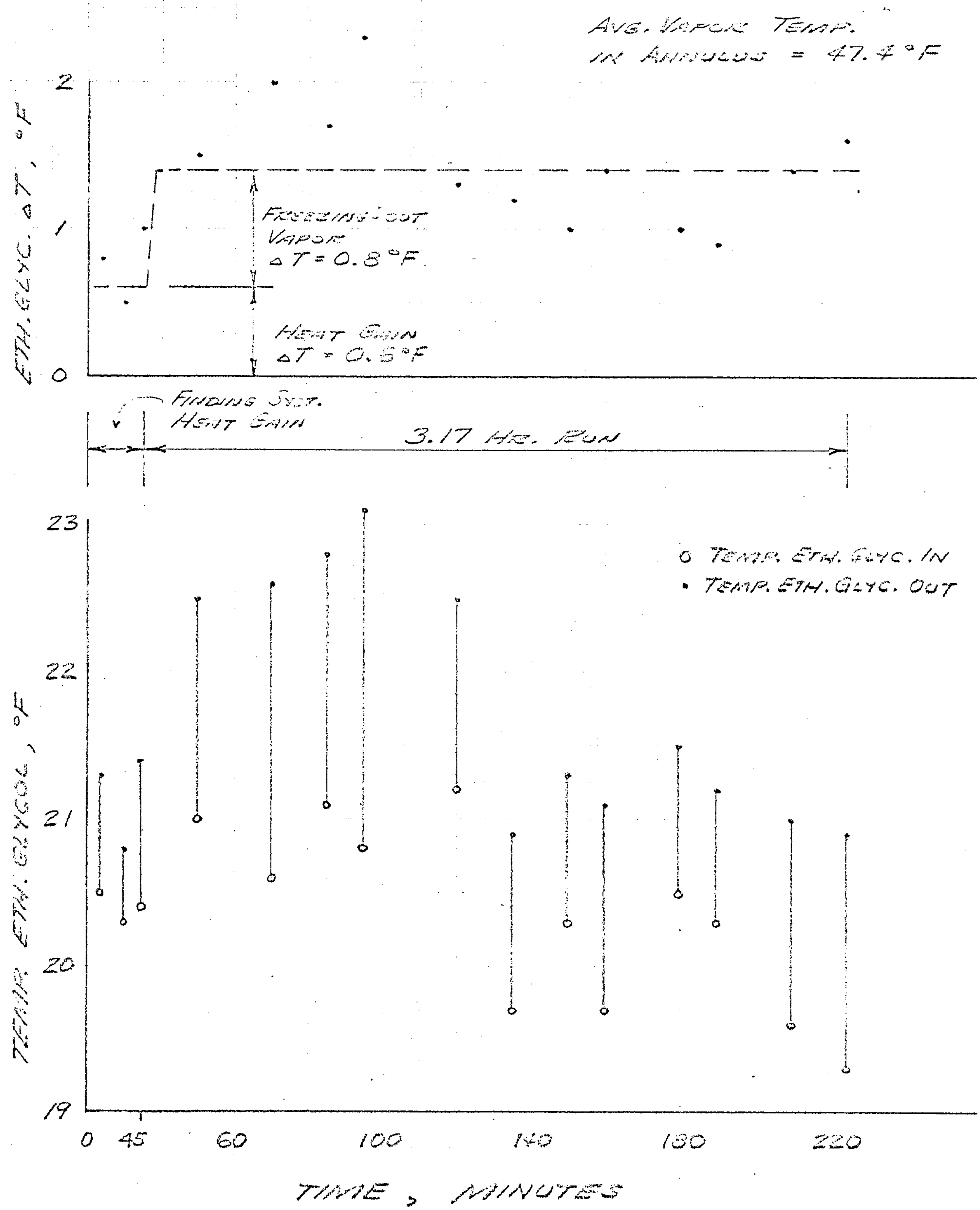
GRAPH 8
THERMOCOUPLE DATA
RUN 1



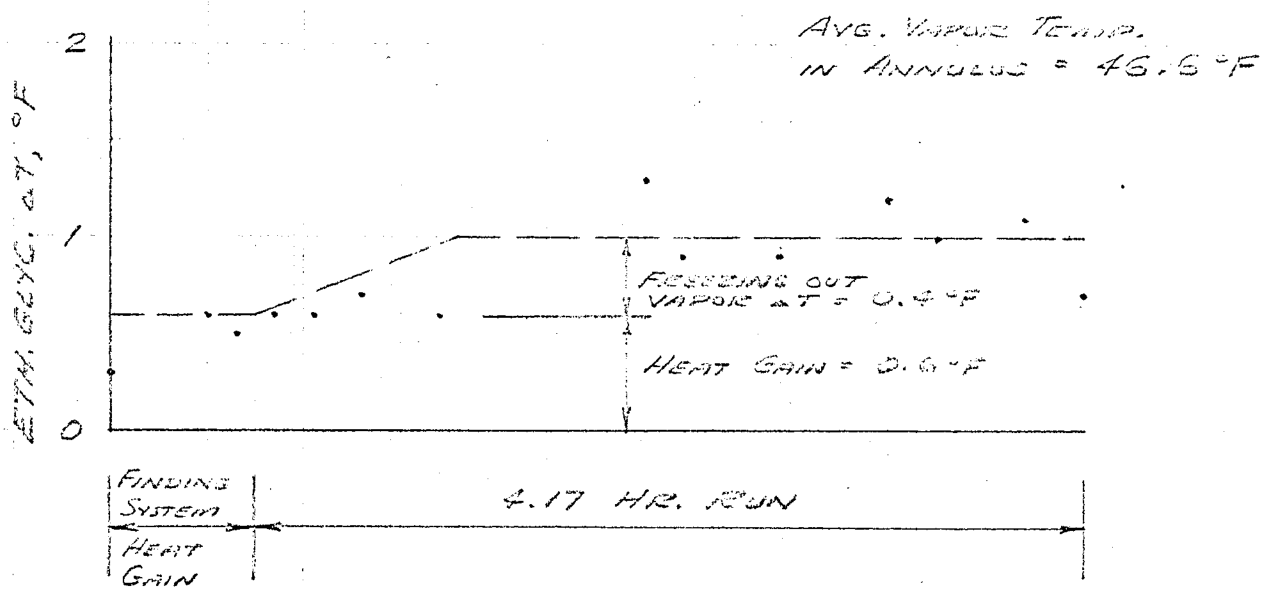
GRAPH 9
THERMOCOUPLE DATA
RUN 2

