

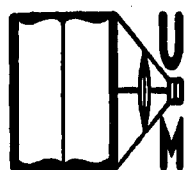
DOCTORAL DISSERTATION SERIES

TITLE EFFECT OF VAPOR AGITATION
ON BOILING COEFFICIENTS AT LOW
TEMPERATURE DIFFERENCES

AUTHOR DONALD B. ROBINSON

UNIVERSITY OF MICHIGAN DATE 1949

DEGREE Ph.D PUBLICATION NO. 2366



UNIVERSITY MICROFILMS
ANN ARBOR • MICHIGAN

EFFECT OF VAPOR AGITATION
ON BOILING COEFFICIENTS
AT
LOW TEMPERATURE DIFFERENCES

by

Donald B. Robinson

A dissertation submitted in partial fulfillment of the requirements for the degree of Doctor of Philosophy in the University of Michigan.

Committee in Charge:

Dr. D. L. Katz, Chairman
Dr. A. S. Foust
Dr. J. J. Martin
Dr. F. C. Moesel
Dr. F. L. Schwartz

Ann Arbor, Michigan

August, 1949

I. ACKNOWLEDGEMENT

The writer wishes to express his sincere gratitude to the following:

To Dr. D. L. Katz and the other members of the Doctoral Committee for their guidance and interest in the work.

To the Wolverine Tube Division of the Calumet and Hecla Copper Co. for their generous fellowship which made this work possible.

To Mr. G. Foster and his staff for their cooperation in the construction of the equipment.

To all those whose assistance in the collection of the data and the installation of the thermocouples was indispensable.

II. SUMMARY

A small horizontal tube and shell refrigeration cooler has been designed and constructed for purposes of studying the effect of vapor agitation on boiling coefficients. Recirculated water was used for the heating fluid and Freon 12 was used as the boiling liquid. Heat transfer measurements were made on four individual $3/4$ in. O.D. plain copper tubes placed on $1\frac{1}{2}$ in. centers in a vertical row.

The effect of one tube on another on the boiling coefficients was studied for boiling film temperature differences from 5.4°F to 19.5°F at heat fluxes from 874 to 35,100 BTU/hr. sq. ft. The effect of vapor artificially injected beneath the tube bundle was studied on all four tubes for three different conditions, and on the single bottom tube for boiling temperature differences from 8.27°F to 18.35°F at injection rates from 0 to 235 cu. ft/hr. The coefficient was found to increase by 218% at a boiling temperature difference of 6°F and 0% at a temperature difference of 18.35°F .

The effect of the surface on boiling coefficients was observed over a period of 32 days. During this period the coefficients changed to about 67% of their original value at low temperature differences and about 39% at higher temperature differences.

An equation in which the Reynolds number appears to the 0.88 power has been determined from the experimental

data for calculating water coefficients inside the tubes. The Wilson plot method has been used to calculate boiling coefficients at two different conditions.

A satisfactory method of installing thermocouples in thin-walled copper tubes has been described in detail.

TABLE OF CONTENTS

I.	Acknowledgement.....	1
II.	Summary.....	ii
III.	List of Tables.....	vii
IV.	List of Figures.....	viii
V.	Introduction.....	1
VI.	Theoretical Considerations.....	2
	A. Introduction.....	2
	B. Previous Investigation.....	5
	C. Experimental Method.....	6
VII.	Experimental Equipment and Measurements.....	7
	A. Equipment.....	7
	1. Freon Circuit.....	7
	(a) Flowsheet.....	7
	(b) Evaporator.....	12
	(i) Body.....	13
	(ii) Cover Plates.....	15
	(iii) Packed End.....	18
	(iv) Tubesheets.....	18
	(v) Tube Bundle.....	20
	(vi) Packing and Gasketing.....	23
	(vii) Pressure Test.....	23
	(c) Compressor and Accessories.....	24
	(d) Vapor-Liquid Separator.....	25
	(e) Storage Tank.....	25
	2. Water Circuit.....	26
	(a) Pump.....	26
	(b) Heater.....	26
	(c) Manifolds.....	28
	(d) Storage and Piping.....	28
	3. Insulation.....	29
	(a) Evaporator.....	29
	(b) Piping.....	29
	B. Measurements.....	30
	1. Flow Measurements.....	30
	(a) Water Rate.....	30
	(i) Orifice Installation.....	30
	(ii) Orifice Calibration.....	32
	(b) Freon Rate.....	32
	(i) Orifice Installation.....	32
	(ii) Orifice Calibration.....	33

TABLE OF CONTENTS CONTINUED

2.	Temperature Measurements.....	33
(a)	Water Temperatures.....	33
(i)	Thermometer Installation.....	33
(ii)	Calibration and Setting.....	36
(b)	Tube Wall and Freon Temperatures....	37
3.	Pressure Measurements.....	37
VIII.	Experimental Procedure.....	38
A.	Starting Up.....	38
1.	Water Circuit.....	38
2.	Freon Circuit.....	39
B.	Collection of Data.....	42
1.	Regular Runs.....	42
2.	Wilson Plot Runs.....	43
3.	Injection Vapor Runs.....	44
C.	Shutting Down.....	44
IX.	Treatment of Experimental Data.....	45
A.	Introduction and Preliminary Analysis....	45
B.	Boiling Coefficients.....	51
1.	Effect of Agitation.....	51
(a)	Tubes in a Vertical Row.....	51
(b)	Artificially Injected Vapor.....	56
2.	Effect of Surface Conditions.....	68
3.	Coefficients for Freon 12.....	71
C.	Inside Coefficients.....	73
1.	Correlation of Experimental Data.....	73
2.	Wilson Plot Analysis.....	75
D.	Accuracy of Measurements.....	79
X.	Conclusions.....	81
A.	Factors Influencing Boiling Coefficients.	81
B.	Conclusions Summarized.....	86
XI.	Appendix A. Installation of Thermocouples.....	A1
A.	Introduction.....	A1
B.	Method of Installation.....	A2
1.	Preparation of Tubes.....	A3
2.	Assembly of Thermal Elements.....	A6
3.	Assembly of Elements and Tubes.....	A7
4.	Packing of Lead Wires.....	A8
5.	Switching Arrangement and Cold Junction	A13
6.	Freon Liquid Couples.....	A14
C.	Calibration.....	A14

TABLE OF CONTENTS CONTINUED

XII.	Appendix B. Auxiliary Operations Pertaining.....	A16
	to Operation of the Freon Circuit.	
	A. Charging Liquid to the Evaporator.....	A16
	B. Removing Freon from the System.....	A16
	C. Charging Oil to Compressor.....	A17
	D. Adjustment of Oil Pressure.....	A18
XIII.	Appendix C. Experimental and Calibration Data...	A19
XIV.	Nomenclature.....	A40
XV.	Bibliography.....	A41

III. LIST OF TABLES

I.	Wilson Plot Analysis.....	75
II.	Summary of Experimental Data.....	A20
III.	Gas Injection Calculations.....	A29
IV.	Cross Plot Calculations.....	A30
V.	Calculation of Inside Coefficients.....	A31
VI.	Wilson Plot Calculations.....	A32

IV. LIST OF FIGURES

1. Flowsheet of Freon Circuit.....	10
2. Evaporator Assembly	14
3. Cover Plate-Unpacked End	16
4. Cover Plate-Packed End	17
5. Packing Follower Detail	19
6. Tubesheet - Packed End	21
7. Plain Tube Bundle	22
8. Flowsheet of Water Circuit	27
9. Orifice Assembly	31
10. Beckmann Installation	35
11. Effect of Fouling on Overall Coefficients	47
12. Effect of Tube Position on Boiling Coefficients	53
13. Effect of Vapor Injection on Boiling Coefficients - Runs 128-131	61
14. Effect of Vapor Injection on Boiling Coefficients - Runs 132-137	62
15. Effect of Vapor Injection on Boiling Coefficients - Runs 140-143	63
16. Effect of Vapor Injection on Boiling Coefficients For a Single Tube	65
17. Boiling Coefficient as a Function of Vapor Injection....	67
18. Increase in Coefficients Resulting from Vapor Injection..	69
19. Effect of Surface on Boiling Coefficients	72
20. Coreelation of Inside Coefficients	74
21. Wilson Plot - Runs 123-126	76
22. Wilson Plot - Runs 144-147	77
23. Thermocouple Assembly	A4
24. Jig for Drilling Tube Wall	A5

25.	Packing Gland Assembly.....	A9
26.	Packing Gland Detail	A11
27.	Packing Gland Installation.....	A12
28.	Orifice Calibration	A33
29.	Freon Orifice Calibration	A34
30.	Beckmann Calibration	A35
31.	Temperature Corresponding to Beckmann Reading	A36
32.	Pressure Gauge Calibration - C2-108	A37
33.	Pressure Gauge Calibration - C2-378	A38
34.	Thermocouple Calibration	A39

THE EFFECT OF VAPOR GENERATION ON
BOILING COEFFICIENTS AT LOW TEMPERA-
TURE DIFFERENCES.

V. INTRODUCTION

The experimental program which resulted in this dissertation was begun in December, 1947. At that time it was proposed to study boiling coefficients for Freon 12 under conditions resembling somewhat those in a large horizontal tube and shell water cooler. The effect on the heat transfer coefficient of the agitation resulting when bubbles from one tube strike the tube above was to be studied for a bundle of six tubes in a vertical row, using both plain and finned tubes. The effect of artificially injected vapor on the coefficient for the bottom tube was also to be studied.

Accordingly, work was started on the design of suitable experimental equipment. Full time was devoted to its construction from February until October, 1948, when the first data were recorded. Difficulty with the analysis of the data led to considerable rebuilding of part of the equipment in May, 1949. This in turn resulted in a slight reduction in the scope of the investigation. Boiling coefficients were subsequently studied for Freon 12 at low temperature differences on four plain tubes in a vertical row. The effect of artificially introduced vapor was observed over a limited range of rates and temperature differences.

VI. THEORETICAL CONSIDERATIONS

A. INTRODUCTION

Boiling may be defined as that phenomenon which occurs when heat is transferred to a liquid across a finite boundary or medium which separates the liquid from the source of heat at such a rate and under such conditions that bubbles of vapor form at the interface. These bubbles form at few or many points on the interface, depending on the rate of heat transfer, the history of the surface, and the dynamics of flow of the liquid at the interface at the time of bubble formation.

Experimental observations indicate that the boiling which results from the contact of a hotter metal surface with a colder liquid occurs in three more or less clearly defined regions which have been called nuclear, filmwise, and critical. Nuclear boiling is characterized by low temperature differences and a vapor-liquid mixture in contact with the heated surface. Filmwise boiling is characterised by relatively higher temperature differences and an unstable thin film of vapor almost completely separating the heated surface from the bulk of the cooler liquid mass. The critical region is a transition zone between these more clearly defined regions, similar to the transition between laminar and turbulent flow of fluids.

In the refrigeration industry, wherever horizontal tube and shell coolers are used, the liquid boils during

the evaporation of the low-pressure refrigerant which is in indirect contact with the circulated cooling medium. Evaporation of this liquid at a temperature as close as possible to the required temperature of the cooling medium possesses significant advantages over evaporation at lower temperatures. The higher the evaporating temperature the less work must be done by a compressor operating at a fixed condensing temperature. The objective in this type of heat transfer is, therefore, to get the size of the heat transfer equipment and the required compressor work at an overall economic optimum. Low temperature differences reduce the required compression energy on one hand but reduce the rate of heat transfer on the other. The optimum will therefore be found by getting the maximum heat transferred at the lowest temperature difference. Under these conditions boiling will be in the nuclear region.

It has been general practice in this country to correlate so-called boiling film coefficients for given liquids boiling in the nuclear region on the basis of the difference in temperature between the heating surface and the liquid. Practically all previous studies (5, 6, 7, 9, 11, 12) indicate that for boiling from plain surfaces in the vertical or horizontal position, the rate of heat transfer decreases very rapidly as the temperature difference is decreased. The coefficients for given liquids have been represented by an equation of the type

$$h = E (\Delta T)^m \text{ - - - - - (1)}$$

where $\underline{h}$ is the boiling film coefficient based on the area on which boiling is taking place in BTU/hr. sq. ft. $^{\circ}\text{F}$; $\underline{E}$ is a constant which depends largely on the liquid and the surface; $\underline{\Delta T}$ is the temperature difference between the boiling liquid and the interface across which heat is being transferred in $^{\circ}\text{F}$; and $\underline{m}$ is a constant which apparently lies between 2.4 and 2.6 for most liquids (5, 9). This indicates the rapid rise in the coefficient with temperature difference, and hence its relatively low value at low temperature differences.

Contrary to the above equation, data presented by Austin (3) for boiling under violent mechanical agitation indicate no change in heat transfer coefficient with temperature difference. These data indicate that temperature difference is not the only major fundamental variable, even for a given liquid boiling on a given surface. Since the dynamics of the boiling liquid seem to have such a dominant effect, it seems reasonable to assume that the tube arrangement and vapor-liquid flow pattern around the tubes will significantly affect the rate of heat transfer in a horizontal tube and shell evaporator. This should be particularly apparent at low boiling rates where normal agitation is relatively poor.

It was primarily with a study of this effect in mind that the present research was undertaken. At the same time, the study would permit measurement of boiling coefficients for Freon 12.

B. PREVIOUS INVESTIGATIONS

Out of the vast array of literature on heat transfer to boiling liquids, six papers pertaining more or less directly to the points in question are worthy of mention.

Two other authors have reported boiling coefficients for Freon 12. Ashley (2) studied heat transfer coefficients for Freon 12 in a direct expansion type double-pipe exchanger. An attempt was made to use a Wilson (16) plot type of analysis to evaluate the film coefficients. Jones (13) reports data on a 430 ton cooler containing 5/8 in. root diameter integral spiral low finned tubes for water velocities from 3.45 to 7.51 ft./sec; evaporating temperatures from 30.24 to 34.50 °F, and water inlet temperatures from 42.86 to 55.50 °F. These conditions corresponded to a boiling film temperature difference from about 4.3 to 5.2 °F and boiling film coefficients from about 140 to 360 BTU/hr. sq. ft. °F.

One other investigation on the comparison of single tube coefficients with tube bundle coefficients is reported. Akin (1) studied overall coefficients in an evaporator having 60 nearly horizontal 1/2 in. O.D. chrome-plated copper tubes 10 in. long. The rows were 10 deep and 6 across on a square pitch. Heat transfer was from condensing steam to boiling water. Condensate was collected separately from each of 5 pairs of horizontal tubes, i.e., 12 tubes. He reports that the data for the 60 tube tests were virtually the same as those for a single tube under the same conditions, but that the third pair from the bottom produced

slightly more condensate than the others. No data were taken on the bottom pair because of an equipment failure.

Jakob (12) makes reference to the stirring effect of the bubbles. He experienced difficulty in making measurements on vertical plates because the boiling temperature difference was decreased near the top of the plate. He attributes this to the stirring effect of the rising steam bubbles. Jakob also reports his studies of the surface effect on the coefficient. Coefficients for a freshly sand blasted copper surface and for the same surface after longer periods of use indicate a decrease in coefficient of approximately 25% at the same heat flux.

Pridgeon and Badger (15) report a series of studies on boiling water from plain copper tubes. They measured overall coefficients and found a decrease of about 60% for a surface that had stood in water for 24 days as compared to an acid-clean surface. A surface cleaned with emery cloth gave overall coefficients about 60% of the acid-clean values.

The work of Austin (3) has been referred to. He reports that when a pool of water boiling at 212°F was mechanically agitated with a stirrer rotating at 200 r.p.m, the boiling coefficient was apparently unaffected by a change in temperature difference from 1 to 10°F .

C. EXPERIMENTAL METHOD

The method finally used in this investigation differed slightly from that originally conceived. The original plan

was to study rates of heat transfer to individual horizontal tubes in a vertical row tube bundle. The evaporating liquid was to be Freon 12. The heating liquid was to be carbon tetrachloride pumped at known velocity through the tubes. This liquid was chosen because of its low specific heat and its desirable liquid range. Inlet and outlet temperatures of the carbon tetrachloride and the temperature of the Freon were to be measured with mercury thermometers graduated in $1/10^{\circ}\text{F}$ intervals. The carbon tetrachloride velocities were to be measured with orifices. Studies on the effect of artificially introduced Freon vapor were to be made by passing known quantities of the vapor at a given temperature and pressure through a perforated tube located beneath the bottom tube in the bundle.

As will be explained later, several difficulties arose when an attempt was made to analyze the data taken under the above conditions. In fact all attempts at evaluating the boiling film coefficient satisfactorily resulted in failure.

The method of taking measurements was then changed to one which proved successful. First of all, the heating fluid was changed from carbon tetrachloride to water for two reasons. The carbon tetrachloride had a corrosive effect on the copper and other piping which led to the deposition of a powdery layer on the inside of the evaporator tubes. This layer had a tremendous influence on the rate of heat transfer. The other reason was that by sacrificing the lower specific heat of carbon tetrachloride

in favor of water, the higher inside coefficient of water was obtained. To overcome the decrease in accuracy which would result from measuring smaller temperature differences with $1/10^{\circ}\text{F}$ thermometers, Beckmann thermometers graduated in $1/100^{\circ}\text{C}$ intervals were used. To permit analysis of the heat transfer data into its separate coefficients, thermocouples were installed in the tube walls of the bottom four plain tubes. The two top tubes were equipped with neither thermocouples nor Beckmanns. Other than this the experimental method was unaltered.

VII. EXPERIMENTAL EQUIPMENT AND MEASUREMENTS

A. EQUIPMENT

Two separate main fluid circuits and two auxiliary circuits were necessary for the measurement of the required rates of heat transfer. The first main circuit was capable of containing and handling the Freon used for cooling the water. The second was a water circuit which contained and handled the water used for evaporating the Freon. The first auxiliary circuit was for steam used to reheat the water and maintain it at any desired uniform temperature. The second was a cold water circuit used for removing heat from the condensing Freon.

1. FREON CIRCUIT

(a) Flowsheet

The Freon and its auxiliary circuit consisted of four main units: the evaporator, the compressor and its accessories, the vapor-liquid separator, and the storage tank. Their arrangement is indicated in the flowsheet of Fig. 1.

The high pressure Freon gas leaving the compressor divided into two streams, one going to the oil separator and the other being by-passed to the suction side of the compressor. It was for capacity control that part of the discharge stream was by-passed, and the relative magnitude of the two streams depended on the heat load of the evaporator. As the heat load was increased less Freon was by-passed and more went to the condenser.

The stream of gas entering the condenser was condensed

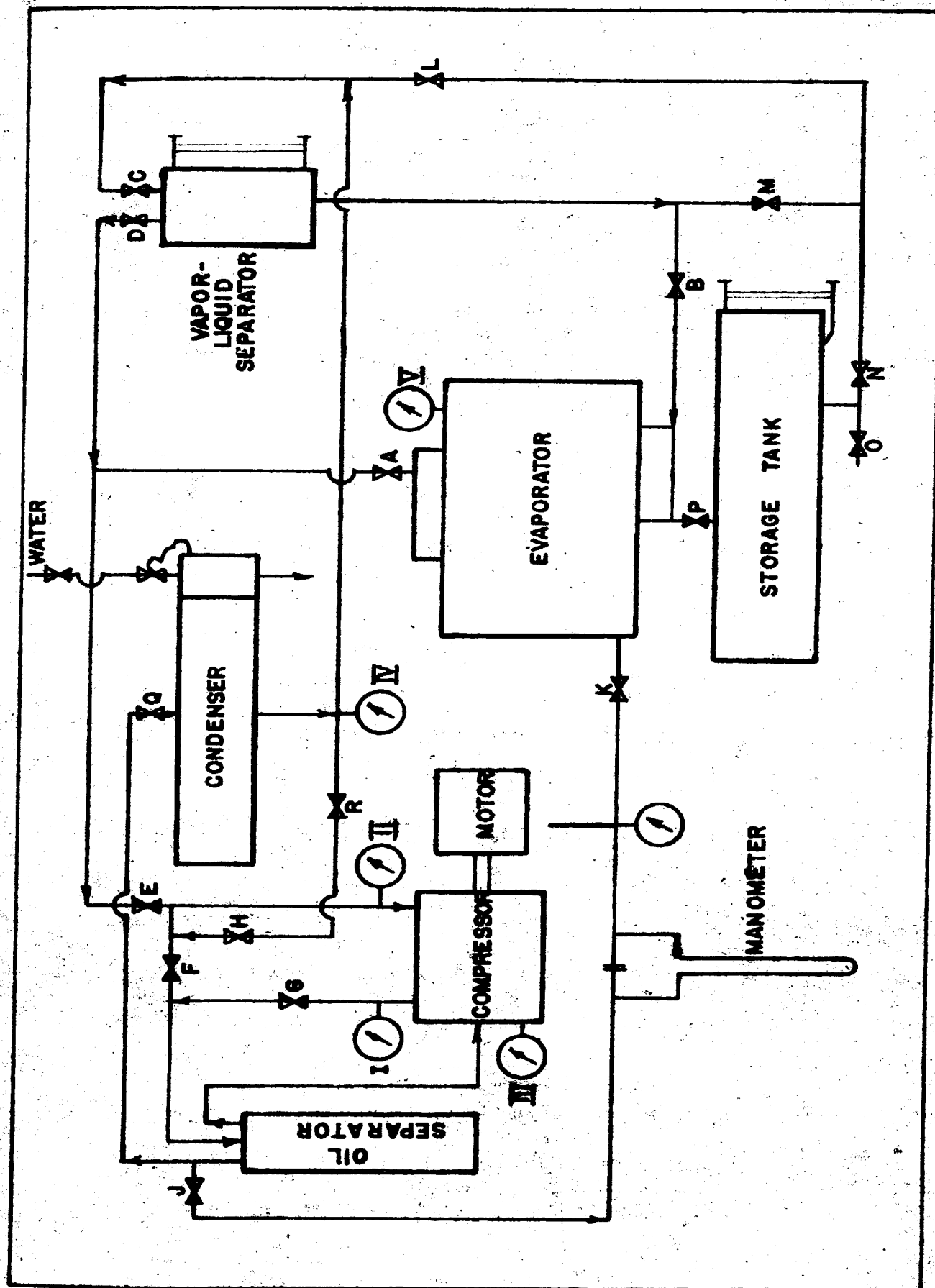


FIG. 1 FLOWSHEET OF FREON CIRCUIT

by cooling water from the city mains. The flow rate of the cooling water was automatically controlled by a pressure controlled valve in the water line. The stream of high pressure liquid from the condenser was divided into one main stream in a $3/4$ in. copper tube and another small stream in a $1/4$ in. copper tube. The small stream entered the vapor by-pass on the compressor just before it joined the main suction line. Here it was throttled to suction pressure by a hand operated needle valve H. The flashing of this high pressure liquid kept the compressor cool and eliminated the extreme overheating which otherwise resulted from continuous by-passing of hot high pressure vapor.

The main liquid stream entered the vapor-liquid separator after being throttled by a hand operated throttle valve C. Here the high pressure liquid stream flashed into two phases at approximately the evaporator pressure. The low pressure vapor stream from the separator was controlled by valve D and returned to the suction side of the compressor. The low pressure liquid stream from the separator flowed to the evaporator by gravity flow and entered the bottom at two points. Here the Freon was evaporated and the main vapor stream returned to the suction side of the compressor.

The vapor stream indicated as flowing directly from the discharge of the oil separator to the evaporator was for introducing the Freon gas beneath the bundle of tubes, and was completely isolated from the rest of the system when not in use. It was a $3/4$ in. copper line.

The remainder of the piping around the storage tank was only used while charging or discharging Freon from the evaporator. Its use will be described more fully in Appendix B.

All main vapor lines were 1-1/2 in. streamline copper and main liquid lines were 3/4 in. streamline copper. connections were made with brass sweat fittings. Valves were standard 150 lb. globe and gate valves which caused considerable trouble with leakage. Expansion valve C was a standard 3/4 in. hand controlled 300 lb. needle valve.

(b) Evaporator

The evaporator is worthy of rather detailed discussion because it was the unit on which the heat transfer measurements were made and it possessed some features of design which made it useful for experimental purposes.

The evaporator was designed with the following points in mind. It had to have a tube bundle consisting of six tubes in a vertical row, and there were to be two kinds of tubes. For this reason it was desirable to be able to remove and install different tube bundles without too much inconvenience and without destroying the tubes. It had to be able to withstand the pressure of the Freons and other low pressure refrigerants and packing glands had to be of a type which would not permit loss of these relatively expensive fluids. It was also preferable that the evaporator should operate on as little refrigerant as possible for economic reasons. The ends of the evaporator

had to be adaptable to the connection and disconnection of the water pipes. Provision had to be made at one end for the expansion and contraction of the evaporator tubes at different temperatures.

(i) Body

The above considerations led to the construction of a rectangular shaped evaporator. It provided space for the tube bundle and yet very little waste space which had to be filled with Freon. It also had a relatively small end area so that lighter weight tubesheets than would have been necessary for a circular shaped evaporator could be used.

An assembly drawing of the evaporator appears in Fig. 2. It measured $17\text{-}\frac{3}{8}$ in. high by $7\text{-}\frac{1}{4}$ in. wide by 39 in. long inside and was fabricated of $\frac{1}{2}$ in. steel plate welded at the seams and reinforced at three positions with 3 in. steel I-beams welded in place. The ends were welded to collars fabricated of 1 in. steel plate suitably drilled for bolting to the cover plate at one end and the packing gland follower at the other end.

The body of the evaporator contained two 1 in. welded pipe couplings in the bottom to introduce the liquid Freon. It had two $1\text{-}\frac{1}{2}$ in. couplings in the top to remove the vapor. The $\frac{3}{4}$ in. coupling was for a safety valve set to relieve at 150 p.s.i.g. The $\frac{1}{2}$ in. coupling in the top was for a pressure gauge and the ones in the side were for thermometer wells.

Three bull's eyes were welded into the sides of the

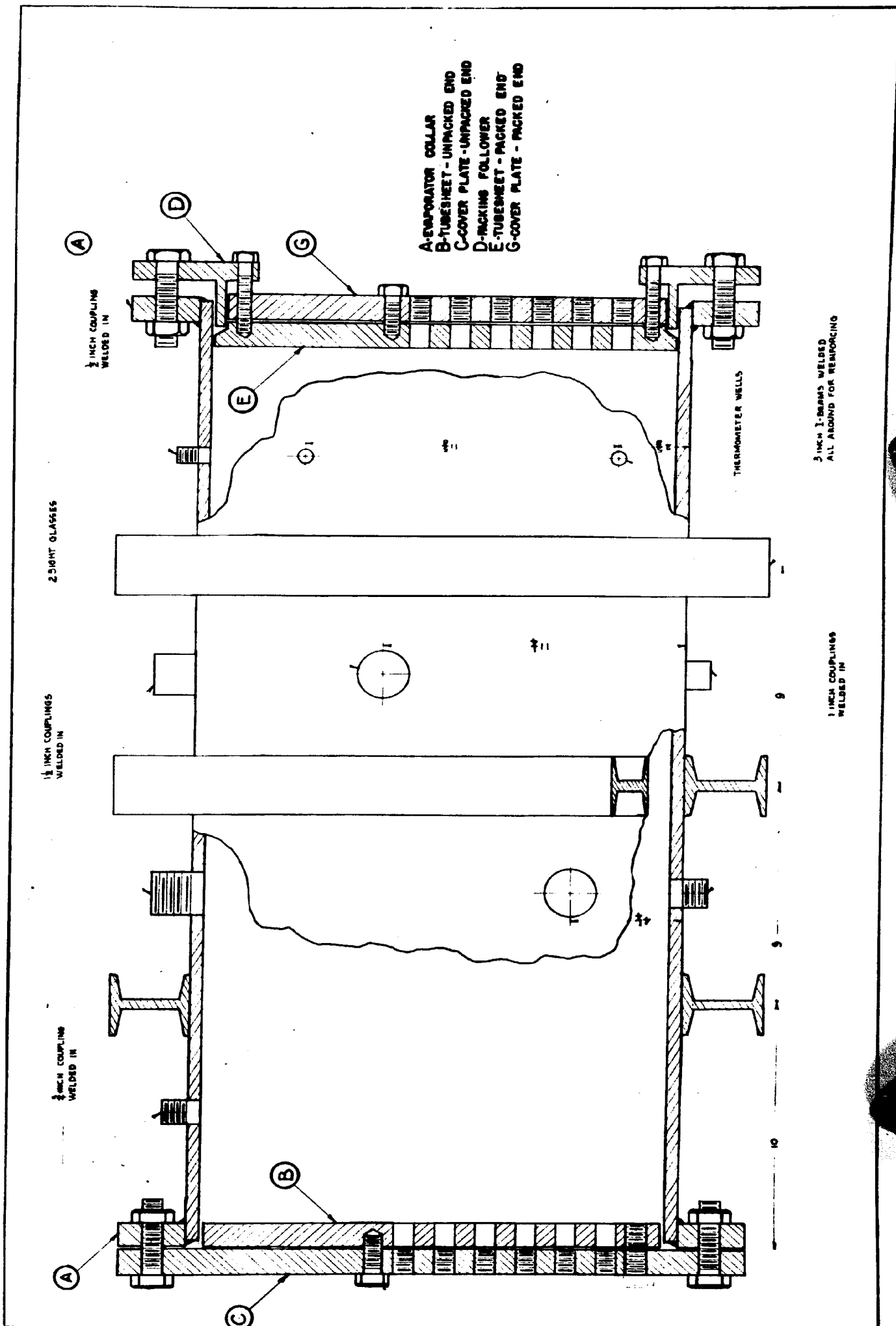


FIG. 2 EVAPORATOR ASSEMBLY

evaporator, two on the front and one on the back. The one on the back was half way between the front two. These were to permit visual observations of the boiling and the liquid level.

(ii) Cover Plates

The detail of the cover plate for the unpacked end is indicated in Fig. 3. It was fabricated of 1 in. steel plate. The outside ring of 14 holes was used to bolt the cover plate to the collar on the evaporator body. The inner ring of 8 holes was used to bolt the cover plate to the tubesheet in order to make a tight water seal between tubes and also a seal between the water and Freon circuits. The threaded $1/2$ in. pipe holes were for adapting the water pipes to the cover plate in order to introduce the water into the evaporator tubes. The $1/2$ in. pipe hole at the bottom was for adapting the Freon gas pipe for introducing the gas into the perforated injector.

The detail for the cover plate on the packed end is indicated in Fig. 4. It was fabricated from 1 in. steel plate. Ten of the outer ring of bolt holes corresponding to those in the packing gland follower were to permit the stud bolts used with the follower to pass through the cover plate into the tubesheet. In this way the outward pull of the bolts was exerted against the tubesheet rather than the cover plate and did not tend to separate the plate from the tubesheet as the packing was tightened up. The remaining six holes were for the bolts used to hold the cover plate and gasket to the tubesheet as on the unpacked

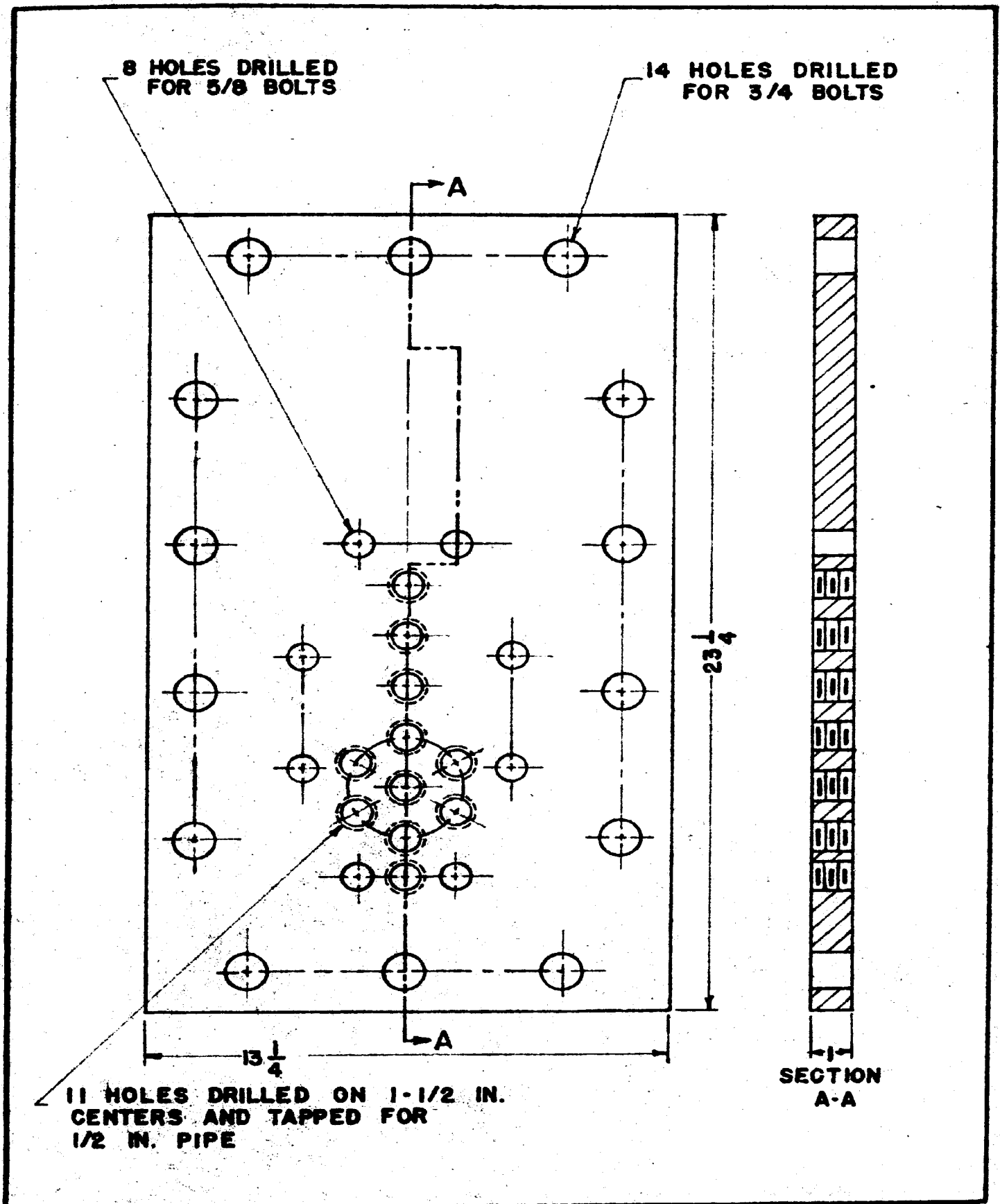


FIG.3 COVER PLATE DETAIL - UNPACKED END

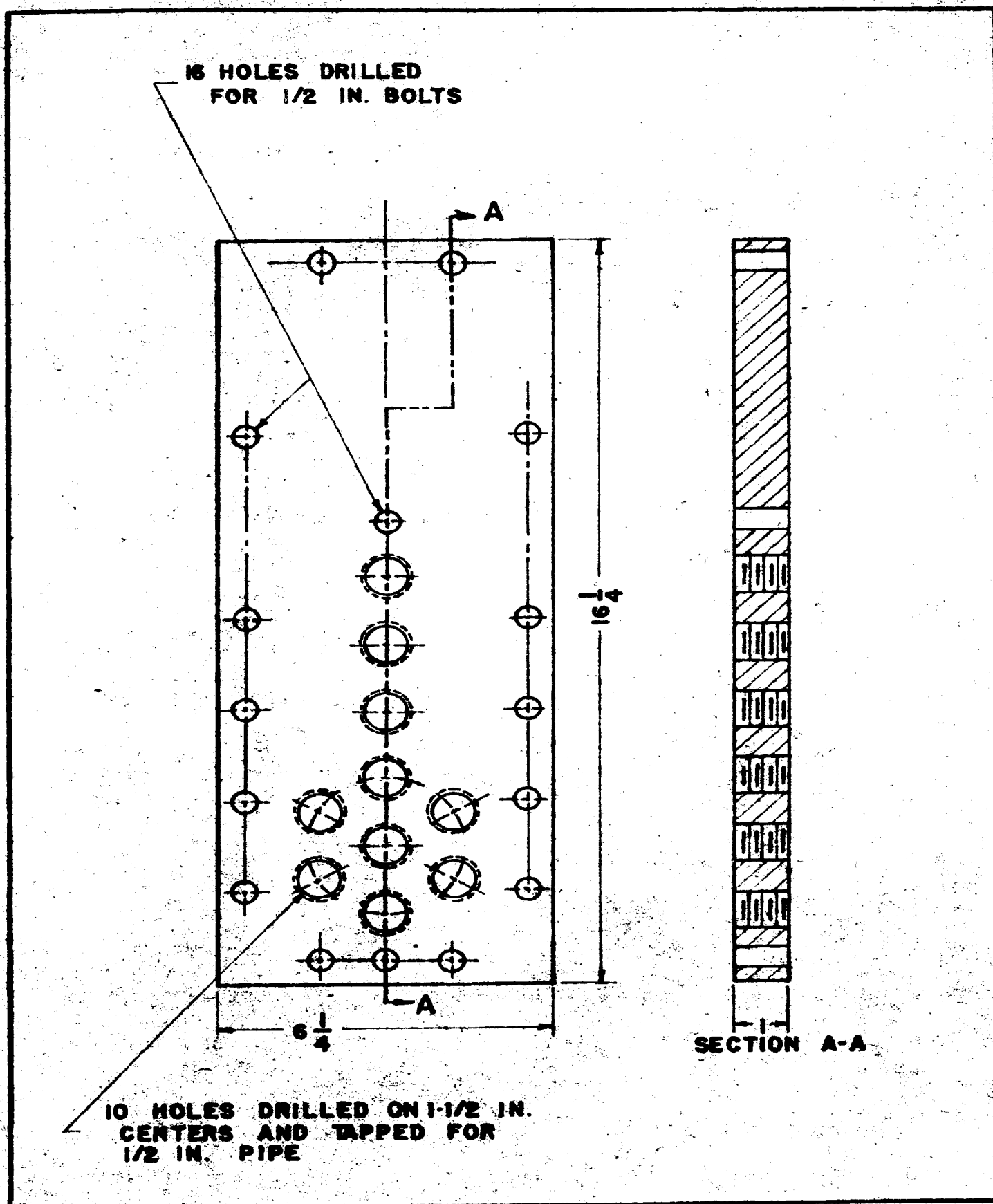


FIG. 4 COVER PLATE - PACKED END

end. The group of 1/2 in. pipe holes were for the water outlet pipes.

(iii) Packed End

The detail for the packing gland follower is illustrated in Fig. 5. It was fabricated by welding 3/4 and 3/8 in. plate. The outer ring of 14 holes was for bolting the follower to the evaporator collar. These bolts took the stress from the Freon pressure exerted on that end of the tube bundle. They were tightened little more than hand tight without any Freon in the system. Any excess tightening would result in bent evaporator tubes. The inner circle of 10 holes was for accommodating stud bolts used for drawing up the packing. The raised portion of the follower bore down on the surface of the packing when the stud bolts were tightened.