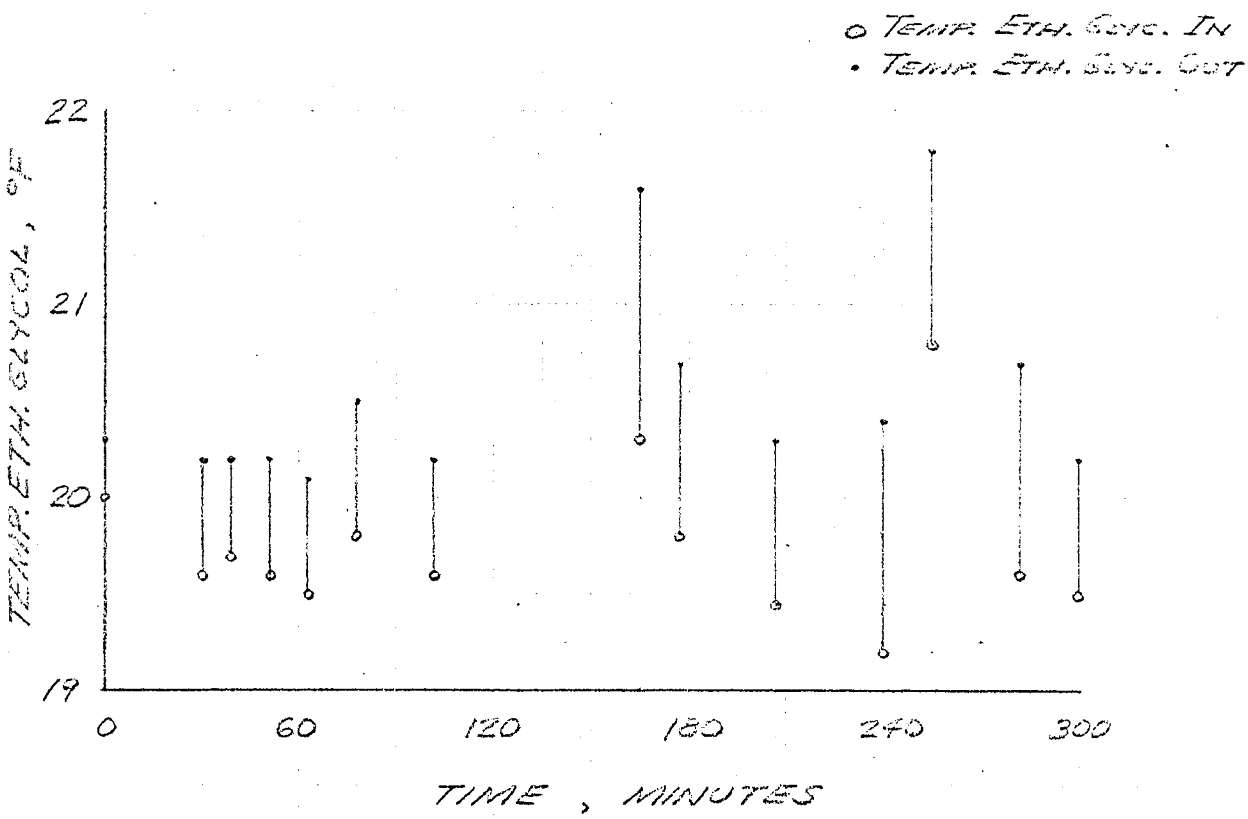


GRAPH 10
THERMOCOUPLE DATA
RUN 3

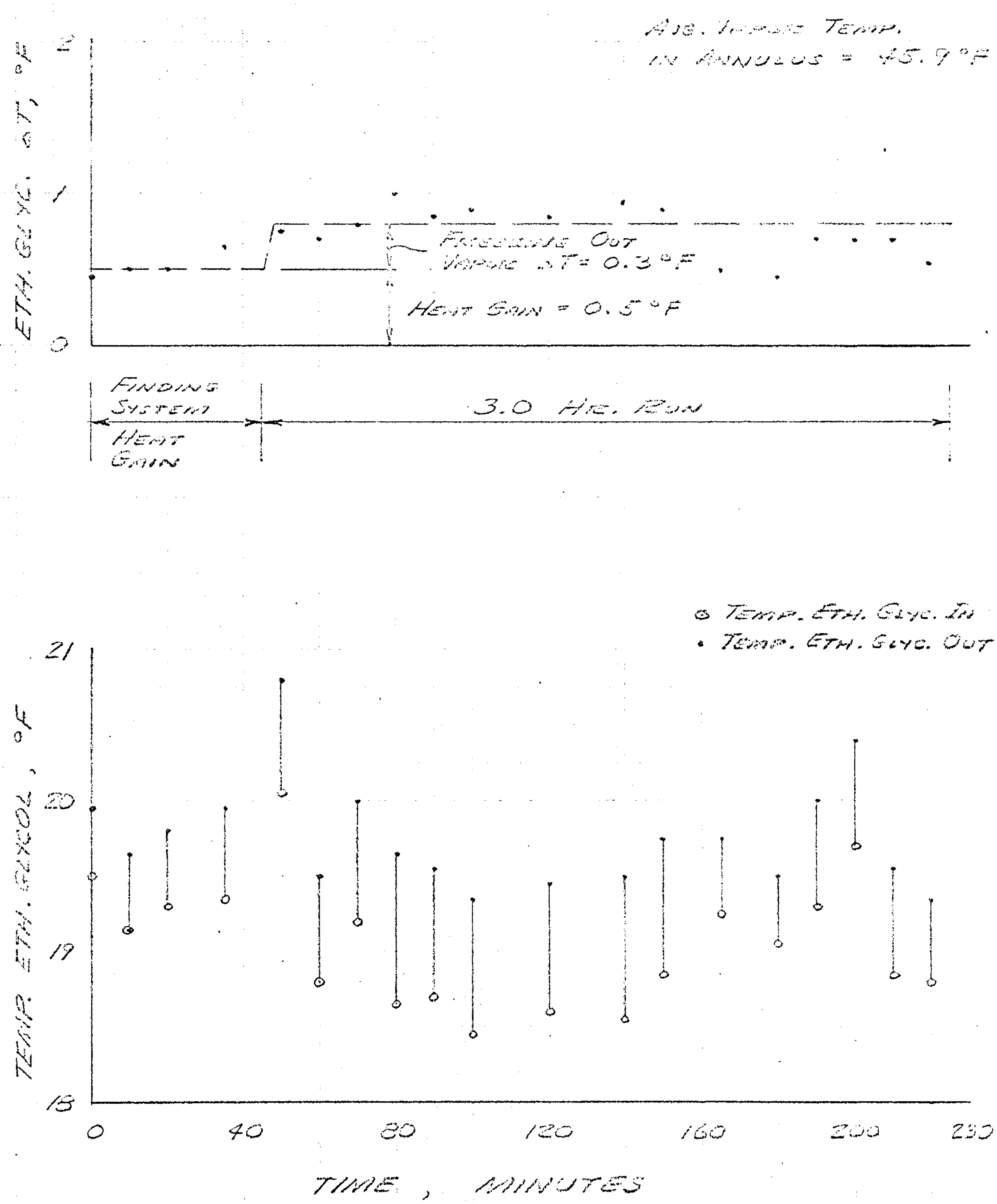


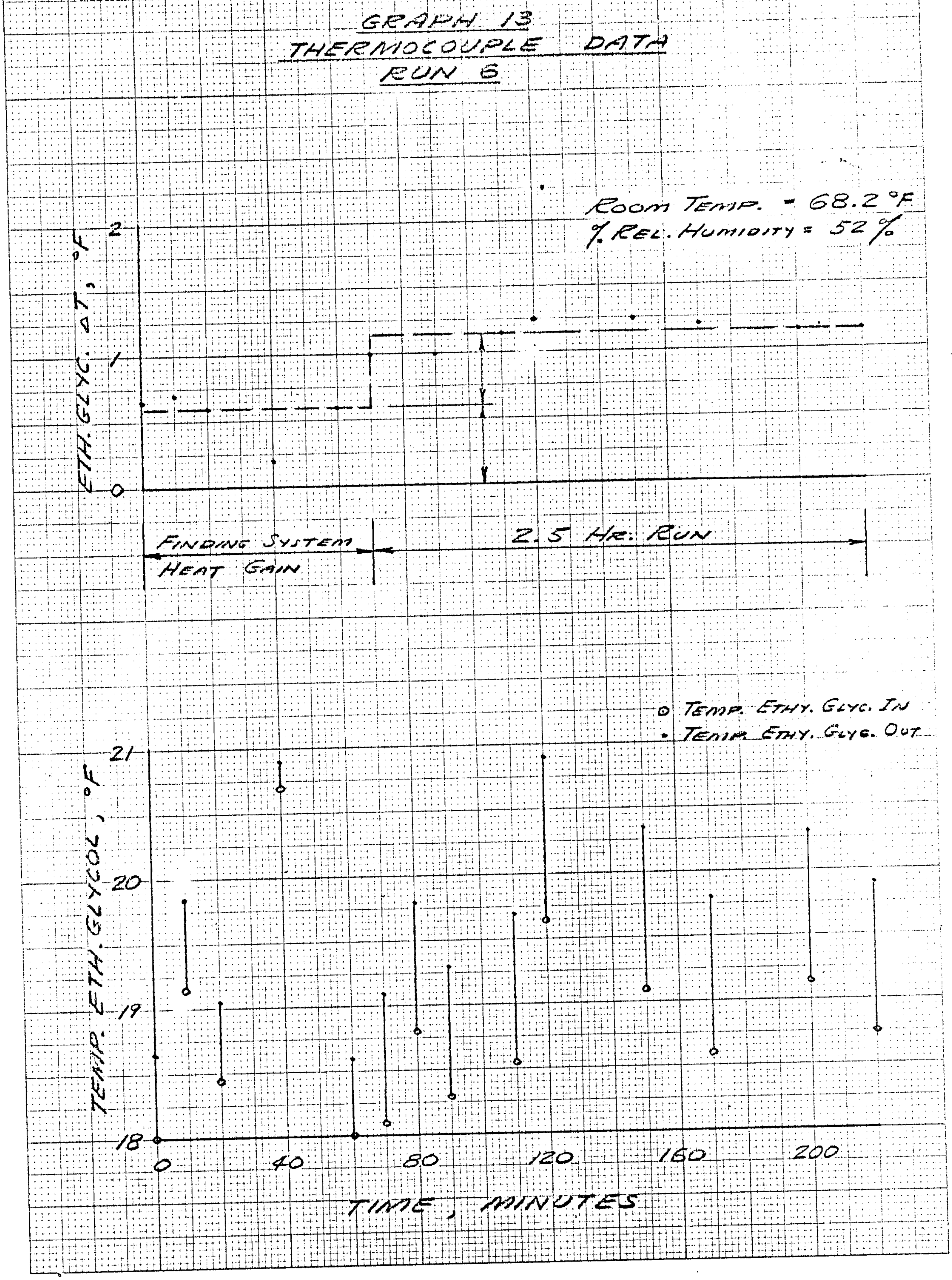