(iv) Tubesheets

The tubesheets for both ends of the tube bundle were very similar. The detail of the tubesheet for the unpacked end is indicated by Fig. 6. It was fabricated from 1 in. steel plate. Ten of the outer ring of threaded holes were for the follower stud bolts. The other six threaded holes were for the cover plate bolts. The center group of holes were for the evaporator tubes. These holes were drilled and reamed to 0.758 ± 0.002 in., and grooved with two grooves approximately 1/8 in. wide by 1/64 in. deep. The bevelled edge of the tubesheet was to provide a space for packing and to give it a wedge-type action when being drawn up.

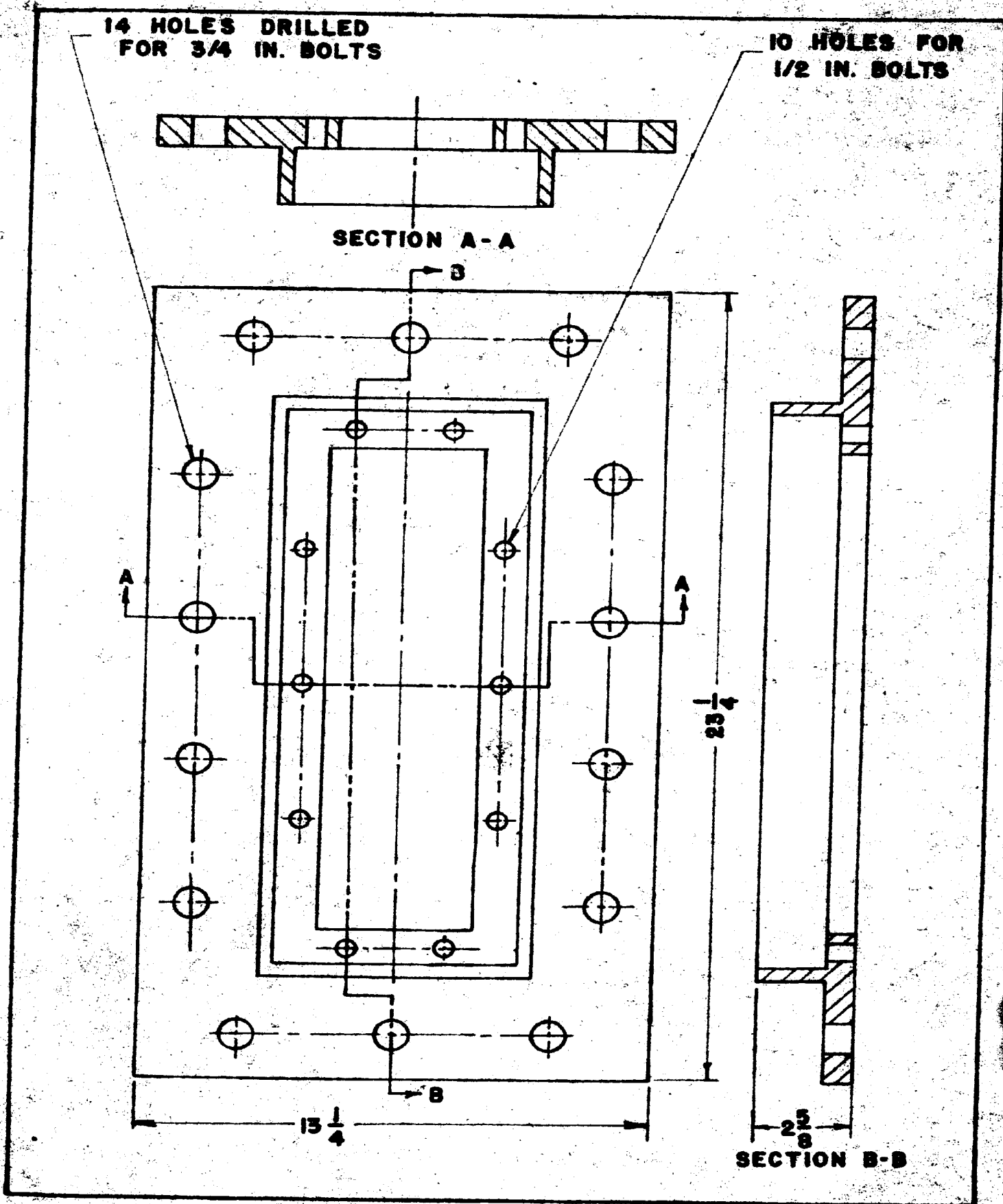


FIG. 5 PACKING FOLLOWER DETAIL

The other tubesheet had a 1/2 in. pipe hole for connecting the Freon gas injector beneath the bottom tube. It did not require bevelled edges nor holes for the stud bolts.

(v) Tube Bundle

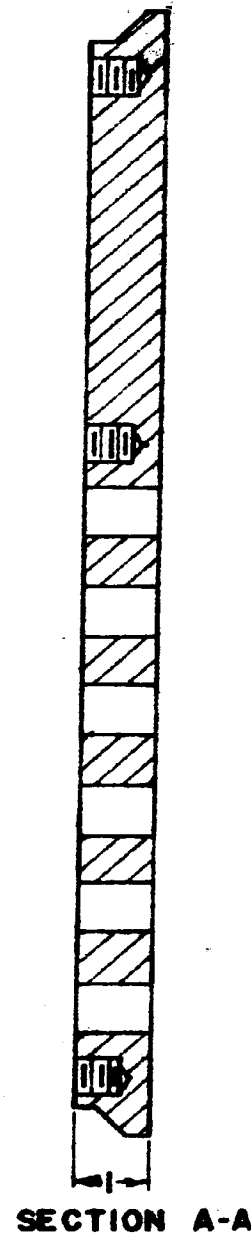
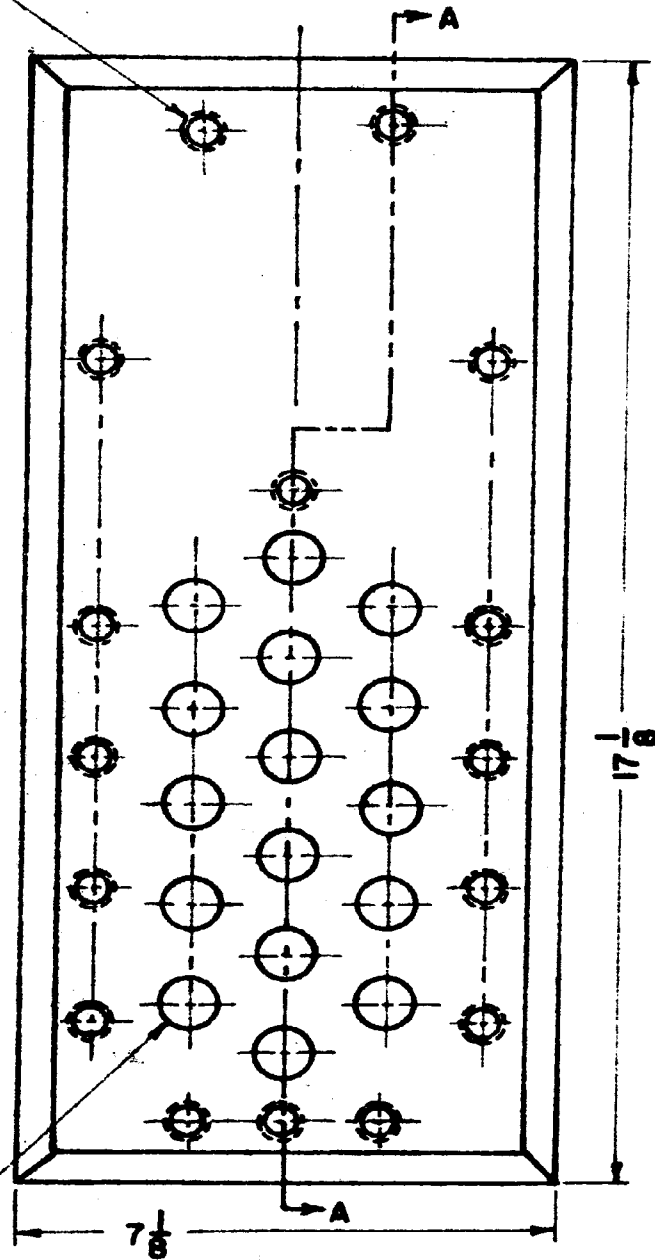
The tube bundle contained 16 copper tubes rolled into the tubesheets with a suitable roller. The tubes had a 0.750 in. O.D., and a wall thickness of 0.065 in. This gave a 0.620 in. I.D. which compared favorably with the 0.622 in. I.D. of the 1/2 in. pipe used for the water connections. The distance between tubesheets was 36.2 in. giving an area per tube of 0.593 sq. ft.

The only tubes used for the experiments were the bottom four in the middle row. The side rows were to give strength to the bundle, to provide a flow pattern for the bubbles which would approximate that in a commercial unit, and to permit further study on a more compact bundle of tube rather than a single row. Fig. 7 is a photograph of a tube bundle taken before thermocouples were installed.

Water connections were completed to the six tubes in the center row only but corresponding adapting holes were made for the bottom two tubes in each of the side rows.

The perforated vapor injector was made from a piece of 1/2 in. galvanized pipe threaded into the tubesheet so that it was 1/2 in. from the bottom tube. Twenty-

16 HOLES 3/4 IN. DEEP
DRILLED AND TAPPED
FOR 1/2 IN. BOLTS



16 HOLES DRILLED AND
REAMED 0.758 ± 0.002 IN.
ON 1-1/2 IN. CENTERS

FIG. 6 TUBESHEET DETAIL - PACKED END

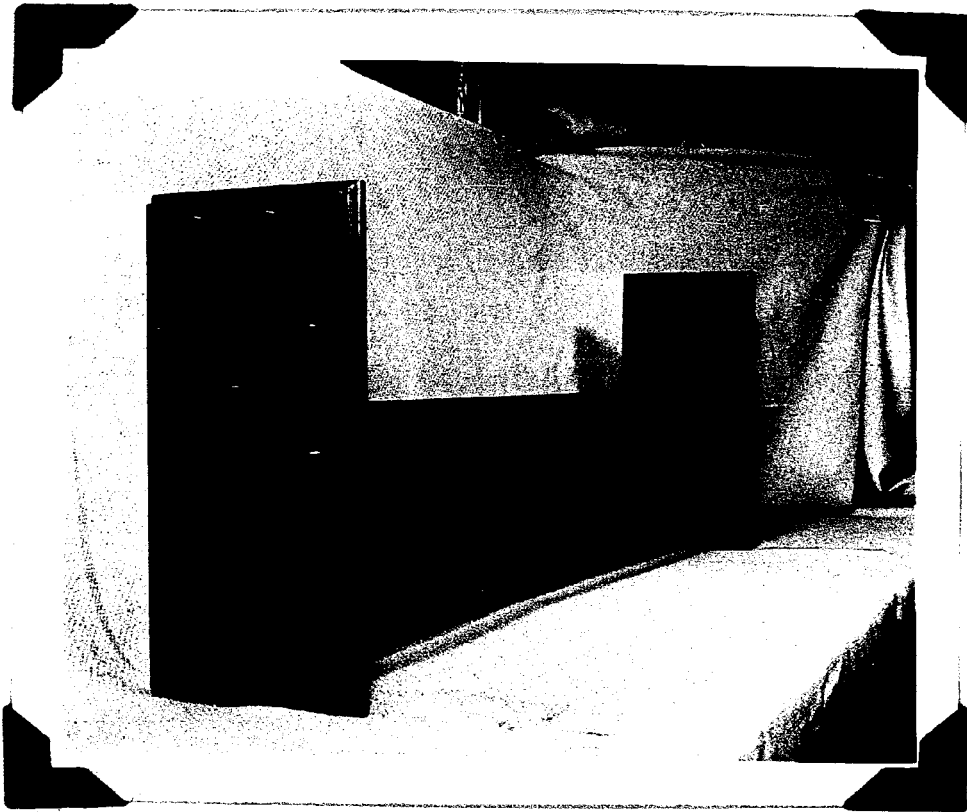


FIG. 7 PLAIN TUBE BUNDLE

three holes 0.040 in. in diameter were drilled 1-1/2 in. apart in a straight line along the top of the pipe. The vapor was directed through these holes against the bottom tube. The outlet end of the pipe was capped.

(vi) Packing and Gasketing

The gasket material used between the cover plates and tubesheets was compressed neoprene - asbestos sheet 1/8 in. thick. One of the gaskets is plainly visible in the photograph of the tube bundle in Fig. 7. The same material was used between the cover plate and the collar of the evaporator on the unpacked end.

Circular graphited asbestos packing was used to make the seal between the tubesheet and the shell on the packed end. First of all a piece of 1/16 in. packing was tamped into the bottom of the groove, making sure that the corners were particularly well filled, and that the ends were smoothly joined. This piece of material actually made the seal, the remainder of the packing just held it in place. On top of this first piece about 3 or 4 layers of the same type of material 1/4 in. in diameter were wound, making sure that they were well packed into the corners. On top of this was placed a piece of 5/8 in. diameter, 1/8 in. wall, neoprene rubber tubing. This just fit snugly in place and held there of its own accord, permitting the follower to be put on and bolted down.

(vii) Pressure Test and Leak Detection

Before piping connections were made, the outlets in

the evaporator shell were plugged with pipe plugs and the unit pressure tested to 225 p.s.i.g. with water. This proved the strength of the shell, the effectiveness of the packed end, and the tightness of the rolled tube joints.

After the piping connections had been made, the unit was tested for major leaks by placing it under about 75 p.s.i.g. air pressure and using soapy water. Smaller leaks were found using a methyl alcohol Freon detector with Freon in the unit.

(c) Compressor and Accessories

The compressor was a 5 ton Carrier type 5F-30 model, operating three cylinders at 1750 r.p.m. from a 5 h.p. drive. It was equipped with an automatic low and high pressure shut-off switch, and an automatic oil safety switch which shut off the compressor whenever the difference between the oil and suction pressure became less than about 45 p.s.i.g, generally indicating a loss of oil from the crank case. Gauge I, Fig. 1, indicated the discharge and gauge II the suction pressure of the compressor. Gauge III indicated the oil pressure.

The oil separator was an Aminco Refrigeration Products Co. type 890F separator which worked very satisfactorily when the compressor was operated properly.

The condenser was a water cooled multipass type, containing finned tubes. It had a purge cock for eliminating air from the system. The condensing pressure was indicated by gauge IV.

(d) Vapor-Liquid Separator

The purpose of the vapor-liquid separator was to separate the low pressure liquid from the vapor. It was fabricated from a piece of 6 in. pipe 20 in. long, with 3/4 in. steel heads welded in. A 1 in. pipe coupling was welded in the bottom for the liquid outlet. Two 1-1/2 in. couplings were welded in the top to provide an inlet for the throttled vapor liquid mixture and an outlet for the separated vapor. Two 1/2 in. couplings were welded in the side to make connections for a sight glass to permit observation of the liquid level. A thin steel baffle plate was ~~suspended~~ from the top end between the 1-1/2 in. couplings. It extended all the way across and was 5 in. long. It was designed to prevent as much liquid entrainment as possible.

(e) Storage Tank

The purpose of the storage tank was to hold the Freon during any changes in equipment that might be necessary during the course of operations, or during the changing of tube bundles. It was fabricated from a piece of 12 in. pipe 54 in. long, with 3/4 in. steel heads welded in. It had a 3/4 in. coupling in the top through which the evaporator could be drained. It had a 1/2 in. coupling in the top for a relief valve and connections on the end for a sight glass in which to observe the liquid level. A 1-1/4 in. pipe connection in the bottom permitted discharge of the liquid to the evaporator.

2. WATER CIRCUIT

The water and its auxiliary circuit consisted of three main units in addition to the piping and storage tank. These units were a pump, a heater, and two manifolds. Their arrangement is indicated in the flow sheet of Fig. 8.

The discharge from the pump divided into two streams, one going to the manifold and the other being by-passed to the suction side of the pump for capacity controls. The quantity by-passed was controlled by valves A and B.

The main stream was distributed to any of the desired six tubes by adjustment of the valves C. After passing through the evaporator into the downstream manifold, the streams were reunited and sent to a storage tank. The pump took suction from the storage tank through a small steam heater.

Steam was supplied from a 120 p.s.i.g. line. Its flow rate was controlled by 1/4 in. valve D and 1/8 in. valve E. Condensate was discharged from the trap to a drain.

(a) Pump

The pump was a Viking type gear pump driven at 300 r.p.m. by a 3 h.p. motor. It was capable of delivering approximately 65 g.p.m. at a pressure of 25 p.s.i.g.

(b) Heater

The steam heater was a Ross type BCF water heater.

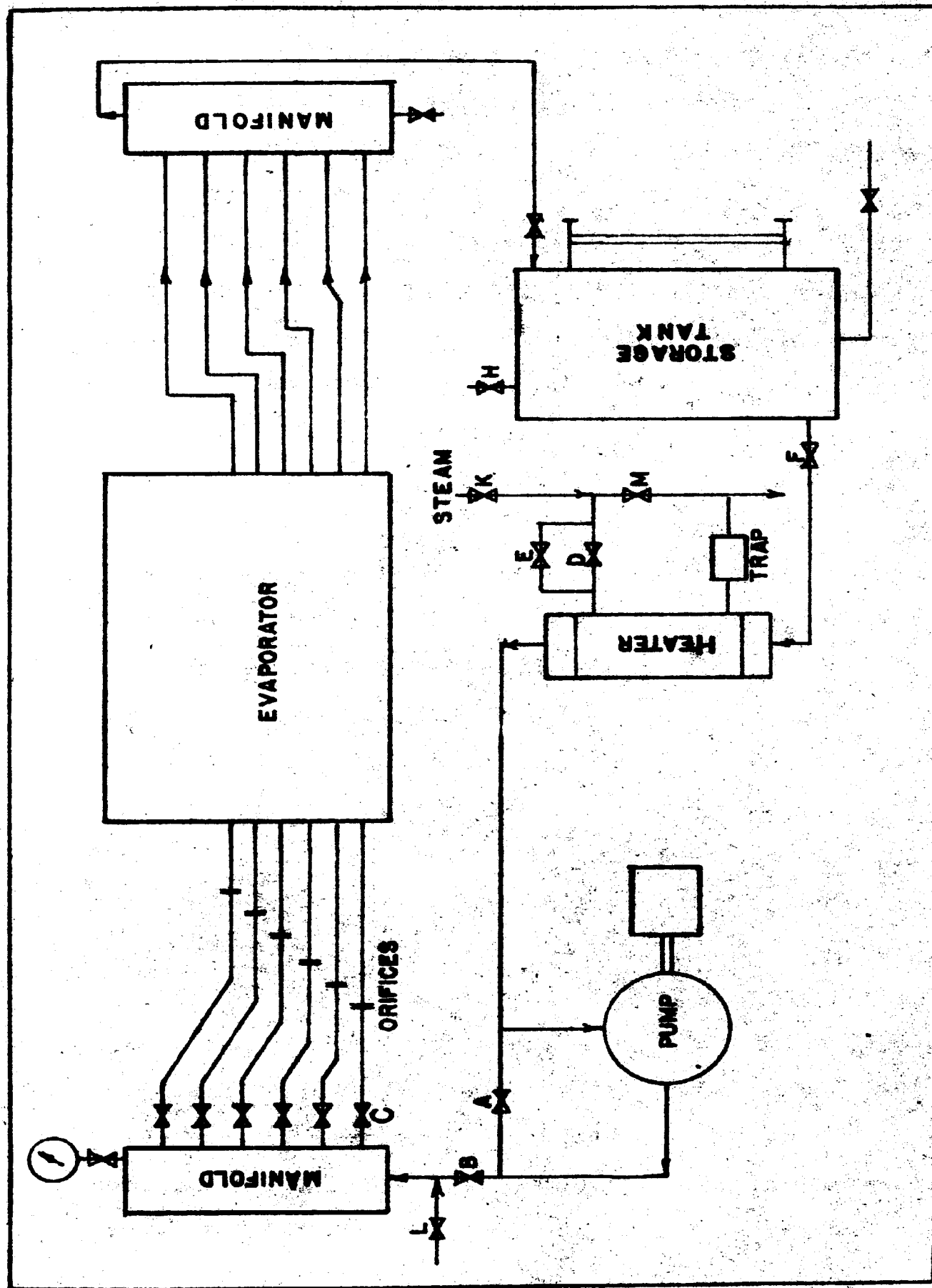


FIG. 8 FLOWSHEET OF WATER CIRCUIT

(c) Manifolds

The manifolds were fabricated from 6 in. pipe with 1/4 in. steel heads welded in. They were each 22 in. long and had six 1/2 in. pipe outlets evenly spaced on 3 in. centers on one side.

The upstream manifold had a 2 in. inlet in the bottom on the opposite side to the 1/2 in. outlets. A vertical partition consisting of a steel plate drilled with 3/4 in. holes extended all the way across and up from the bottom in the center to a height of 18 in. This was to give the water a more uniform velocity profile as it approached the 1/2 in. pipes, and to avoid a cross-sweep action which might have influenced the steadiness of flow in the individual tubes. The top of the manifold had a pressure gauge connection.

The downstream manifold had a 2 in. outlet at the top and a drain in the bottom.

(d) Piping and Storage Tank

All main piping was 2 in. galvanized pipe. The individual pipes leading to the evaporator tubes were 1/2 in. galvanized.

The storage tank was a 16 in. pipe about 5 ft. long with heads welded on both ends, suitable inlet, outlet, and drain connections, and a sight glass on the side. This tank was originally included as storage for the carbon tetrachloride but was useful with the water because it permitted more hold-up in the system and consequently slower fluctuation in water temperature.

3. INSULATION

All water circuit and all low pressure Freon circuit units and piping were insulated.

(a) Evaporator

The evaporator was covered with 3 in. slabs of Johns-Manville rock cork insulation on all sides. This filled in the space between the 3 in. I-beams. A layer of Johns-Manville hair felt one inch thick was then wrapped around the whole unit and tied in place with twine. This was followed by winding around strips of 4 in. Johns-Manville Zero-Tape, overlapping on every turn to insure a complete seal against moisture. The edges were carefully tucked in around all fittings and the whole was then painted with two coats of black asphalt paint to keep out moisture.

(b) Piping

All piping on the water and low pressure Freon circuits was insulated with two layers of Johns-Manville hair felt wrapped on tightly, tied with twine, and then wrapped with overlapping layers of the black tape. This was then painted the same as the evaporator. The liquid line from the vapor-liquid separator was wrapped with 3 layers of felt to keep the heat gain at an absolute minimum.

The manifolds, water storage tank, and vapor-liquid separator were wrapped with layers of hair felt and treated for moisture in the same way.

The 1/2 in. water piping was wrapped with two layers of hair felt and tied on but not water-proofed because it

had to be removed every time the tube bundle was changed. The ends of the evaporator were wrapped in the same way. Observation indicated that the amount of moisture which penetrated to these surfaces was very small and it was not felt sufficient to in any way destroy the insulating properties of the material.

B. MEASUREMENTS

In order to make the required heat transfer measurements it was necessary to know water inlet and outlet temperatures for each test tube, Freon liquid temperature, tube wall temperatures, and water flow rates in each tube. The temperature, pressure, and flow rate of the injected Freon gas also had to be known for those runs where gas was introduced.

1. FLOW MEASUREMENTS

- (a) Water Rate
- (i) Orifice Installation

Water flow measurements were made using orifices inserted in the half inch pipes. The assembly is indicated in Fig. 9. The plates were made of brass machined $3/32$ in. thick with a 0.344 in. sharp cornered hole drilled and reamed at the center. They fit into the gasketed seat of an orifice union. Orifice taps were taken off $3/4$ in. above and below the plates by drilling through the screwed ends of the union where they fastened onto the pipe with a $1/8$ in. drill. Three-sixteenth inch copper tubing connections were made from the orifice to a small manometer

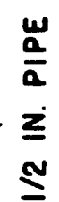


FIG.9 ASSEMBLY OF ORIFICE UNION

manifold made from 1-1/2 in. pipe 8 in. long having individual side outlets for each orifice. Each outlet had a brass stop-cock so that the flow rate in any desired orifice could easily be indicated on the manometer. The top of the manifold had an air purge cock and the bottom had a connection leading to the manometer.

The manometer fluid was mercury and the steel scale was graduated in mm. It was 130 cm. long.

(ii) Orifice Calibrations

The orifices were calibrated in place with water. Water from the city mains was admitted through the orifice at any desired constant rate and the time required for a given weight of water to be collected was recorded. Orifice constants were calculated from the recorded data. A plot of the calibration appears in Fig. 28 of Appendix C.

The constants for all orifices were practically identical so the same curve was used for all calculations. The high coefficient (0.748 to 0.765) is probably a result of the ratio between the plate thickness and the diameter of the hole which was 0.273. This would give more of a short nozzle effect than an orifice. The ratio of the hole diameter to the pipe diameter was 0.553. This ratio permitted reading flow rates up to 12 ft./sec. in the evaporator tubes.

(b) Freon Rate (i) Orifice Installation

The flow rate of the Freon gas being injected beneath the tube bundle was measured by an orifice in the 3/4 in.

copper tube. Manometer connections were similar to those used on the water orifices. The manometer fluids were glycol and mercury under Freon, and the scale was graduated in tenths of an inch. Glycol was chosen because of its density, 1.101 gm/cc. at 68 °F, and its extremely low vapor pressure which eliminated contamination of the Freon.

(ii) Calibration

The orifice was calibrated with air at room temperature. The calibration data are plotted in Fig. 29 of Appendix C. Air from a 90 p.s.i.g. source was passed through a Fischer-Porter critical flow prover, then through the orifice being calibrated. The reading on a mercury under air manometer was recorded for any given set of conditions at the flow prover. The corresponding quantity of air was calculated from the flow prover data and from this known quantity the orifice constant was evaluated. These constants were used without further modification for calculating rates of flow of Freon. The hole in the orifice plate was 0.281 in. and the plate was 3/32 in. thick.

2. TEMPERATURE MEASUREMENTS

(a) Water Temperatures

(i) Thermometer Installation

Beckmann thermometers graduated in 1/100 °C intervals were used for measuring water temperatures. One thermometer was installed in the fourth tube from the bottom at the inlet and one in each of the bottom four tubes at

the outlet. The inlet thermometer was 24 in. from the evaporator and the outlet thermometers were 30 in. from the evaporator. They were placed these distances to overcome mechanical difficulties adjacent to the evaporator where the 1/2 in. pipes were less than 3/4 in. apart and did not leave room for the 1 in. pipe tees in which the thermometers were located. Another reason for the location downstream was to provide sufficient mixing distance for the fluid leaving the evaporator before it reached the thermometer. In the bottom tube, which ran straight from the evaporator to the manifold, a mixing chamber was located ahead of the thermometer well. In all other pipes there were at least two elbows ahead of the thermometer. All piping was thoroughly insulated between thermometers in the manner previously described.

The method of installing the Beckmanns is illustrated in Fig. 10. A 3/4 in. nipple 3 in. long was screwed into a 1 by 3/4 in. bushing. A rubber stopper, bored to take the Beckmann was then inserted into the top of the nipple and the Beckmann pushed through. A 3/4 in. pipe cap, drilled to take the Beckmann, was then screwed down onto the rubber stopper to make it water tight. Another more loosely fitting rubber ring was then pushed into the space around the bulb of the thermometer where it emerged from the bushing. This acted as a spacer to give more rigidity to the thermometer and prevent lateral motion at the bottom.

This type of installation is recommended because it

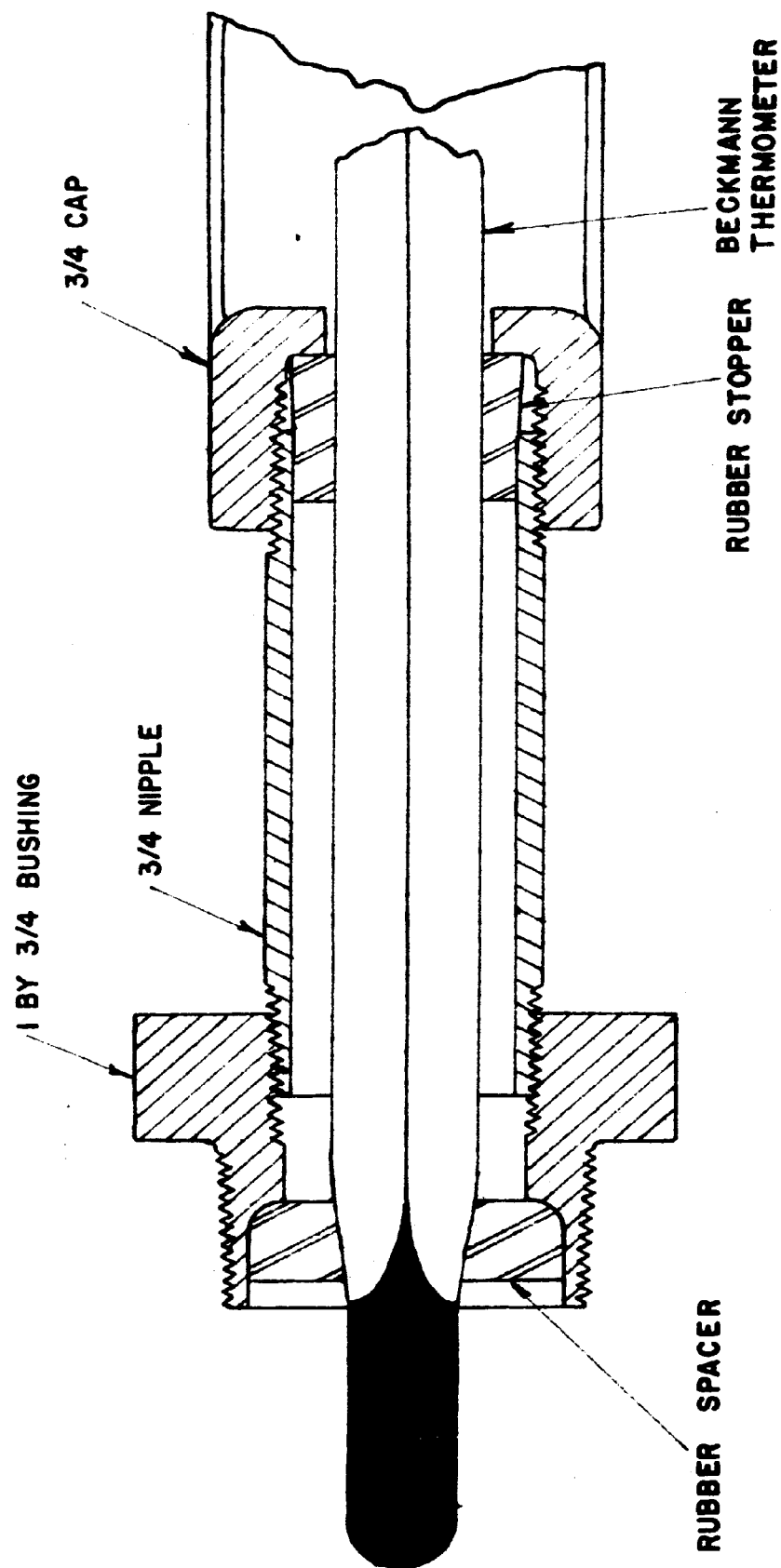


FIG. 10 INSTALLATION OF BECKMANN THERMOMETER

permits removal of the thermometer in its assembly whenever a new setting is required. This avoids the necessity of breaking a rubber to glass seal each time. Natural rubber was used for all the seals so that it could be dissolved out with benzene or carbon tetrachloride if the thermometers ever had to be removed.

A guard made from a half section of 1-1/4 in. brass tubing was brazed to the 3/4 in. cap. This was apparently effective and proved very handy when moving the thermometers around during resetting.

(ii) Calibration

A large well-circulated water bath was used for calibrating the thermometers. The bath was cooled down first of all with ice. The temperature was then maintained at any desired point on the Beckmann scale for the particular setting by recirculating ice water through a copper coil in the bath. At low temperatures, the heat input from the room and the stirrer could not be compensated for with ice water alone, so a brine solution was used.

For purposes of calibration, the thermometer used at the inlet was taken as the reference and all others were referred to it. It was referred to a N.B.S. thermometer graduated in 0.1 °C intervals to get the absolute temperature corresponding to the Beckmann reading. The method of taking the data was to hold the bath as close as possible to a fixed temperature and read the reference thermometer and each of the other thermometers simultaneously, making

four pairs of readings for the five thermometers. This was repeated for three rounds at each calibration point and the values averaged. These averages were subtracted to get the difference between each thermometer and the reference at the same temperature. A typical curve appears in Fig. 30 of Appendix C.

The true temperature corresponding to a reading on the reference Beckmann was also plotted for each setting that was used. One of these curves appears in Fig. 31 of Appendix C. Similar curves were used to calculate the inlet water temperature at each different setting.

(b) Freon and Tube Wall Temperature

The installation of the thermocouples used for measuring the tube wall and liquid Freon temperatures formed a separate and rather special phase of the work and is treated fully in Appendix A.

3. PRESSURE MEASUREMENTS

Pressure measurements were taken at the evaporator on gauge V, Fig. 1, for two reasons. They permitted a check on the thermocouple readings of Freon temperature, and they were used as a control during operation to tell whether or not conditions were steady. A calibration curve for this gauge appears in Fig. 32 of Appendix C.

A pressure gauge was also used below the orifice in the Freon gas line in order to record the pressure at which the gas left the orifice. A calibration for this gauge appears in Fig. 33 of Appendix C.

VIII. EXPERIMENTAL PROCEDURE

A. STARTING UP

1. WATER CIRCUIT

The water circuit was always started up first. Control valves F and B, Fig. 8, and vent H were opened wide. Water from the city mains was then turned on at valve L to fill the system. During the filling, the by-pass valve A and inlet valves C were closed so that the main 2 in. lines would be flushed out with fresh water. When the storage tank became nearly full the water was shut off and the tank allowed to drain empty. This eliminated rust which had collected in the system over the shut down period. The drain valve on the tank was closed and the system refilled with fresh city water to within a few inches of the top of the sight glass on the tank. The by-pass valve A and inlet valves C were then opened, the reversing motor switch put in the forward position, and the pump started. As soon as the pump took suction, the by-pass was adjusted until the manometer indicated about half-scale reading. Each orifice lead was then opened separately and the cock on the top of the orifice manifold opened to purge air from the lead lines. The cocks on the top of the manometer were then opened to purge air from the manometer lines. The control valves C were then adjusted until as nearly as possible the same rate of flow was obtained in each of tubes 1 to 4 (numbering from the bottom up). This involved closing the lower valves slightly and was a trial and error procedure. Valves in tubes 5 and 6 were left

closed except at low heat loads where the load on these tubes made control of the compressor easier. The by-pass valve was then adjusted to get the desired rate of flow in the individual tubes, and they were checked with the manometer to see if the rates were still the same.

It was extremely important that this procedure be followed closely, especially when the Beckmanns were set to read at 40 °F or lower. Unless the water at room temperature laying in the piping were purged thoroughly from the system before the pumps were started, the setting on the thermometers would be lost.

2. FREON CIRCUIT

The operation of the Freon circuit will be described assuming the Freon to be in the evaporator at the required level, about half way up in the top bull's eye. Additional operating instructions pertaining to other phases of the Freon circuit operation are presented in Appendix B.

Valves A, B, E, G, and Q, Fig. 1, were opened wide. Valve R on the liquid-control line to the by-pass was opened wide and the valve on the pressure gauge at IV was cracked. By-pass valve F was opened about half way. Valve H and all other valves were left closed tight. The importance of keeping valve H closed at this stage should be emphasized because if it were left open the liquid from the condenser would flow into the suction side of the compressor as soon as it were started up.

The compressor was then started. As soon as the condenser had accumulated some liquid, throttle valve C was

opened slightly to admit Freon to the separator. The condenser water would automatically start to flow when the pressure in the condenser had built up sufficiently. The liquid in the evaporator would start to foam and swirl over the top bull's eye as soon as C was opened. Valve H was then opened gradually until the by-pass pipe felt cold to the touch over most of its length. A hot by-pass pipe with H open usually meant that there was no liquid in the condenser. Pressure on the suction side could then be reduced by shutting down on valve F slightly until the suction pressure at gauge II started to decrease. The shut-off valve on gauge V was opened when the suction was less than 60 p.s.i.g.

This was a rather critical stage in the operation and the procedure here had to be followed carefully. The oil in the crankcase was saturated with Freon at the crankcase pressure. Any decrease in that pressure resulted in the oil becoming supersaturated with Freon at the new pressure. The sudden release of gas caused such violent frothing of the oil that it boiled right out of the crankcase and went over into the condenser and evaporator with the liquid. The oil separator could not handle such large volumes of oil and became ineffective whenever this frothing-over occurred. The way this could be avoided was to decrease the suction pressure very slowly and keep observing the oil in the sight glass on the crankcase. When the pressure was not being decreased too fast, the oil was creamy but not frothy. After stable operation at a fixed suction pressure, the oil became its natural color.

At this stage in the procedure, it was usually necessary to open the steam control valves E and D, Fig. 8, slightly in order to maintain the water temperature. The throttle valve C, Fig. 1, might also need adjusting to keep the desired liquid levels in the evaporator and condenser. The liquid level in the condenser was determined by locating the level where it was cold to the touch.

At this time the water circuit was operating, the steam was on sufficiently to hold the mercury on the scale of the thermometers, and the compressor was operating at approximately the desired suction pressure on direct expansion, that is, low pressure liquid and vapor were both going to the evaporator.

The next stage involved adjustment of a liquid level in the vapor-liquid separator so that only low pressure liquid went to the evaporator. This was done by opening valve D slightly and observing the sight glass closely. When Freon began to froth and foam in the glass, the valve was closed momentarily or at least far enough to prevent the liquid rising too far in the separator. This was an extremely important point because if liquid were carried over it would go directly to the suction side of the compressor. It was usually necessary to let the Freon foam up and down in the glass a few times in order to cool it down before a level could be held. Finally, the level was adjusted by control of valves C, D, H, and F. Opening valve C or the by-pass F tended to depress the liquid level. The level was also influenced by the adjustment of valve H,

but not radically. The level had to be watched very closely whenever other conditions were unsteady, and in fact, at all times.

At this point the liquid in the evaporator should have become relatively quiescent and will be lower than it was because of hold-up in the separator. In the rare cases where an extremely low heat load was used, valve A had to be partially closed before the pressure differences were such that liquid would appear in the separator.

The system was now ready for final adjustment to the exact conditions under which data were to be taken.