GRAPH 11
THERMOCOUPLE DATA
RUN 4



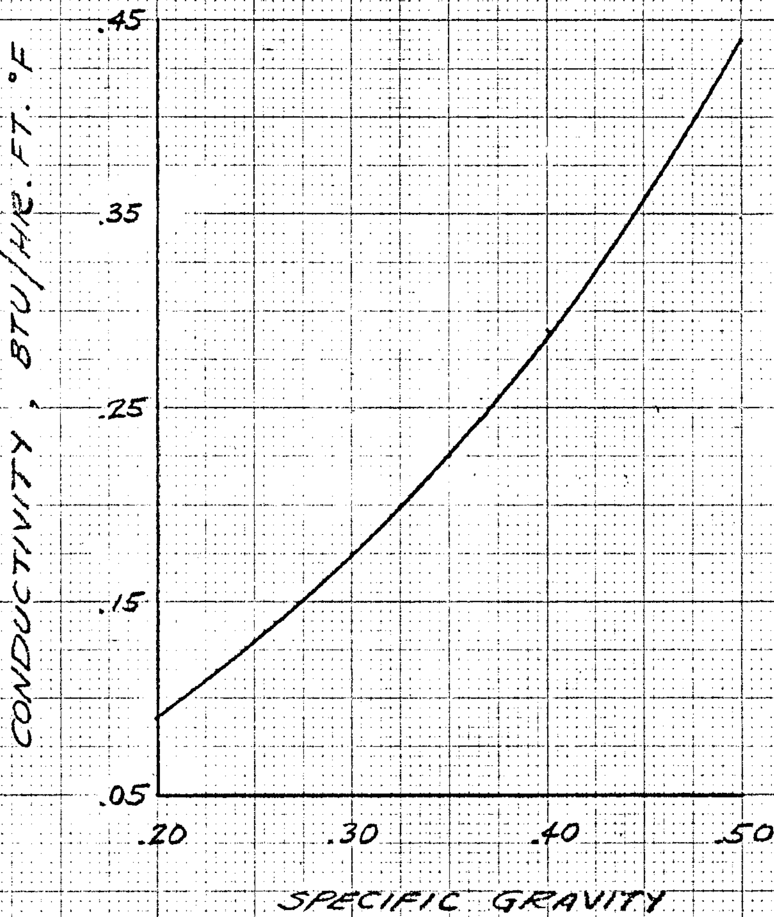


GRAPH 12
THERMOCOUPLE DATA
RUN 5



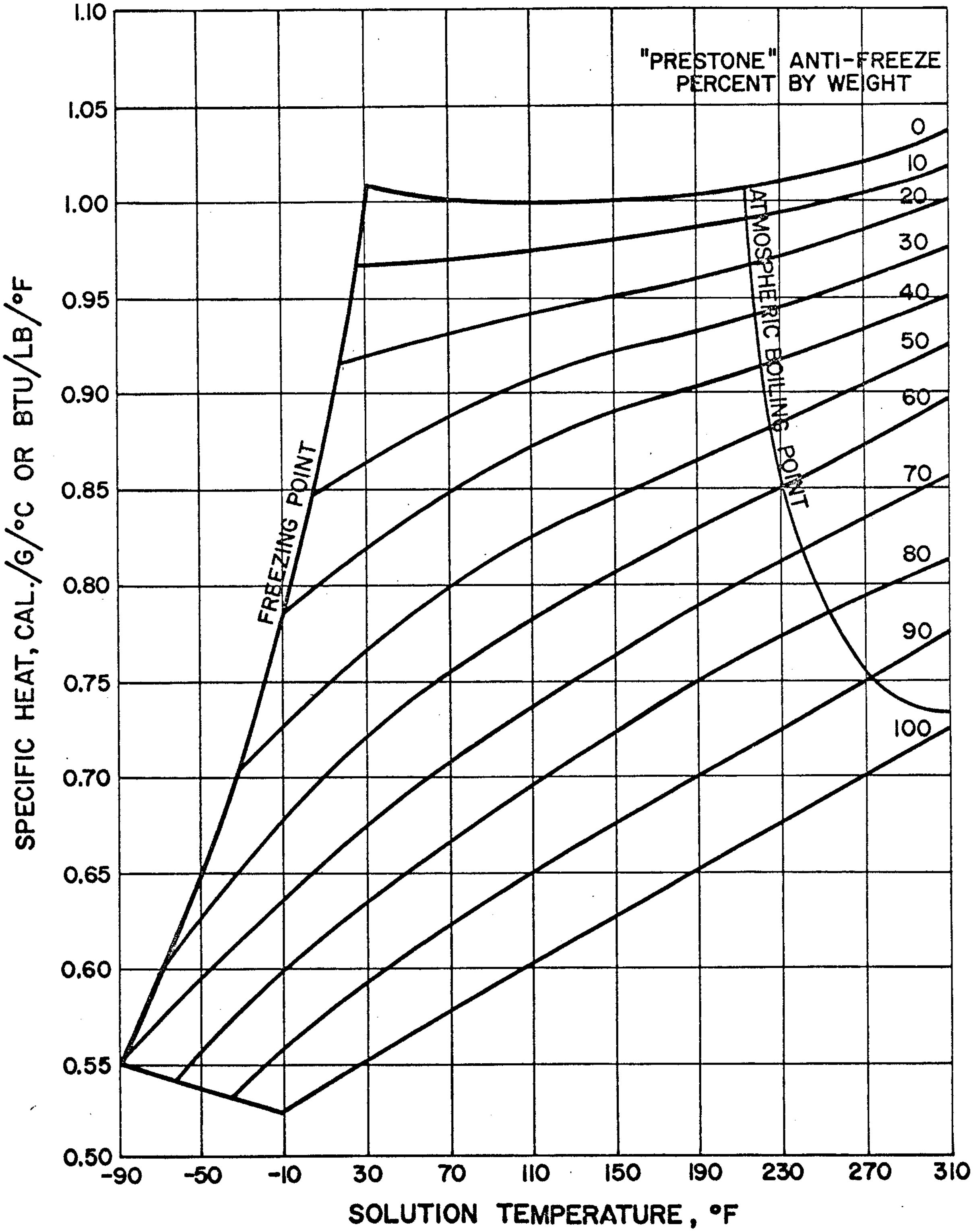

GRAPH 13
THERMOCOUPLE DATA
RUN 6
Room Temp. = 68.2 °F
% Rel. Humidity = 52 %
ETH. GLYC. ΔT, °F
2
1
0
Finding System Heat Gain
2.5 Hr. Run
Temp. Ethy. Glyc. In
Temp. Ethy. Glyc. Out
TEMP. ETH. GLYCOL, °F
21
20
19
18
0
40
80
120
160
200
TIME, MINUTES

GRAPH 14
THERMAL CONDUCTIVITY
OF FROST [8]