B. COLLECTION OF DATA

1. REGULAR RUNS

For the runs made to determine boiling coefficients for the individual tubes, nothing remained but to hold all conditions steady. The water velocity seldom if ever needed adjustment during the course of a run. It was attempted to hold the inlet temperature of the water constant within 0.03°C , and to keep the Freon temperature constant. A change in the Freon temperature was indicated immediately by a change in its pressure. A rise in pressure resulted from two things. The water temperature was either getting too high or not enough Freon was being sent to the evaporator. These conditions were corrected by proper adjustment of the steam or by-pass valves. It was very important to keep the Freon temperature absolutely constant because of the sensitiveness of the boiling coefficient to temperature difference.

As soon as conditions had been steady for about 15 minutes and usually about 1-1/2 hours after starting up, data were taken. The lower Freon liquid thermocouple was read, followed by the tube wall temperatures for tube 1. While the middle couple was being read, the inlet and outlet water temperatures were also read. This procedure was repeated for the other three tubes, with the upper Freon liquid couple being read for tubes 3 and 4. Three complete cycles of readings were made at each condition. The water manometer was read at the beginning and end of each run.

2. WILSON PLOT RUNS

During a series of runs for a Wilson plot, it was necessary to keep the heat load as constant as possible by leaving the compressor untouched during the entire course of data-taking for all the runs. The first run was made at the maximum water velocity attainable with the Beckmanns reading at the lower end of the scale. The water velocity was decreased for the next run and the inlet temperature of the water increased until the new larger overall temperature difference was sufficient to transfer the same amount of heat at the lower velocity and same Freon temperature. The limiting velocity was determined at the low end by the maximum temperature difference permissible with the Beckmann setting, and at the high end by the size of the orifices.

In all other respects the data taking for these runs was the same as described above.

3. INJECTION VAPOR RUNS

Runs taken with Freon gas being introduced beneath the tubes were essentially the same as the others except that the desired quantity of gas was introduced by control valves J and K, Fig. 1. No difficulty was encountered in keeping the flow rates constant. The temperature and pressure of the inlet gas were recorded when the manometer was read.

C. SHUTTING DOWN

At the conclusion of data taking, valve D, Fig. 1, and the steam valves were closed tightly. The by-pass valve F was opened slightly and the control valve H closed. Valves C and B were then closed rapidly and simultaneously and the compressor shut off. The shut-off valve on gauge V, the liquid line valve R on the condenser, and valve A on the evaporator were closed.

The water pump was then turned off and the system drained through the storage tank. Valves B and F, Fig. 8, were closed and a trickle of water turned on from the mains at valve L so that it would flow continuously through the tubes. This was done to prevent rust settling out on the evaporator tubes during the shut-down period and to keep the Beckmanns cool enough to hold their setting.

IX. TREATMENT OF EXPERIMENTAL DATA

A. INTRODUCTION AND PRELIMINARY ANALYSIS

When a relatively warm fluid is passed through a tube surrounded by another colder fluid, heat is transferred from one fluid to the other through the tube wall. It is possible to calculate an overall heat transfer coefficient for this process from an equation of the type

$$\frac{Q}{A} = U\Delta T \text{ - - - - - (2)}$$

where Q is the quantity of heat transferred in BTU/hr., U is the overall heat transfer coefficient in BTU/hr. sq. ft. $^{\circ}\text{F}$, A is the area in sq. ft. on which U is based, and ΔT is the bulk temperature difference between the two fluids in $^{\circ}\text{F}$.

The heat load Q under these conditions can be calculated from the relation

$$Q = WC (T_1 - T_2) \text{ - - - - - (3)}$$

where W is the weight rate of flow in lb./hr. per tube, T_1 , the inlet temperature, T_2 the outlet temperature, and C the specific heat of the fluid flowing in the tube in BTU/lb. $^{\circ}\text{F}$.

In order to determine the separate so-called film coefficients which make up the overall coefficient further information is necessary. This information can be in either of two forms. It can be in the form of additional experimental information or in the form of independent quantitative theoretical relationships which will permit calculation of the film coefficient from the known physical and

dynamic properties of the fluid under the conditions of flow in the tube. A known relationship for the inside of the tube would permit, under certain conditions, the calculation of the outside coefficient from the resistance concept, which expressed mathematically take the form

$$\frac{1}{U} = \frac{1}{h} + \frac{A}{A_i h_i} + \frac{XA}{kA_m} + r_f \quad - - - - - (4)$$

where h is the film coefficient on the outside of the tube in BTU/hr. sq. ft. $^{\circ}\text{F}$, A_i is the inside area of the tube in sq. ft., h_i is the film coefficient on the inside of the tube in BTU/hr. sq. ft. $^{\circ}\text{F}$, k is the thermal conductivity of the metal in the tube wall, BTU/hr. ft. $^{\circ}\text{F}$, x is the thickness of the tube wall in ft., A_m is the mean area of the tube wall, and r_f is the total resistance of the fouling which may or may not be present on either the inside and/or outside of the tube in $^{\circ}\text{F}$ sq. ft. hr./BTU.

The condition under which a measured value of U and a calculated value of h_i will permit the calculation of h is that r_f shall be negligible or known. Serious error may result if this condition is not fulfilled. Fig. 11 has been included to indicate the effect of even an apparently small amount of fouling. The carbon tetrachloride which was originally used as the heating fluid inside the tube attacked the copper and the steel and deposited a thin layer of powdery material on the inside of the tubes. The curves in the figure indicate the difference in overall coefficients obtained for these tubes before and after the deposit had been cleaned off.

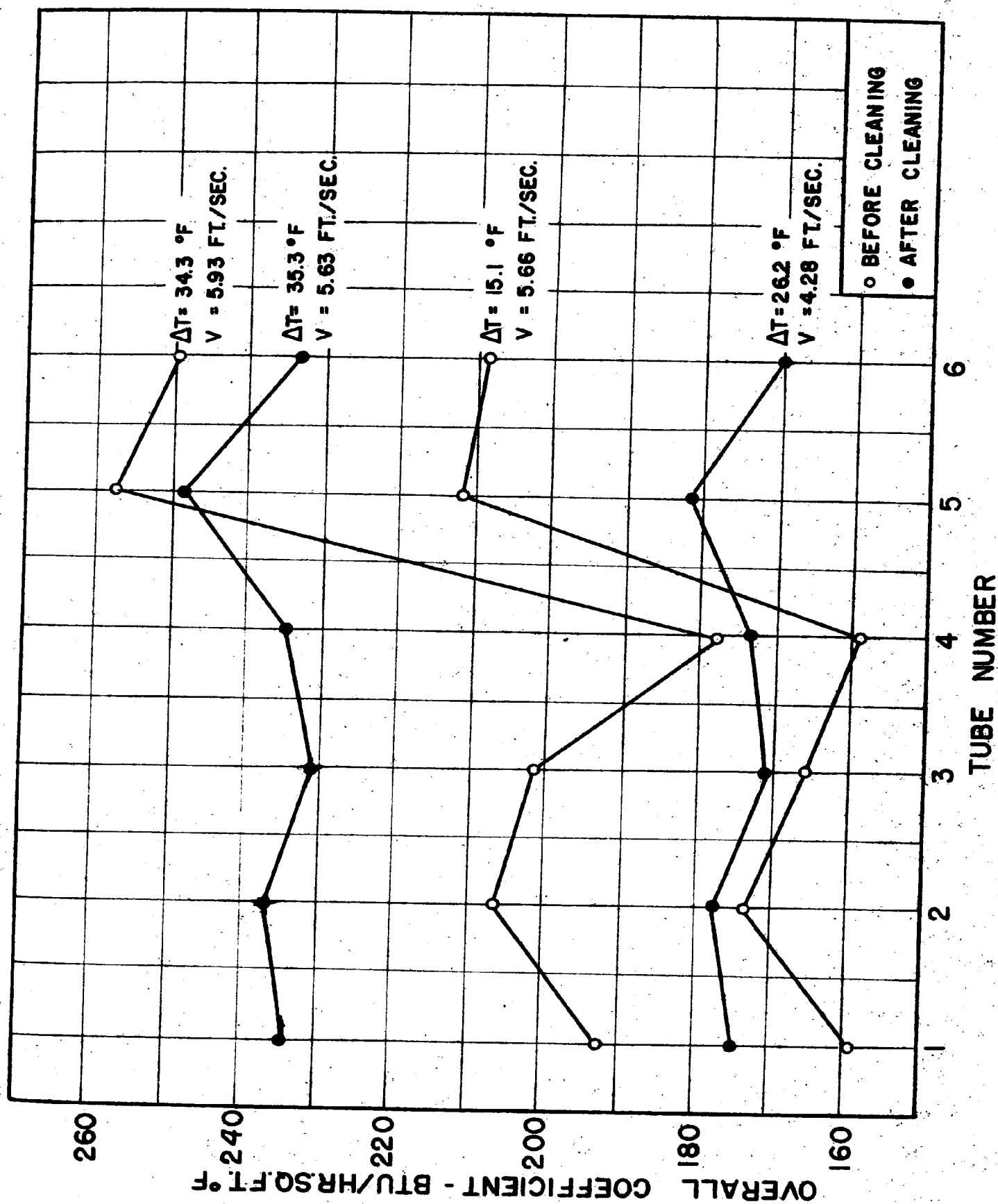


FIG. 11 EFFECT OF INSIDE FOULING ON OVERALL COEFFICIENTS

A relationship for calculating the coefficient on the inside of the tubes is known to be of the form

$$\frac{h_i D}{k} = B \left(\frac{DV\rho}{\mu} \right)^a \left(\frac{C\mu}{k} \right)^b \quad - - - - - (5)$$

where D = inside diameter of the tube, ft.

k = thermal conductivity of the fluid, BTU/hr. ft. $^{\circ}\text{F}$.

B = a constant depending largely on the geometry of the system, generally assumed to be 0.023 for long smooth tubes.

V = velocity in the tube, ft./sec.

ρ = density of the fluid, lb./cu. ft.

μ = viscosity of the fluid, lb./ft. hr.

C = specific heat of the fluid, BTU/lb. $^{\circ}\text{F}$.

a = a constant generally assumed to be 0.8.

b = a constant generally assumed to lie between 0.3 and 0.4, with 0.3 being recommended for cooling (8).

The ratio of length to diameter of the tubes in the evaporator was 58, which does not permit them to be considered as long smooth tubes with negligible end effects. Therefore it cannot be said that the constant B was known. This prevented a direct calculation of the inside coefficients even with the assumption that $r_f = 0$ and that the constant $a = 0.8$ and $b = 0.3$.

Due to the method of Wilson (16) this difficulty can be avoided under certain experimental conditions. The conditions are that h be a constant for a series of runs made at different velocities, that r_f be known, and that the

constant a be known accurately. For the case of boiling a given liquid on a given tube surface at a constant temperature, in the presence of no or constant external disturbances in the liquid, Eqn. (1) indicates that Q is a function only of the temperature difference between the boiling liquid and the outer surface of the tube wall. The reasoning for this is as follows:

$$Q = hA (\Delta T_b) \text{ ----- (6)}$$

$$\text{and since } h = E (\Delta T_b)^m \text{ ----- (1)}$$

$$\therefore Q = EA (\Delta T_b)^{m+1} \text{ ----- (7)}$$

If Q is held constant it follows that ΔT_b must be constant. If ΔT_b is constant, then h must be constant from Eqn. (1). Holding Q constant would therefore fulfill the first condition required to make a Wilson plot.

For copper tubes that have been thoroughly cleaned and used with non-corrosive liquids such as water it may be reasonably assumed that r_f equals zero. However, this condition was not fulfilled when carbon tetrachloride was used as the heating fluid and r_f was therefore an unknown, although probably nearly a constant for any given series of runs at constant heat load. The constant a was still an unknown but assumed to be 0.8.

Several series of runs were made using carbon tetrachloride inside the tubes and the data were analyzed by the Wilson plot method and also by using a relation of the type in Eqn. (4) and (5) with the accepted constants. The boiling coefficients obtained by the Wilson Plot

method were from 50% to 100% higher than those obtained by using Eqn. (4) and (5). Furthermore, when the boiling coefficients obtained by the equations were plotted as a function of the Reynolds number inside the tubes, they were observed to increase considerably as the Reynolds number increased at constant heat load. Theoretically the Reynolds number on the inside should have no effect on the boiling coefficient at constant heat load.

The conclusions drawn from these results were that the power a on the Reynolds number in Eqn. (5) was not 0.8 but some higher number, and that until this power were known accurately, neither the Wilson plot nor Eqn. (5) could be used to analyse the data to obtain boiling coefficients.

Accordingly, it was evident that the further information necessary to calculate boiling coefficients must be in the form of additional experimental information which would permit direct computation of boiling coefficients from the relationship expressed by Eqn. (6). Thermocouples in the tube wall gave the necessary data for this computation and also for the calculation of inside coefficients from the relation

$$Q = h_i \cdot A \cdot \Delta T_i \text{ ----- (8)}$$

Inside coefficients so calculated could then be used to determine accurately the power a on the Reynolds number in Eqn. (5). This power could then be used to make a Wilson plot analysis and check it with the coefficient ob-

tained by direct calculation. This is the method of analysis which was followed for the entire data which is presented in Table II of Appendix C.

B. BOILING COEFFICIENTS

1. EFFECT OF AGITATION

The effect of agitation was studied under two conditions. First of all the effect of the position of one tube relative to the others in the tube bundle was studied, and secondly the effect of agitation resulting from artificially introduced gas was investigated.

(a) Effect of Tube Position

The conditions under which these data were taken are as follows:

For any given setting on the Beckmann thermometers boiling coefficients were determined at various temperature differences for the four tubes by varying the evaporating temperature of the Freon, the water velocity, and the inlet water temperature over as wide a range as possible. Three different settings on the thermometers were used, the first reading between about 59° and 68° F, the second between 38° and 47° F, and the third between 68° and 79° F.

The boiling coefficients were calculated by using Eqn. (6). The calculated values appear in Table II of Appendix C. The heat load, Q , was obtained by use of Eqn. (3). The boiling temperature difference, ΔT_b , was obtained from the difference between the tube surface and Freon temperatures. The temperature of the tube wall was calculated by averaging the three thermocouple readings for each tube. The temperature

so obtained was assumed to be the temperature at the center of the tube wall. The temperature drop across the copper tube wall, ΔT_{cu} , was calculated for the given heat flux, and half of this drop added to the temperature obtained by the thermocouple readings. This temperature was then used as the surface temperature in evaluating ΔT_b . Half of the drop across the copper tube was subtracted from the thermocouple temperature to get the inside surface temperature for evaluating ΔT_i .

In the bottom tube, the thermocouples were placed on the bottom, side, and top. The thermocouple in the top read up to 4% higher than the one in the bottom, and the one in the side read up to 4% lower than the bottom. However, the individual readings of thermocouples in the tubes where the junctions were all at the bottom differed by as much as $\pm 4\%$ from the average. Therefore it was not felt justifiable to make any definite statement regarding the temperature gradient which may or may not exist around a tube wall surrounded by a boiling liquid. For this reason no correction was applied to the thermocouple readings for tubes 2, 3, and 4 and it was assumed that the average tube wall temperature was given by the thermocouples which were all in the bottom of the tube.

The coefficients for each tube were plotted as a function of the temperature difference across the boiling film as in Fig. 12. This plot includes the data for runs 103-127 inclusive and run 180. These runs were selected for studying the effect of tube position for reasons outlined in the discussion on surface stability.

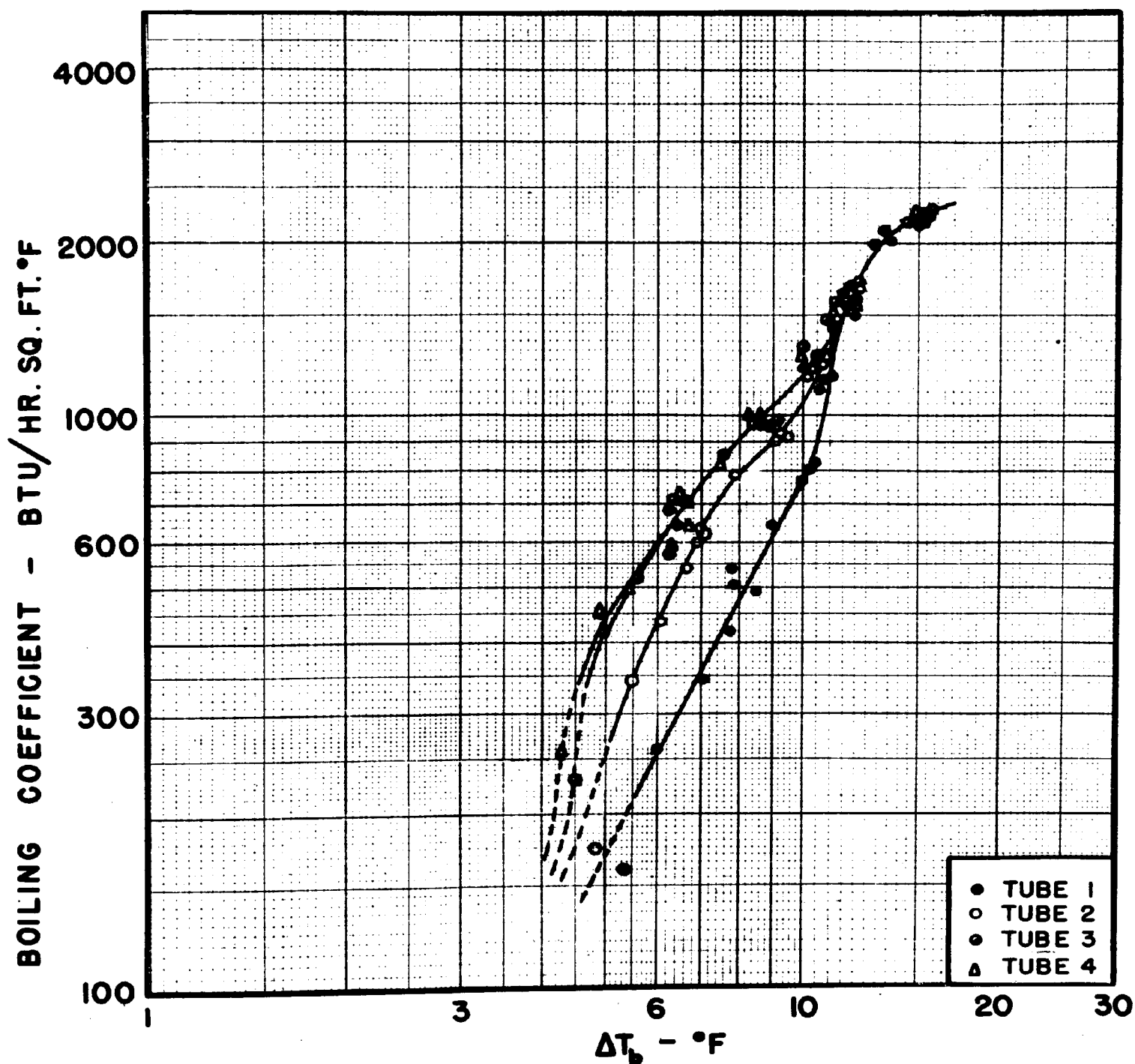


FIG. 12 EFFECT OF TUBE POSITION ON BOILING COEFFICIENTS

Run No. 180, taken on June 23, requires further explanation. It was made while using a small 1/4 ton Freon compressor. The capacity was insufficient to keep a steady state in the evaporator because the heat input from the pump and the surroundings was greater than the heat that could be removed by the compressor. Consequently, the Freon and water temperatures both gradually increased during the course of the data taking which lasted about half an hour. The inlet water temperature changed from 61.72 to 62.35 °F, and the Freon from 54.60 to 55.60 °F. It is understandable that the data may be in considerable error when it is remembered that the heat load per tube was only about 575 BTU/hr., the water temperature drop as low as 0.35 °F, and the measured temperature differences from 4.4 to 5.4 °F. The data for this run have been included, however, because they follow the trend that would be expected at very low temperature differences. The curves have been dashed where they pass through the points for this run.

The influence of one tube on another is clearly shown by the spread in the coefficients obtained for different tubes. At the high temperature differences, where agitation of the liquid is relatively high, the effect of one tube on another is negligible. At medium temperature differences, the vapor rising from the bottom tube creates additional agitation around the second tube which results in a higher coefficient. The combined vapor from tube 1 (bottom tube) and 2 effects tubes 3 and 4 in a similar way. No significant difference is observed between tubes 3 and 4 in this region. Apparently the effect of the agitation

had reached an optimum for the tube arrangement by the time the third tube had been reached. Beyond that the beneficial effect of increased agitation was perhaps countered by the detrimental effect of the surface being partially blanketed off by bubbles of vapor. At the lower temperature differences, the amount of vapor rising from the bottom tube is so small that it has a diminishing influence on the coefficient for tube 2. On the other hand the combined vapors from tubes 1, 2, and 3 are sufficient to increase the agitation around tube 4, but not to blanket it off. It was only at the very low temperature differences that any significant difference was observed between tubes 3 and 4.

At a constant ΔT_b of 6 °F, it may be seen that the coefficient for tube 2 is about 65% higher than for tube 1 and the coefficient for tubes 3 and 4 is about 125% higher than tube 1. A plot could be made showing the percentage increase in coefficient for the different tubes at a series of temperature differences. However, this would have little significance with regard to the performance of commercial equipment where the evaporating temperature, water inlet temperature, water velocity, and consequently the inside coefficient for each tube would be approximately the same. If for some reason an external influence such as vapor agitation tended to increase the outside coefficient for a particular tube, then the ΔT_b for that tube must decrease. This follows from the fact that as Q/A increases ΔT_b must increase; but if ΔT is constant, then ΔT_b must decrease. Therefore there is no use comparing the effect of vapor agitation on boiling

coefficients for individual tubes in a commercial type unit on the basis of the same ΔT_b .

The order of magnitude of the change in ΔT_b for different tubes may be seen by referring to Fig. 12. This point may be emphasized by referring to the data for run No. 127 where the effect of tube position was a maximum. The coefficients for tubes 2, 3, and 4 showed an increase of 37, 61, and 69% over the coefficient for tube 1. However, the boiling temperature difference decreased for each tube, with the net result that the heat flux only increased by 21.8, 33.5, and 41.6% respectively.

(b) Artificially Injected Vapor

The first data taken to study the effect of introducing vapor beneath the tube bundle were runs 128-131. Stable conditions were reached in the system and measurements made on all four tubes. Vapor was then injected beneath the bundle, with the water inlet temperature, evaporating temperature, and water velocity held constant. Measurements were again made on all tubes. This process was repeated for three different gas rates.

The volumes of gas injected in each run were calculated from the data taken at the Freon orifice. The volume of gas flowing through the orifice at the downstream conditions was calculated from the density data for Freon and the calibration curve in Fig. 29 of Appendix C. Since the gas came from the compressor it was superheated relative to the vapor in the evaporator. Each pound of injected gas would result in slightly more than one pound of vapor generated in the

evaporator because the desuperheating would evaporate some liquid. The calculations of the volumes of vapor generated in the evaporator are shown in Table III of Appendix C.

Fig. 13 indicates the relative values of the boiling coefficients for the separate tubes for different gas rates. The rates of gas injection and boiling temperature differences for the bottom tube are indicated on each curve. The boiling temperature difference was not constant but decreased for the reasons outlined above from 8.06°F at no gas flow to 5.95°F at maximum gas flow.

Another series of data was taken in a similar manner but starting with an initially higher boiling temperature difference and using more gas rates. The different values for these runs No. 132-137 are indicated in Fig. 14. The temperature difference changed from 8.65 to 6.99°F as gas was introduced. A third series at still higher boiling temperature differences was taken. The values are shown in Fig. 15.

The coefficient for the bottom tube is seen to have increased considerably as gas was introduced, while the coefficients for the other tubes generally decreased. This is probably because the injected vapor partially blanketed off part of the surface without materially increasing the agitation. The method of introducing the vapor beneath the tube might have deflected the bubbles off the tube in such a way that no beneficial effect resulted on the tubes higher in the row.

An attempt was then made to operate the equipment in

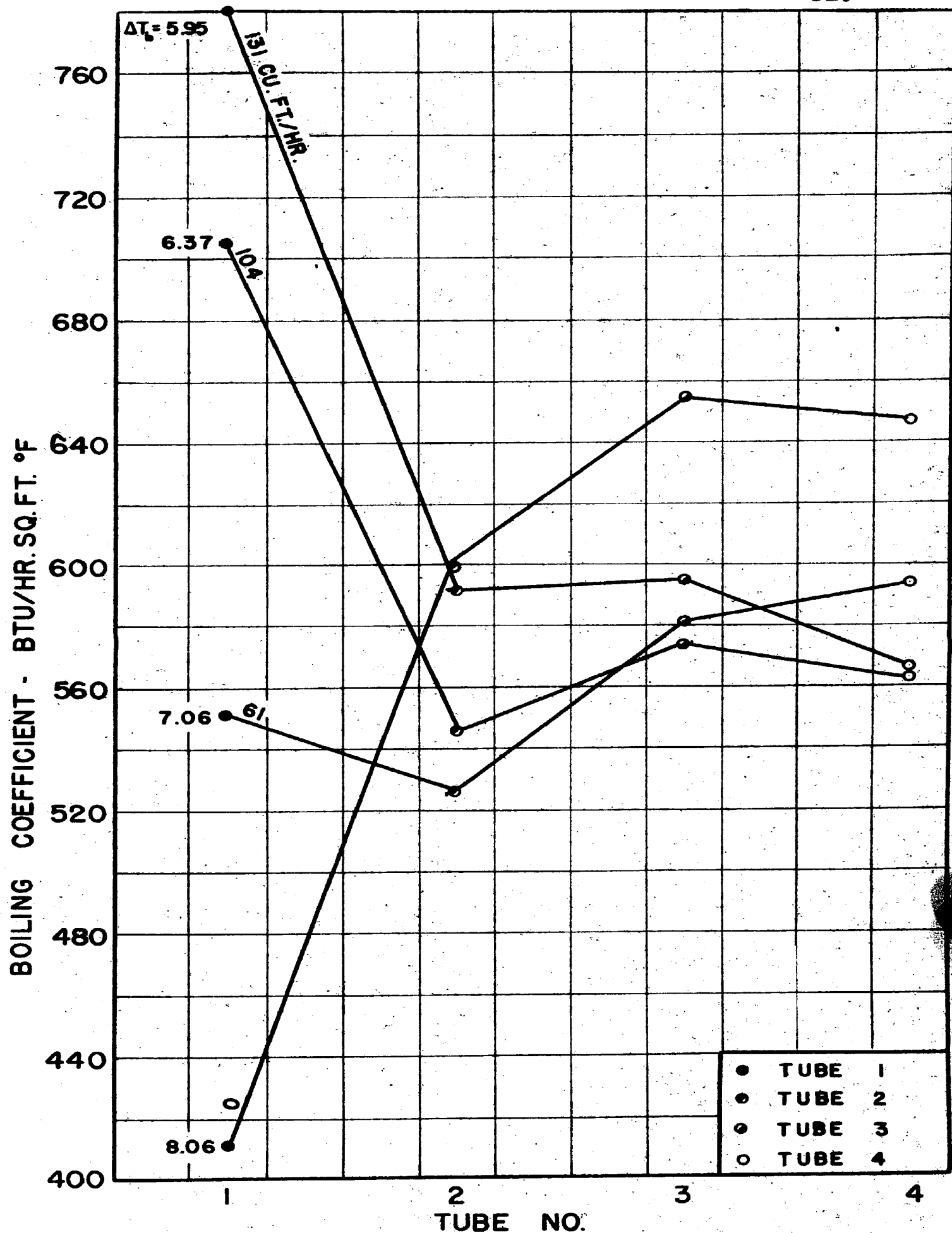


FIG. 13 EFFECT OF VAPOR INJECTION ON BOILING COEFFICIENTS — RUNS 128-131

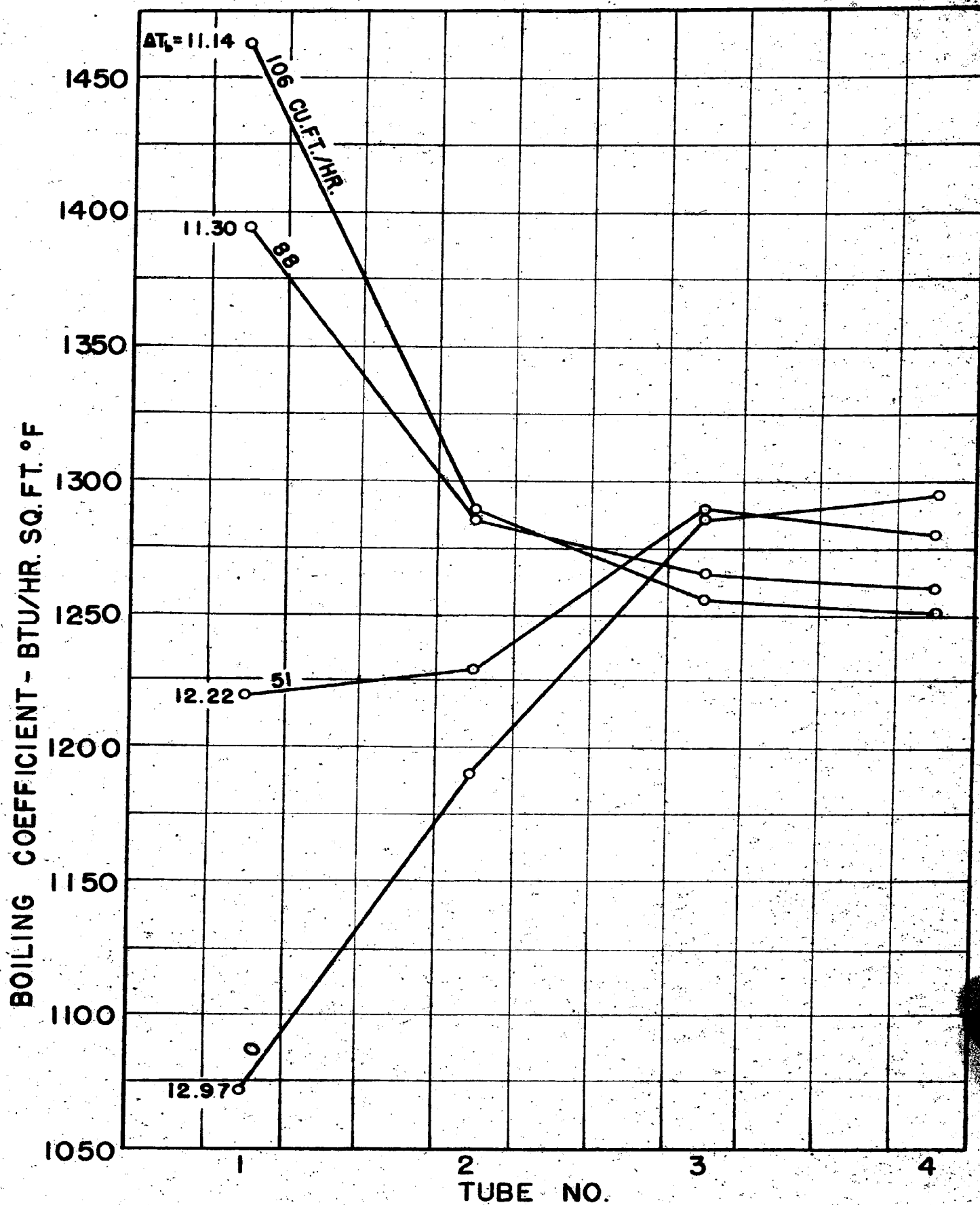


FIG. 15 EFFECT OF VAPOR INJECTION ON BOILING COEFFICIENTS - RUNS 140-143

a way that would permit study of the effect of vapor injection on the coefficient at constant temperature difference. These runs were No. 151-153. In this way it was felt that the effect of vapor injection alone would be shown. However, difficulty in getting the same temperature difference at different conditions and gas rates made this type of data too difficult to take. Calculations for the attempted runs appear in Tables II and III of Appendix C.

The final and most successful series of runs with vapor injection were made as follows. Five different rates of injection were used for four different initial conditions. The inlet water temperature, Freon temperature, and water velocity used for the initial conditions were held constant for each of the five different rates of injection. Measurements were only made on Tube 1. Glycol was used in the manometer for the three lower rates and mercury for the two higher rates of injection. Only three rates were used at the highest initial boiling temperature difference. The data for the two higher rates were taken 24 hr. after the data for the low rates. The boiling temperature differences for each of the four initial conditions covered as wide a range as possible.

A plot of the data for runs No. 154-175 appears in Fig. 16. The plot is boiling coefficient versus ΔT_b and the parameter is rate of injection of gas. The effect of the injected gas on the boiling coefficient is seen to be

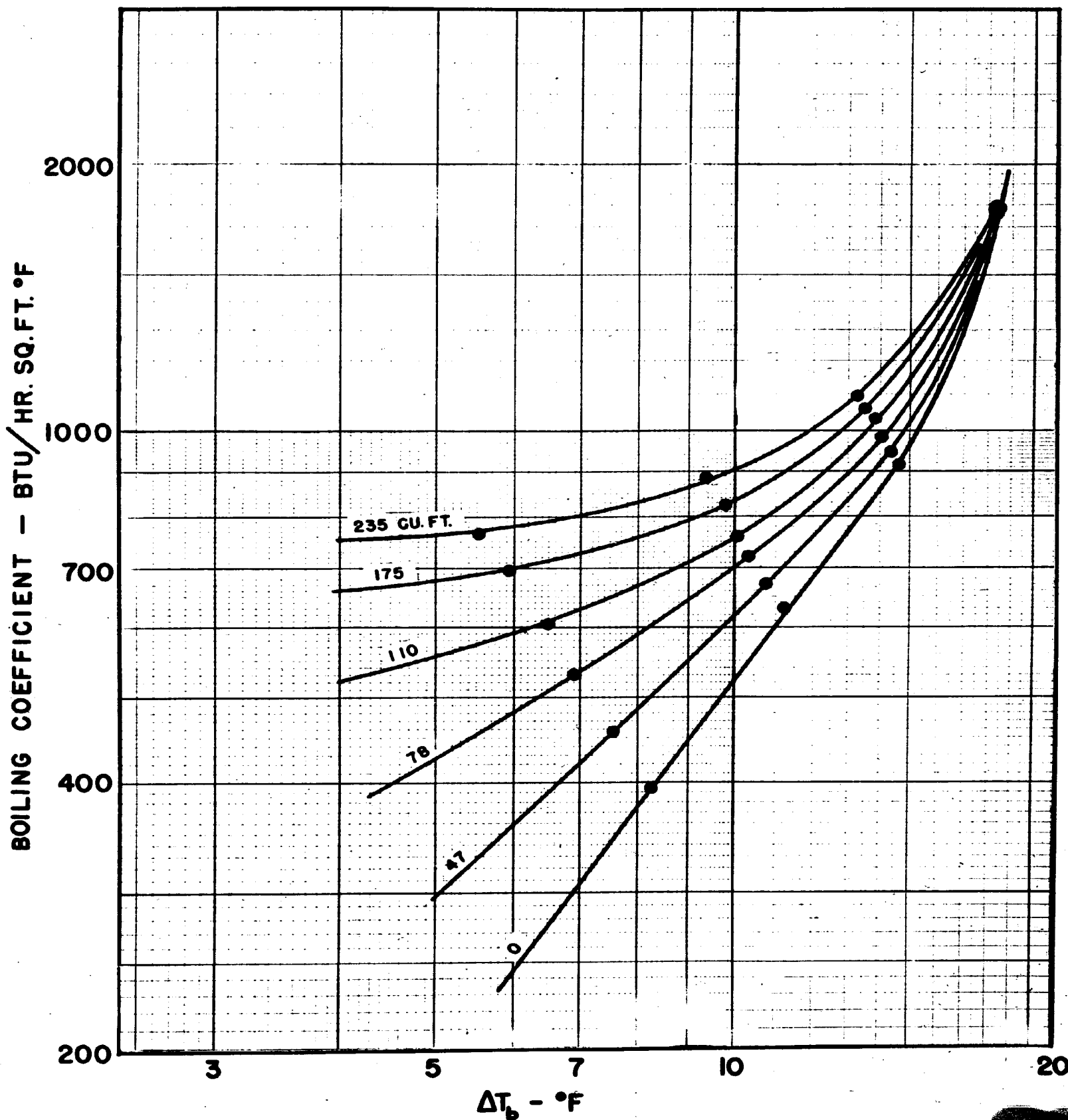


FIG. 16 EFFECT OF VAPOR INJECTION ON COEFFICIENTS FOR A SINGLE TUBE

very large at low temperature differences and to drop off to zero at higher temperature differences. This is in agreement with the previously observed behavior for the case of the four tubes in a vertical row where no difference between coefficients was observed for different tubes at high temperature differences. The same explanation would also apply.

Fig. 17 is a plot of boiling coefficient versus rate of injection at constant ΔT_b obtained by cross-plotting the data of Fig. 16. The calculations appear in Table IV of Appendix C. Each curve at constant boiling temperature difference is apparently asymptotic to some upper limit but it is not possible to say for sure whether or not the limiting coefficient of about 1850 BTU/hr. sq. ft. $^{\circ}\text{F}$ would be reached in all cases. In fact it appears as though some limiting value would be reached for each temperature difference beyond which no further increase in agitation would cause the coefficient to increase. This would be contrary to the work of Austin (3) for purely mechanical agitation. It would establish ΔT_b as a fundamental variable which affects the boiling coefficient. If, on the other hand, the limiting value is 1850 BTU/hr. sq. ft. $^{\circ}\text{F}$ for the particular liquid and surface for all temperature differences, then temperature difference is shown to be purely a measure of the state of agitation of the liquid and not a fundamental variable.