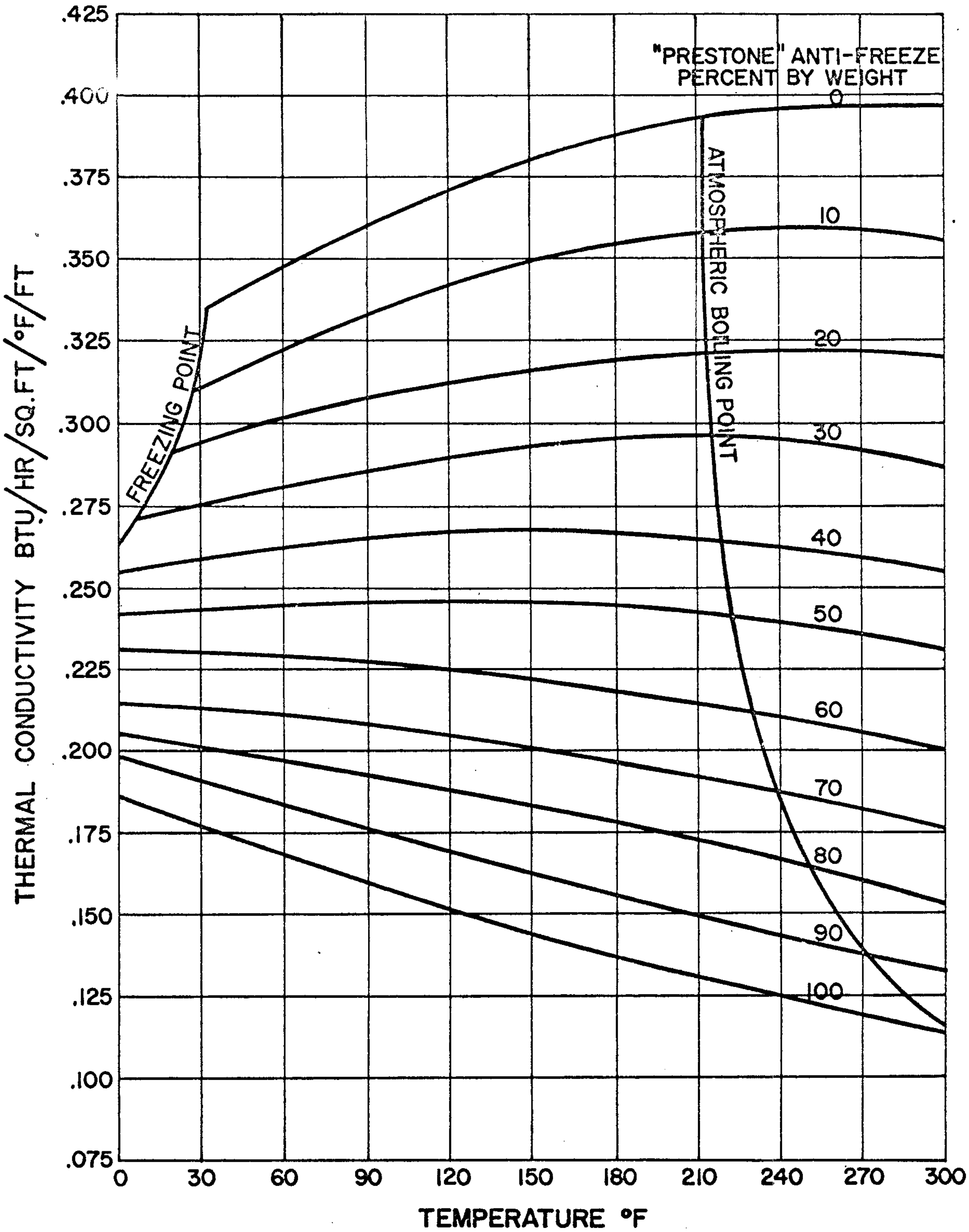
GRAPH #15

SPECIFIC HEAT OF AQUEOUS "PRESTONE" ANTI-FREEZE SOLUTIONS VS. TEMPERATURE [10]



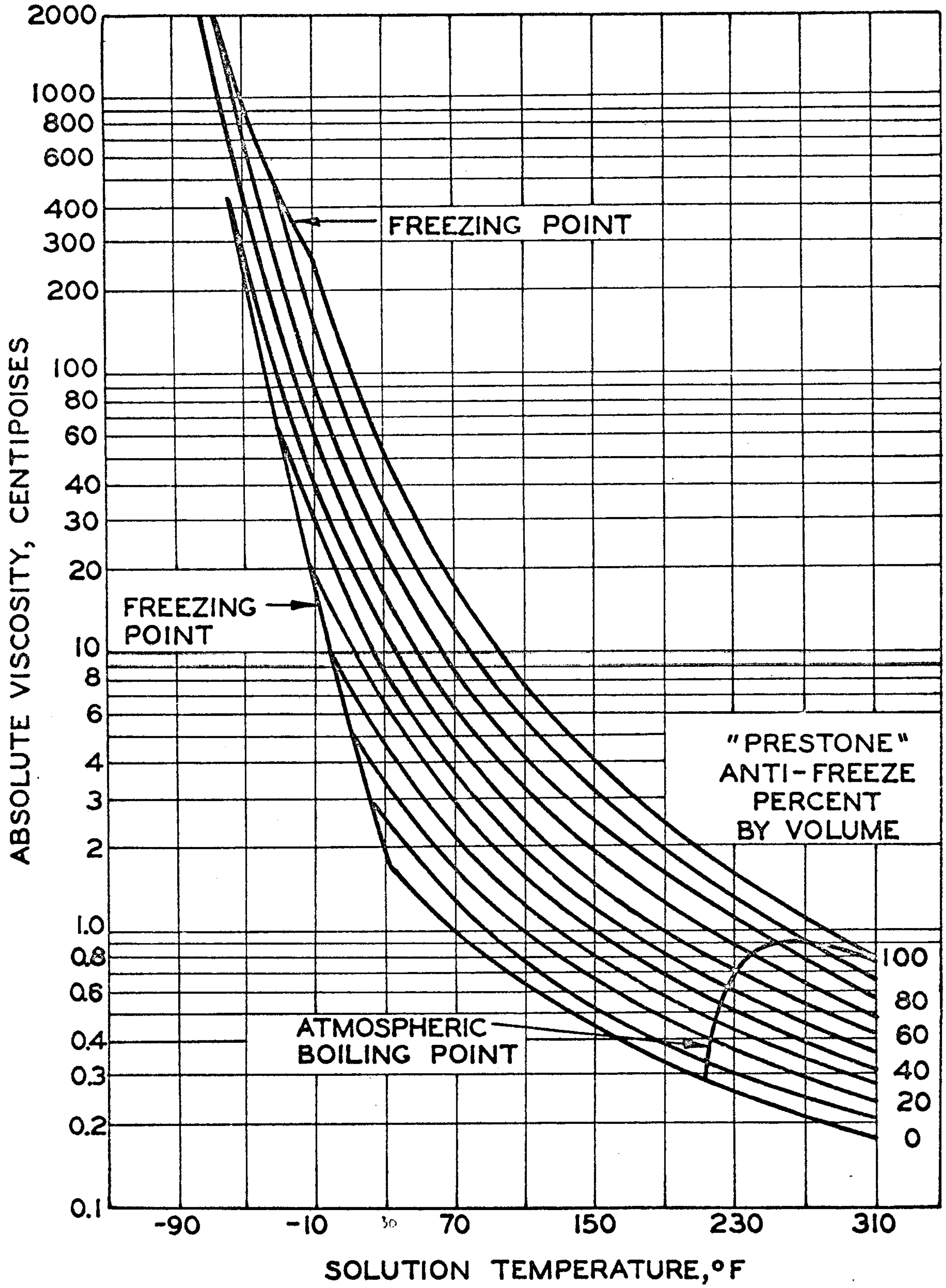
GRAPH #16

THERMAL CONDUCTIVITY OF AQUEOUS "PRESTONE" ANTI-FREEZE SOLUTIONS AT VARIOUS TEMPERATURES.[10]