The same remarks which were made previously on the significance of constant boiling temperature difference

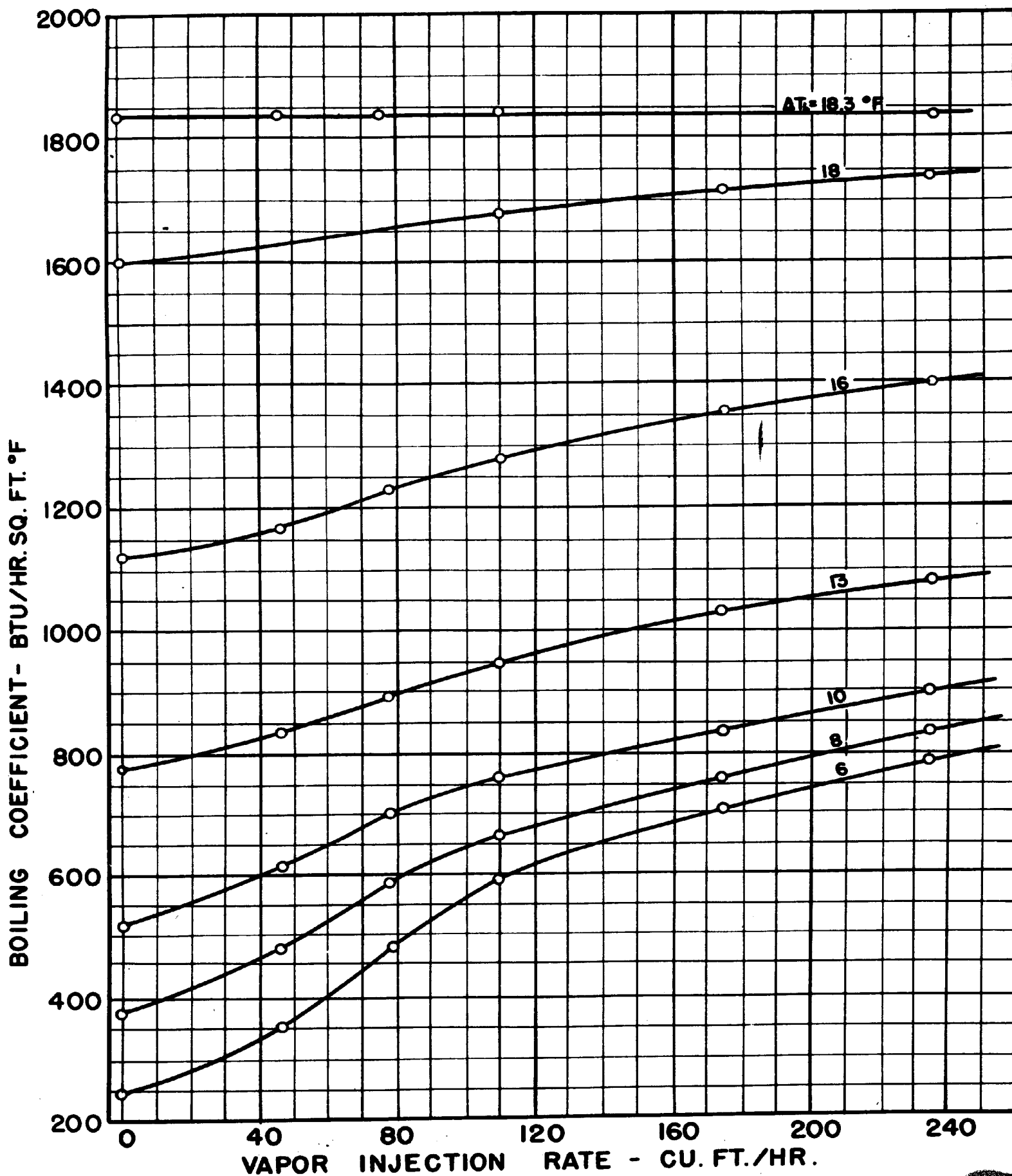


FIG. 17 BOILING COEFFICIENT AS A FUNCTION OF VAPOR INJECTION RATE

in a commercial unit still hold. In the present case, however, the plot at constant temperature difference is significant in that it permits to some extent a study of the fundamental variables affecting the boiling coefficients.

Fig. 18 is a plot of the percent increase in the coefficients resulting from the introduction of gas versus the rate of injection at constant boiling temperature differences. The calculations appear in Table IV of Appendix C. Also included in this plot are three points taken from Fig. 12. The volume of vapor generated by tube 1 was calculated at boiling temperature differences of 6, 8, and 10 °F. The percent increase in the coefficient for tube 2 was also calculated. These points were then plotted on Fig. 18 to compare the effectiveness of the naturally generated vapors with the artificially introduced vapors in increasing the coefficient. It is seen that the naturally generated vapors have caused a greater percentage increase in coefficient at the same boiling temperature difference. From this it may be concluded that the manner in which the vapors strike the tube is significant.

2. EFFECT OF SURFACE CONDITIONS

After about half the data had been assembled it became apparent that the condition of the surface of the copper tubes was having a pronounced effect on the coefficients.

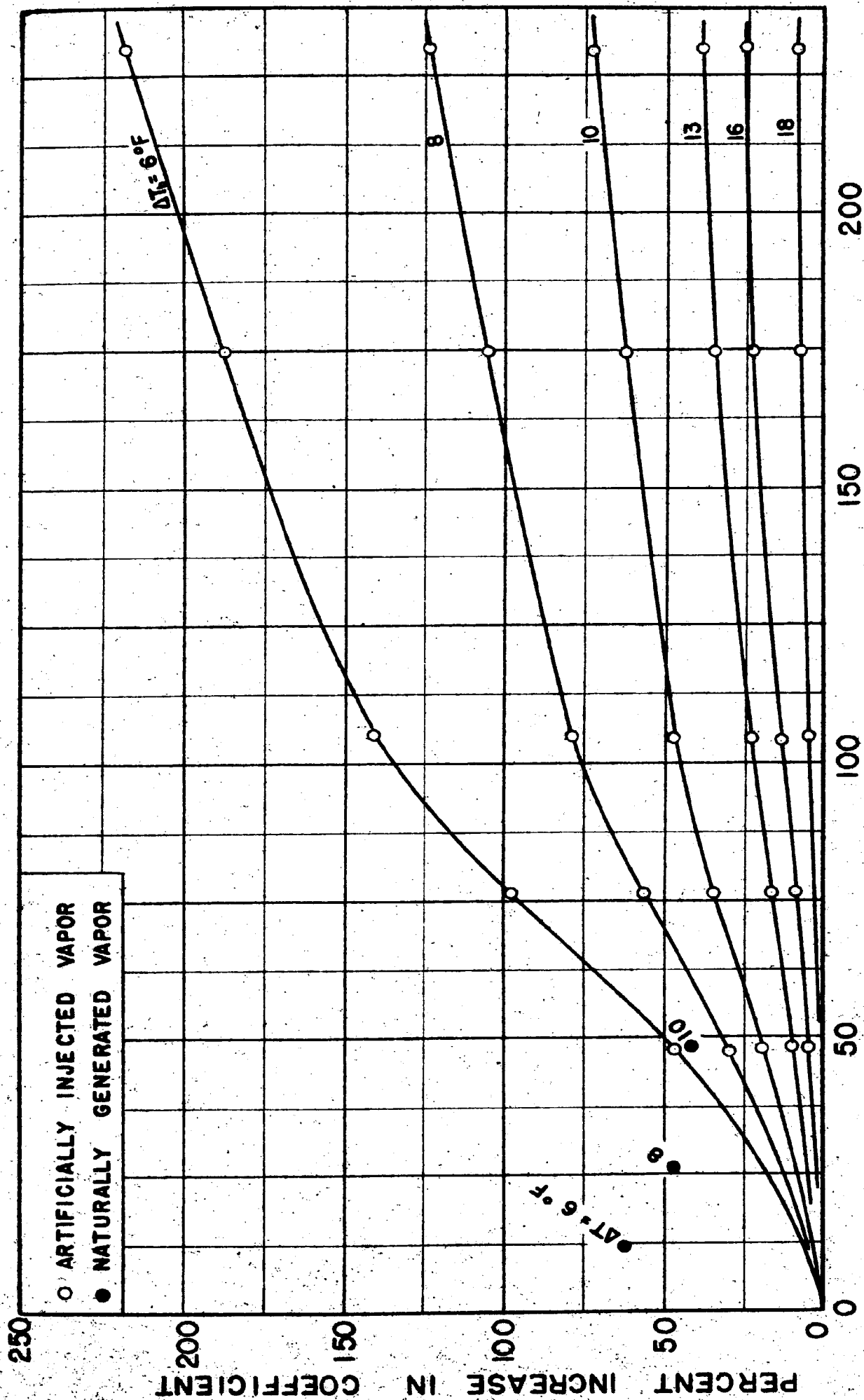


FIG. 18 INCREASE IN COEFFICIENT RESULTING FROM VAPOR INJECTION

Runs No. 100-102 were taken on June 17, the first day the tubes had been in the evaporator after being cleaned with cleaning solution. Runs No. 103-106 were taken on June 19 and were observed to give coefficients which fell below those for the first runs. It was thought that these first points might have been in error because of imperfections in operating the equipment for the first time. From June 21 to June 25, runs No. 107-126 were made at boiling temperature differences from 7.1 to 15 °F. The points taken at the lower temperature differences were reasonably consistent, although the general trend was for them to get lower from day to day. The points taken at higher temperature differences were in a region where the coefficients were changing very rapidly with ΔT_b , and it was difficult to tell whether or not the coefficients were agreeing with each other from day to day.

The Beckmann setting was changed on June 27, and data were taken at low temperature differences from then until June 30. During this period the majority of tests were made with gas introduction. Runs No. 128 and 132, which were made without any gas, fell below the previous data. The Beckmann setting was again changed and runs No. 140, 144-149, and 150-151 were taken between July 2 and July 8. All these points fell significantly below the data for the same boiling temperature difference which had been taken from one to two weeks previously. The effect was particularly noticeable at the higher temperature differences. Runs No. 154, 158, 162, and 166 were taken on July 13, and No. 176-179 on July 19.

Fig. 19 is a plot of the coefficients for the bottom tube for each of the above mentioned runs. The consistent downward trend is obvious. The effect is much greater at higher temperature differences than at lower temperature differences, and does not seem to have reached its minimum after the 32 days standing in Freon. The tubes appeared as clean to the naked eye on the last day as they did on the first. These results are in general agreement with the work of Pridgeon and Badger (15).

Only the data for runs No. 103-127 have been included in Fig. 12 because these runs were all taken in the space of one week and consequently were the only runs that could be reasonably compared on a tube to tube basis.

3. BOILING COEFFICIENTS FOR FREON 12

Boiling coefficients have been determined for Freon 12 for a copper surface that had not reached stability at the end of the runs. It is not expected that the surface would change much more before reaching a stable condition. The significant feature that has been brought out by the data is that the coefficients are less than about 300 BTU/hr. sq. ft. $^{\circ}\text{F}$ at temperature differences less than 7 $^{\circ}\text{F}$. This would be in the range of commercial operation. With water velocities of six ft./sec. inside the tubes, inside coefficients would be about 1100 BTU/hr. sq. ft. $^{\circ}\text{F}$. This would suggest advantages for a tube having a moderate amount of extended surface on the outside.

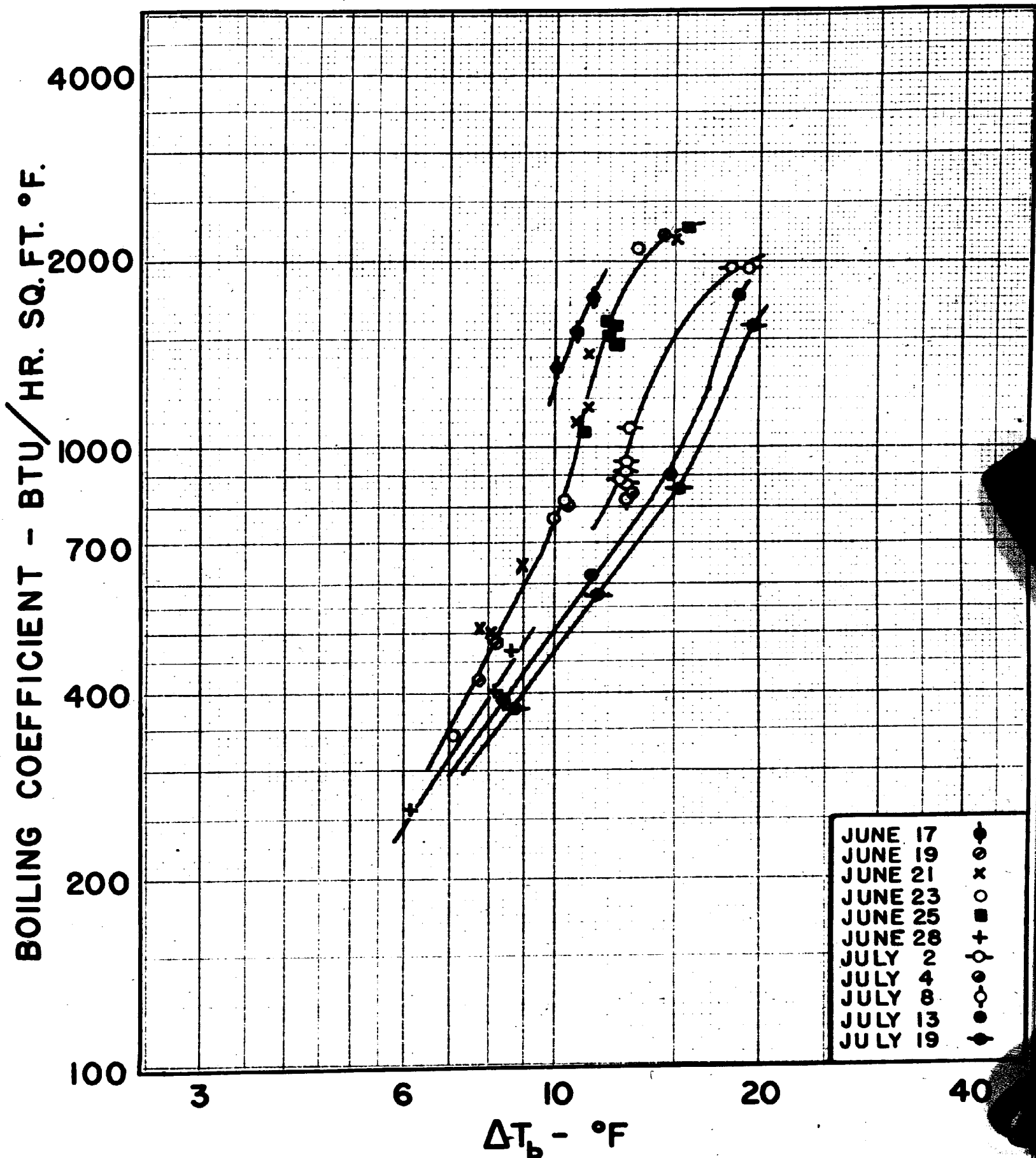


FIG. 19 EFFECT OF SURFACE CONDITIONS ON BOILING COEFFICIENTS

C. INSIDE COEFFICIENTS

1. CORRELATION OF EXPERIMENTAL DATA

Inside coefficients for all runs were calculated and are shown in Table II of Appendix C. They are based on the outside area of the tubes.

Fig. 20 is a plot of the experimental Nusselt number divided by the Prandtl number to the 0.3 power versus the Reynolds number. The data for this figure appears in Table V of Appendix C. The inside coefficients in this table are based on the inside area and only those runs which were made at different velocities and water temperatures or on different days are included. When a series of runs such as No. 138 and 139, or 154-157 were made at constant water conditions on the same day, the values were averaged and appear as run No. 138 or No. 154 respectively in Table V and on the plot. In all other runs the coefficients for the four tubes were averaged.

The slope of the curve through the data in Fig. 20 is 0.88. The equation for calculating the water coefficient then becomes

$$\frac{h_i D}{k} = 0.0134 \left(\frac{DV\rho}{\mu} \right)^{0.88} \left(\frac{C\mu}{k} \right)^{0.3} \text{ ----- (9)}$$

A line representing the equation

$$\frac{h_i D}{k} = 0.023 \left(\frac{DV\rho}{\mu} \right)^{0.8} \left(\frac{C\mu}{k} \right)^{0.3} \text{ ----- (10)}$$

is drawn for comparison.

The unusually high power on the Reynolds number might

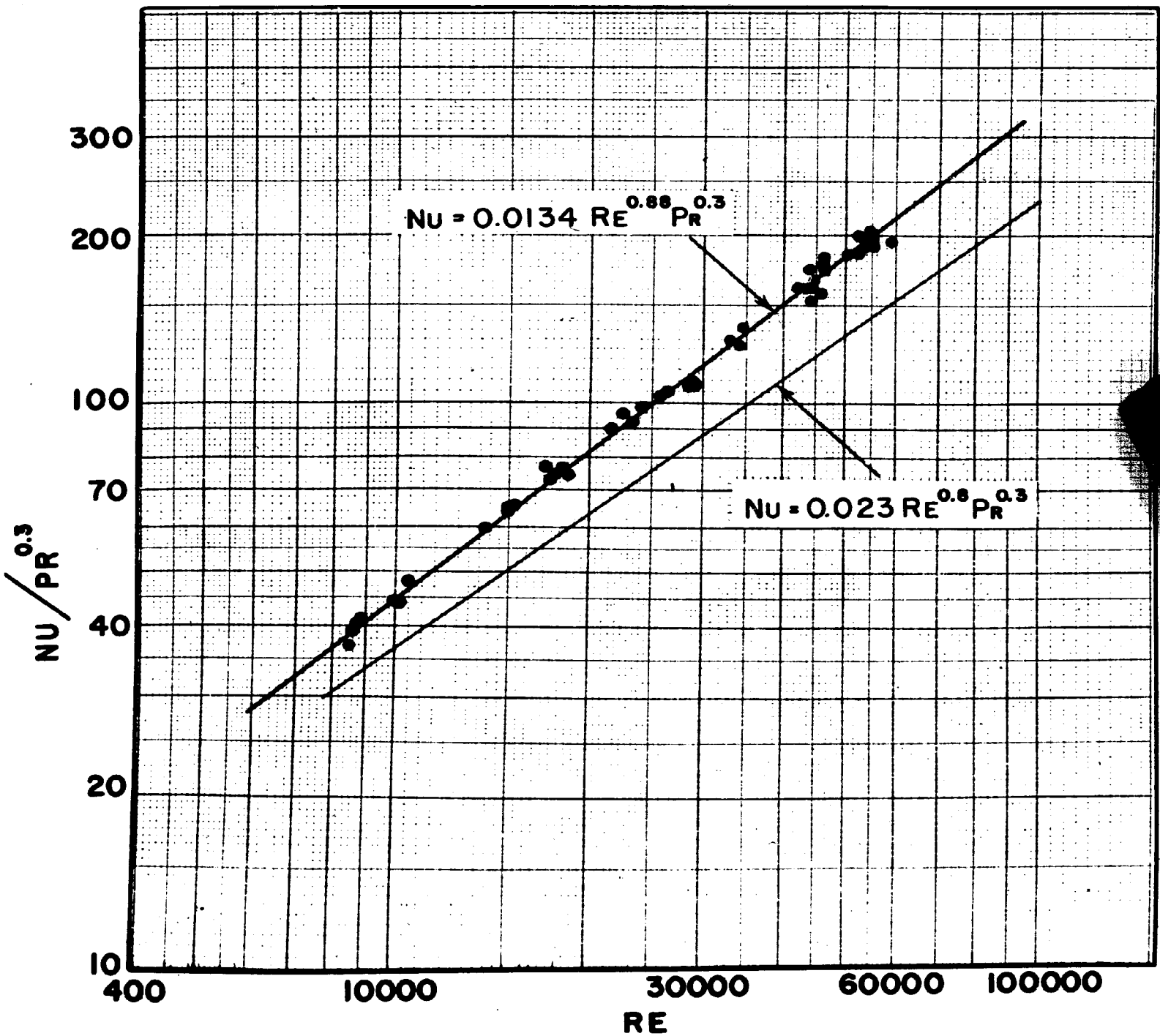


FIG. 20 CORRELATION OF INSIDE COEFFICIENTS

be explained by the fact that the orifices were relatively close to the inlet of the tubes. Tubes 1, 2, 3, and 4 had orifices which were 27, 24, 21, and 18 diameters upstream respectively. The added turbulence resulting from the orifices would result in higher coefficients. These data confirm the indication of the preliminary tests with carbon tetrachloride that the power should be greater than 0.8.