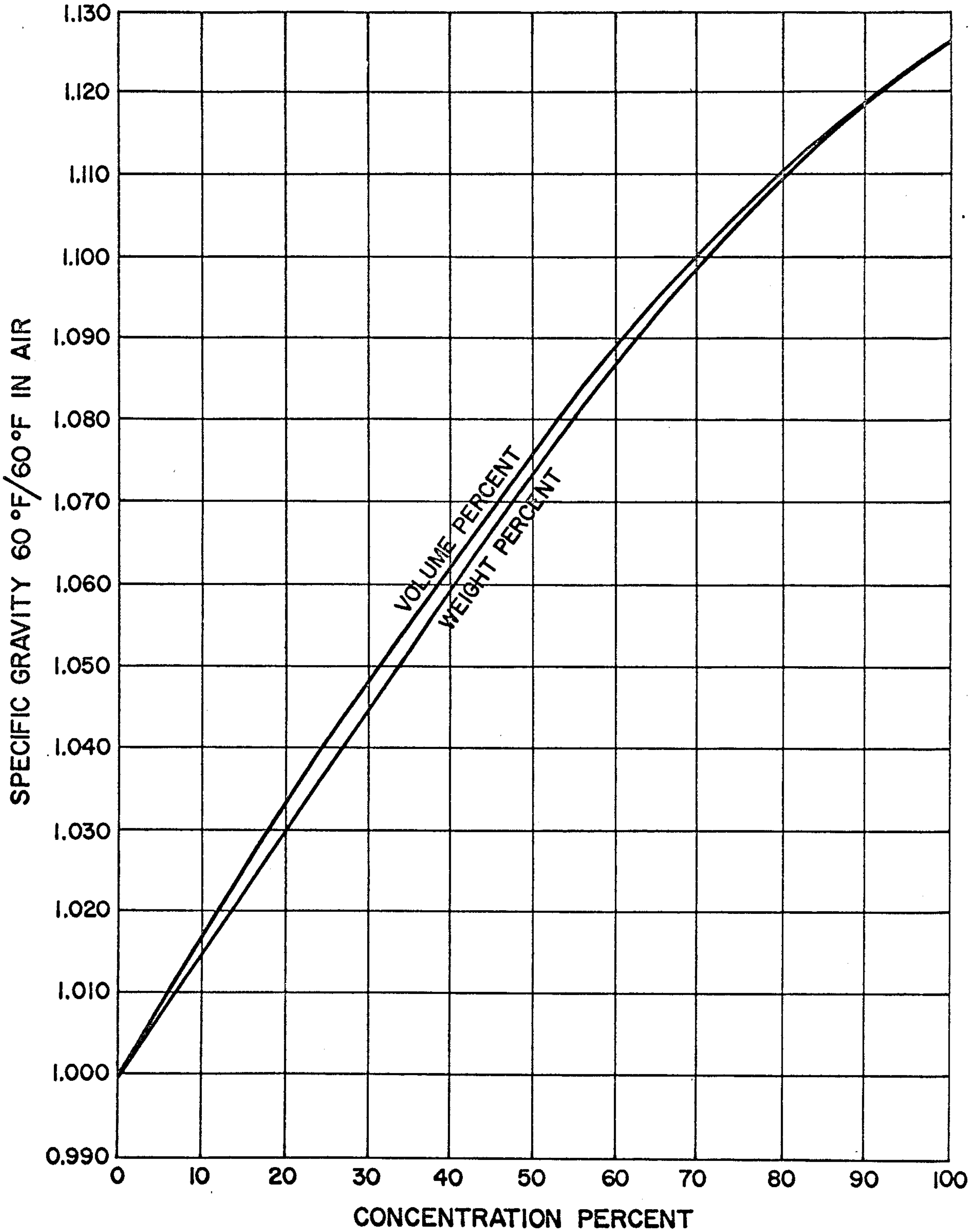
GRAPH #17

ABSOLUTE VISCOSITY OF AQUEOUS "PRESTONE" ANTI-FREEZE SOLUTIONS [10]



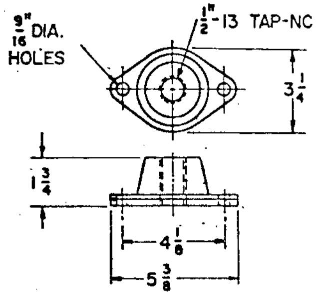
GRAPH #18

SPECIFIC GRAVITY (HYDROMETER DATA) AT 60 °F/60 °F
VS VOLUME PERCENT AND WEIGHT PERCENT
AQUEOUS "PRESTONE" ANTI-FREEZE SOLUTIONS[10]

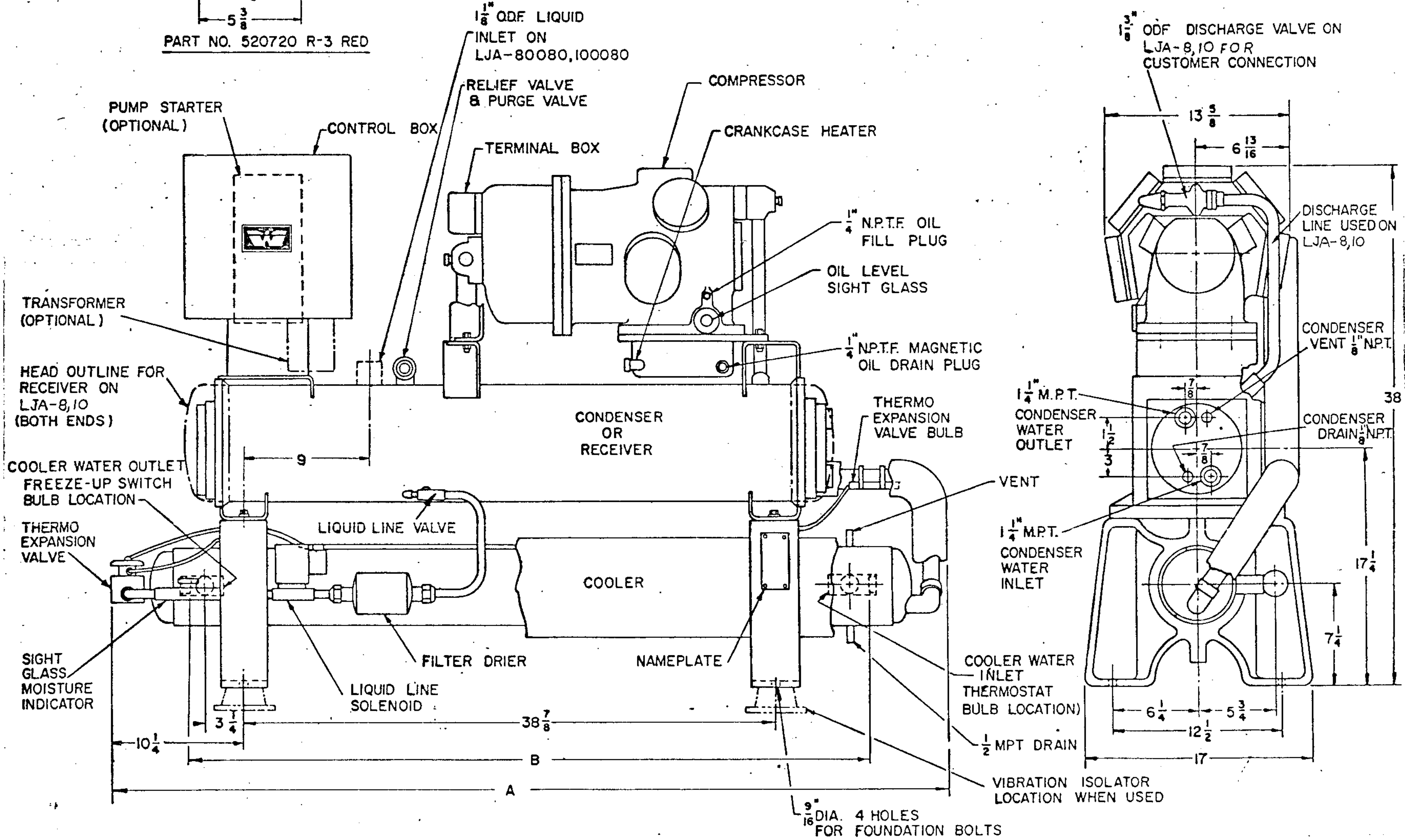


LIQUID CHILLER

UNIT MODEL	R-22 CHARGE	OPERATING WEIGHT		A	B
LJA-8	18 LBS.	665 LBS.		62	47 3/8
LJA-10	20 LBS.	675 LBS.		68	53 3/8



PART NO. 520720 R-3 RED



58

REFERENCES

1. Chen, M.M. and W. Rohsenow, "Heat, Mass, and Momentum Transfer Inside Frosted Tubes", Transactions of the American Society of Mechanical Engineers, series C86, page 334-340, August,1964.

2. Bryan, W.L., "Heat and Mass Transfer in Dehumidifying Surface Coils", American Society of Heating, Refrigeration, and Air-Conditioning Engineers Journal, vol 3, page 51-54, Sept., 1961.

3. Beatty, K.O., E.B. Finch, and E.M. Schoenborn, "Heat Transfer from Humid Air to Metal under Frosting Conditions", Refrigeration Engineering, vol. 59, page 1203, Dec.,1951.

4. London, A.L. and R.A. Seban, "Rate of Ice Formation", Transactions of the American Society of Mechanical Engineers, vol. 65, page 771-778, 1943.

5. McAdams, William H., Heat Transmission, New York: McGraw-Hill Book Company, Inc., 1954.

6. Kern, Donald Q., Process Heat Transfer, New York: McGraw-Hill Book Company, Inc., 1950.

7. Chemical Engineers' Handbook, John H. Perry (ed.), Third Edition, New York: McGraw-Hill Book Company, Inc., 1954.

8. Chung, P.M. and A.B. Algren, "Frost Formation and Heat Transfer on a Cylinder Surface in Humid Air Cross Flow", Heating, Piping and Air Conditioning, vol. 30, page 115-122, Oct. 1958 and vol. 30, page 171-178, Sept., 1958.

9. Wilson, E.E., "A Basis for Rational Design of Heat Transfer Apparatus", Transactions of the American Society of Mechanical Engineers, vol. 37, page 47-82, 1915.

10. Union Carbide Corporation, "Physical Properties Data, Prestone Anti-Freeze Coolant", Tarrytown, New York, 1966.