2. WILSON PLOT ANALYSIS

When the power on the Reynolds number has been established and the tubes are perfectly clean, it becomes possible to determine the boiling coefficient by use of a Wilson plot. The data for runs 123-126 and 144-147 were taken to make Wilson plots. The calculations appear in Table VI of Appendix C, and the points are plotted in Fig. 21 and 22. The information obtained from the plots is tabulated below in Table I, and the coefficients are compared to those determined by direct calculation.

TABLE I. WILSON PLOT ANALYSIS

Run No.	Tube No.	Intercept	Film Resistance	h Wilson Plot	h Direct Calc. (Avg)	% Difference
123-	3	0.000675	0.000648	1542	1541	--
126	4	0.000650	0.000623	1603	1551	3.3
144-	1	0.001110	0.001083	923	908	1.6
147	2	0.000886	0.000859	1163	1084	7.3
	3,4	0.000845	0.000818	1220	1161	5.1

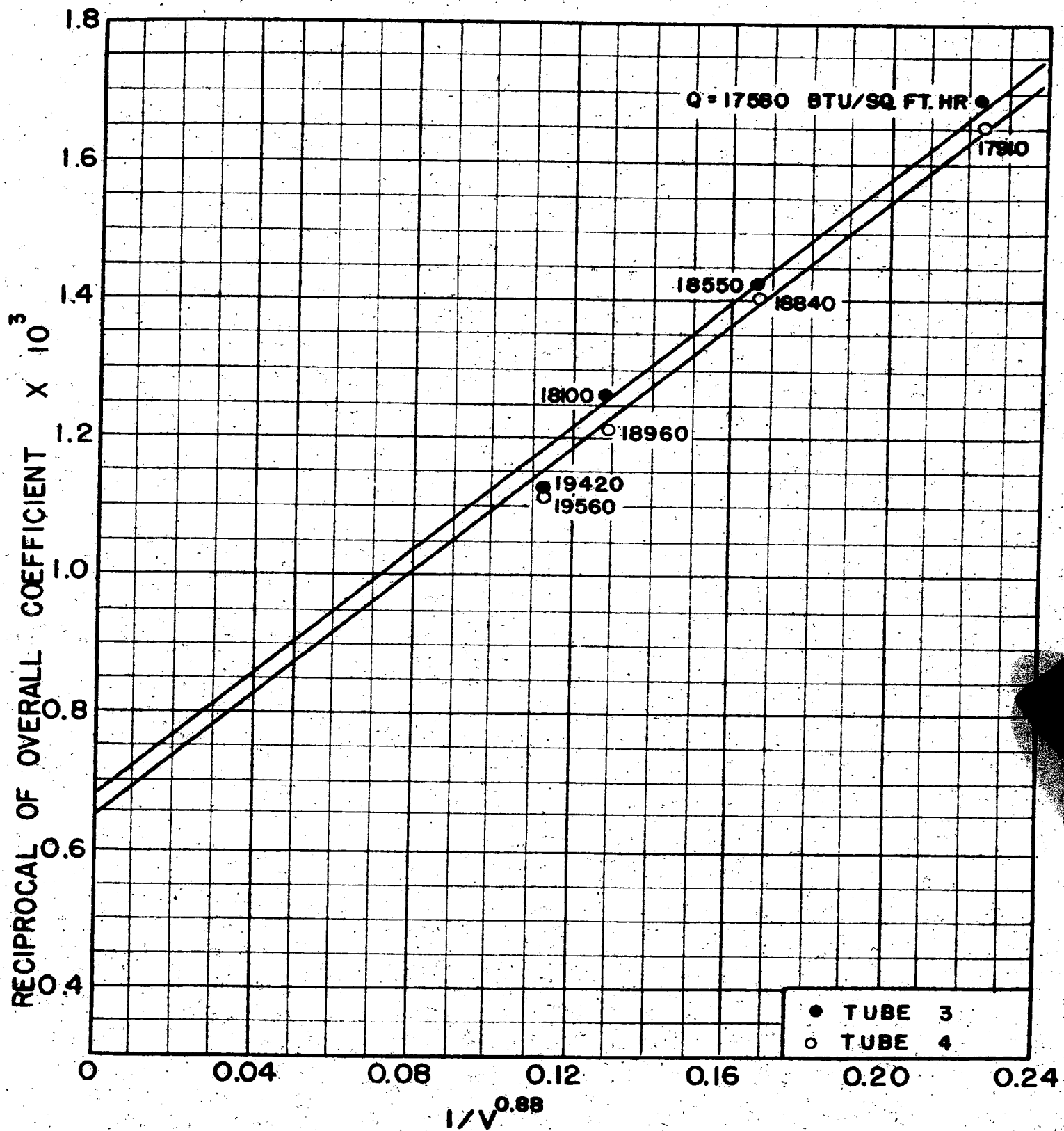


FIG. 21 WILSON PLOT - RUNS 123 - 126

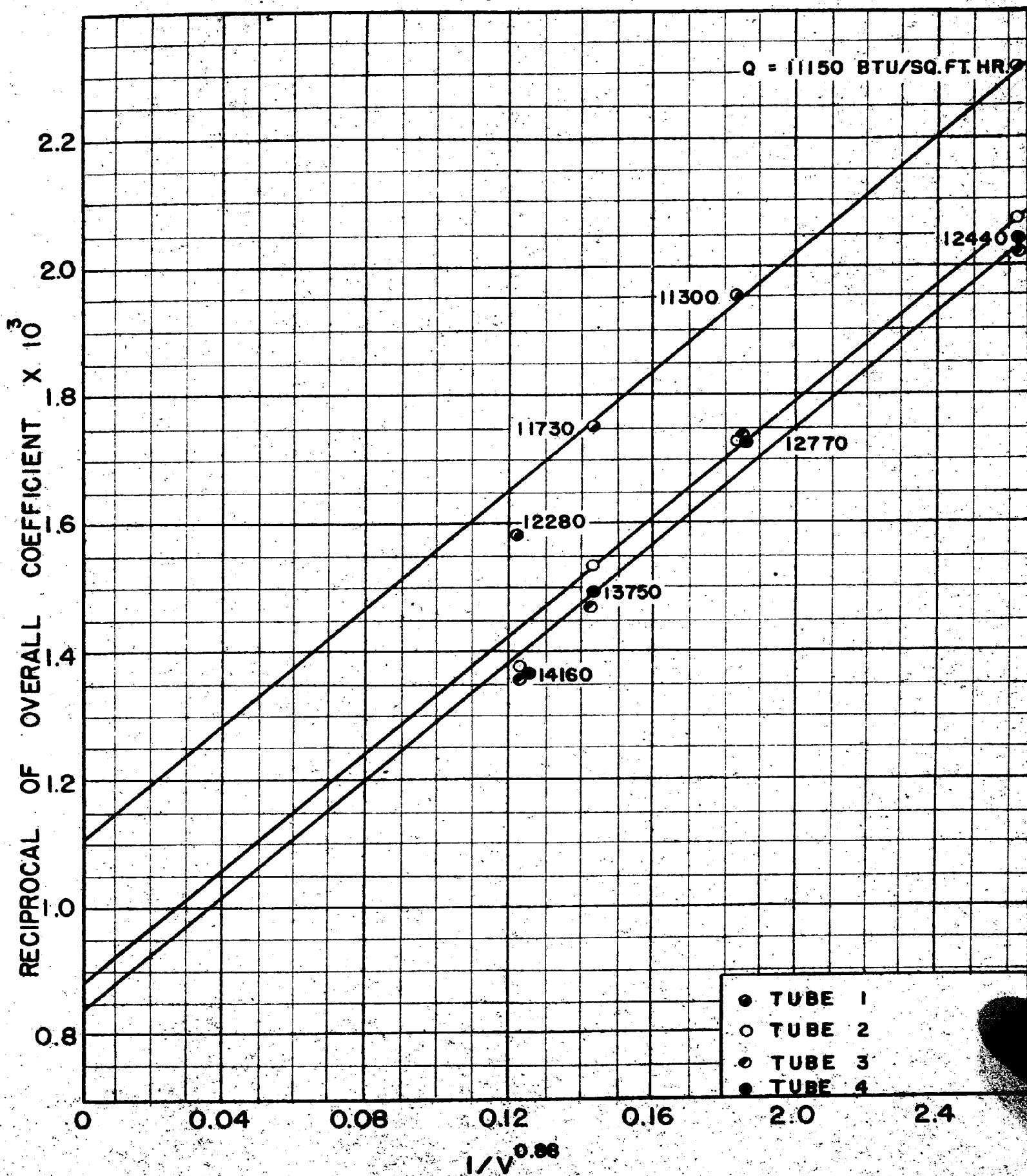


FIG. 22 WILSON PLOT - RUNS 144 - 147

The tube wall resistance for the evaporator tubes was calculated to be 0.000027 hr. sq. ft. $^{\circ}\text{F}/\text{BTU}$ and was subtracted from each of the intercepts.

As was explained earlier, considerable difficulty was experienced in keeping the heat load constant. Run No. 123 was practically ignored when locating the lines in Fig. 21 because the heat load was 8% higher than the average of the other three runs. The deviation of the points from a straight line is even more apparent in Fig. 22 where the heat load was more erratic. This emphasizes the necessity of keeping the heat load constant if the Wilson plot is to be used successfully. A slight trend of the heat load uniformly in one direction might give points lying in a straight line, but the value of the intercept might be considerably in error because it is obtained by extrapolation of a line having the wrong slope.

No correction has been made to the abscissa values for change in properties of the water because the average water temperature never changed by more than 9 $^{\circ}\text{F}$ for a series of runs. The change in properties over this range of temperature did not justify any correction.

D. ACCURACY OF MEASUREMENTS

At low flow rates, giving a manometer deflection of 100 mm or less, the error in water flow was probably about $\pm 1\%$. At higher flow rates, with larger manometer readings, the error resulting from inability to read the manometer scale would be less, but unsteadiness in the position of the meniscus prevented any more significant accuracy.

The Beckmann thermometers could be estimated to 0.001°C and it is felt that the readings were accurate to within $\pm 0.010^{\circ}\text{F}$ after making corrections for differences between thermometers. The minimum temperature difference reported (exclusive of run No. 180) is 0.394°F , which would therefore have a possible error of 2.5% . The average error would probably be less than 1% .

The thermocouples could be estimated to 0.1°F . Since the boiling temperature difference was determined by difference in thermocouple readings the maximum error would be $\pm 0.2^{\circ}\text{F}$. The minimum boiling temperature difference used was 4.98°F , which could therefore be in error by 4.0% . The maximum temperature difference is 19.9°F , with an error of 1.0% .

A check on the accuracy of the thermocouple readings was possible each time several different boiling conditions were used for a given set of inside conditions. If the inside coefficient calculated from the tube wall temperatures remained the same for the various outside conditions it was felt the temperatures of the tube wall were accurate. For example the boiling coefficient for runs 166-169, and 174-175 changed by 194% , but the calculated water coeffi-

cients only changed by $\pm 2.3\%$ from the average value obtained for the same inside conditions.

The temperature measurements have been carried to four figures in all cases because three sets of data were averaged to get each temperature. It is recognized, however, that the last figure is not significant, and that the combined maximum error in the experimentally determined coefficient resulting from a 2.5% error in water temperature, a 4.0% error in temperature difference, and a 1% error in flow measurement could be 8.7% for the low boiling temperature differences and water velocities. For most runs, at higher boiling temperature differences and water velocities, the maximum error would not exceed 3 or 4%.

X. CONCLUSIONS

A. FACTORS INFLUENCING BOILING COEFFICIENTS

The results of this investigation show clearly some of the fundamental variables that must be considered when attempting to correlate boiling coefficients. The profound change in coefficients with time has confirmed the fact that surface is a significant variable which up to this time has been denied quantitative description. The fact that the coefficients changed markedly at the same boiling temperature difference when the boiling liquid was subjected to external agitation effects indicates that boiling temperature difference is by no means a unique variable which will define the boiling coefficient for a given liquid on a given surface. At the same time, the fact that unlimited agitation does not appear as if it would make the coefficient independent of temperature difference for a given liquid and surface suggests that temperature difference is one of the fundamental variables. Liquid properties are also significantly important variables (5). From this it is concluded that the boiling coefficient is a function of surface condition, liquid properties, state of agitation of the liquid, and boiling temperature difference. This is analagous to saying that the inside coefficient is a function of liquid properties, liquid agitation, and geometry of the system. In the latter case the state of agitation may be defined by the Reynolds number, and the liquid properties and geometry by the Prandtl and Nusselt numbers. No such generalized groups have as yet been developed for

correlating boiling coefficients.

Correlating boiling coefficients on the basis of boiling temperature differences alone is analagous to correlating inside coefficients for a fixed system on the basis of inside temperature differences for data taken when the Reynolds number and heat load are simultaneously increased in the same proportions. Boiling temperature difference for a given system is largely a measure of the state of agitation which necessarily increases more or less uniformly as heat flux increases in the nuclear boiling region.

One might theorize somewhat on the mechanism of boiling. Vapor generation may be said to take place at an active center on the surface of the tube. The activity of this center may be due to temperature difference or to temperature difference and agitation of the surrounding liquid. Variations in the measured boiling coefficients result from various centers becoming active or inactive as the conditions change. The greater the number of active centers per unit of surface area the higher will be the coefficient. From this point of view, both boiling temperature difference and agitation are significant variables. At low temperature differences, it seems unlikely that any degree of agitation would cause the coefficient to reach the same high value as for the higher temperature differences.

The effect of time is to reduce the activity of all centers and hence reduce the maximum coefficient. This is in accordance with the observed downward displacement of the coefficients in fig. 19. It is believed that the

effect of agitation would be relatively the same with respect to any of these displaced curves. In other words a family of curves similar to the one in Fig. 16 could be obtained for any condition of the surface. The observed effects of agitation may therefore be quantitative when placed on a percentage basis as in Fig. 18.

B. CONCLUSIONS SUMMARIZED

1. A method has been developed for satisfactorily measuring the temperature of thin tube walls when heat is being transferred through the walls. The method is suitable when the tubes are enclosed in equipment under pressure and it minimizes many of the errors inherent in other methods of installation.
2. The effect of agitation of a boiling liquid on the boiling film coefficient has been studied quantitatively for the case of vapors naturally rising from one tube to another in a vertical row of four tubes, and for the case of gas artificially injected beneath the bottom tube in the bundle.

For the naturally rising vapor, the effect was to increase the coefficient of the second tube over that of the bottom tube, and to increase the coefficient of the third and fourth tubes over the second to a lesser degree. No significant difference was observed between the third and fourth tubes except at very low boiling temperature differences. The effect of one tube on another was negligible for high temperature differences.

For artificially introduced vapor, the effect was to increase the coefficients markedly at low temperature differences and not at all at high temperature differences. The coefficients became increasingly higher for greater volumes of gas injected but indications are that a maximum is reached for any given temperature difference.

3. The effect of a changing copper surface on the boiling coefficient has been observed over a period of 32 days of intermittent operation. At a temperature difference of 11.5°F , coefficients changed from 1750 BTU/hr. sq. ft. $^{\circ}\text{F}$ for a freshly acid cleaned surface to 580 BTU/hr. sq. ft. $^{\circ}\text{F}$ at the end of the period. The effect was less at lower temperature differences. The condition of the surface was controlled by time only.
4. Boiling coefficients for Freon 12 have been determined for a copper surface that had been in contact with Freon for 32 days. The coefficients range from 260 at a temperature difference of 7°F , to 1600 at a temperature difference of 19.5°F .
5. The low outside coefficients at boiling temperature differences less than 7°F , when compared to the higher inside coefficients obtained at moderate water velocities, would suggest that significant advantages would be gained in commercial practice by using tubes having extended outside surfaces.

6. Heat transfer coefficients for water flowing in copper tubes 0.620 in. inside diameter, 38 in. long, with orifices from 27 to 18 diameters upstream have been found to follow the equation

$$\frac{h \cdot D}{k} = 0.0134 \left(\frac{DV\rho}{\mu} \right)^{0.88} \left(\frac{c\mu}{k} \right)^{0.3}$$

7. The Wilson plot method can be used satisfactorily to evaluate boiling coefficients provided the heat load, surface conditions, and state of agitation of the liquid can be held constant for the required range of water velocities, and provided the power to which the Reynolds number occurs in evaluating the inside coefficient is known accurately.
8. It has been concluded that a satisfactory correlation for boiling coefficients will include the temperature drop across the boiling film, and a number or group of numbers that describe the physical properties of the fluid, the state of agitation of the fluid, and the surface from which boiling is taking place.

XI. APPENDIX A

THE INSTALLATION OF THERMOCOUPLES IN THIN-WALLED COPPER TUBES.

A. INTRODUCTION

Numerous attempts have been made using a thermocouple to measure the temperature of a tube wall through which heat is flowing. The conditions which must be satisfied by any particular installation have been recognized by practically all authors (4, 10, 14), but it cannot be said that these conditions have all been satisfied by the proposed installations. The fundamental difficulty that arises in any thermocouple installation in a tube wall is in getting the temperature at the point where the thermocouples first make contact the same as that at any other geometrically similar point in the tube wall where there has been no interruption of the metal or surface resulting from the installation.

Interruption of the metal and other mechanical details of the method can influence this temperature in a number of ways. One of these is by interrupting a high conductivity metal such as copper with a core or groove filled with a low conductivity metal such as solder. Such a discontinuity would have a profound effect on the pattern of the flow of heat at the interrupted point and consequently on the temperature. This condition is certainly present in the installation proposed by Hebbard and Badger (10) where the heat flows through a plug of lead between the junction and the tube wall surface.

Another way in which the temperature may be influenced is to have the tip of the thermocouple junction suitably located to give a relatively correct temperature, but at the same time to have the wires in contact a quarter of an inch or so back from that point. The temperature indicated is of course the temperature at the first point of contact. An illustration of this may be found in the installation of Baker and Mueller (4) and Mohun and Peterson (14).

Two other ways in which the temperature may be affected are in having conduction of heat along the leads to or from the junction, and in having the junction at some unknown position relative to the surface of a comparatively thick-walled tube across which the temperature difference cannot be neglected. The effect of conduction has been well accounted for in the above installations. It is unimportant when small wires are used and when temperature differences along the wire are small. Mohun and Peterson (14) give an excellent analysis of the problem regarding location relative to the surface.

Akin (1) has reported an installation using a single constantan wire in a copper tube which has many desirable features. It was impractical in the present case because the leads could not have been taken through the equipment under pressure.

B. METHOD OF INSTALLATION

Briefly, the method was to measure the temperature of the tube wall at three different places, using a copper and a constantan thermal junction in indirect contact.

The leads were to be brought through the end of the evaporator enclosed in individual stainless steel tubing. Each couple was to have a separate cold junction. The em.f. was to be measured with a Leeds and Northrup portable precision potentiometer.

The installation may be resolved into five more or less distinct operations, namely, the preparation of the tubes, the assembly of the individual thermal elements, the assembly of the elements and the tubes, the packing of the lead wires, and the switching arrangement.

1. PREPARATION OF THE TUBES

Fig. 23 shows the detail of the thermocouple assembly. Two holes 0.035 in. in diameter were drilled opposite each other at an angle of 120° , the open ends of the holes facing in the opposite direction and the bottom of one hole near the bottom of the other. These holes were drilled at three places; the middle and 5 in. from each end of a 38 in. evaporator tube. A small square steel block, drilled and reamed just large enough to slip over the outside of the tube, was used for a jig while drilling the small holes.

Fig. 24 illustrates this jig. The method of locating the correct position for the small guide hole was largely trial and error, after unsuccessful attempts at location by measurement. A hole was drilled at supposedly the correct place in the jig. The drill was then pushed through the hole in the empty jig, and a tube inserted until it butted up against the drill to see exactly where the drill would

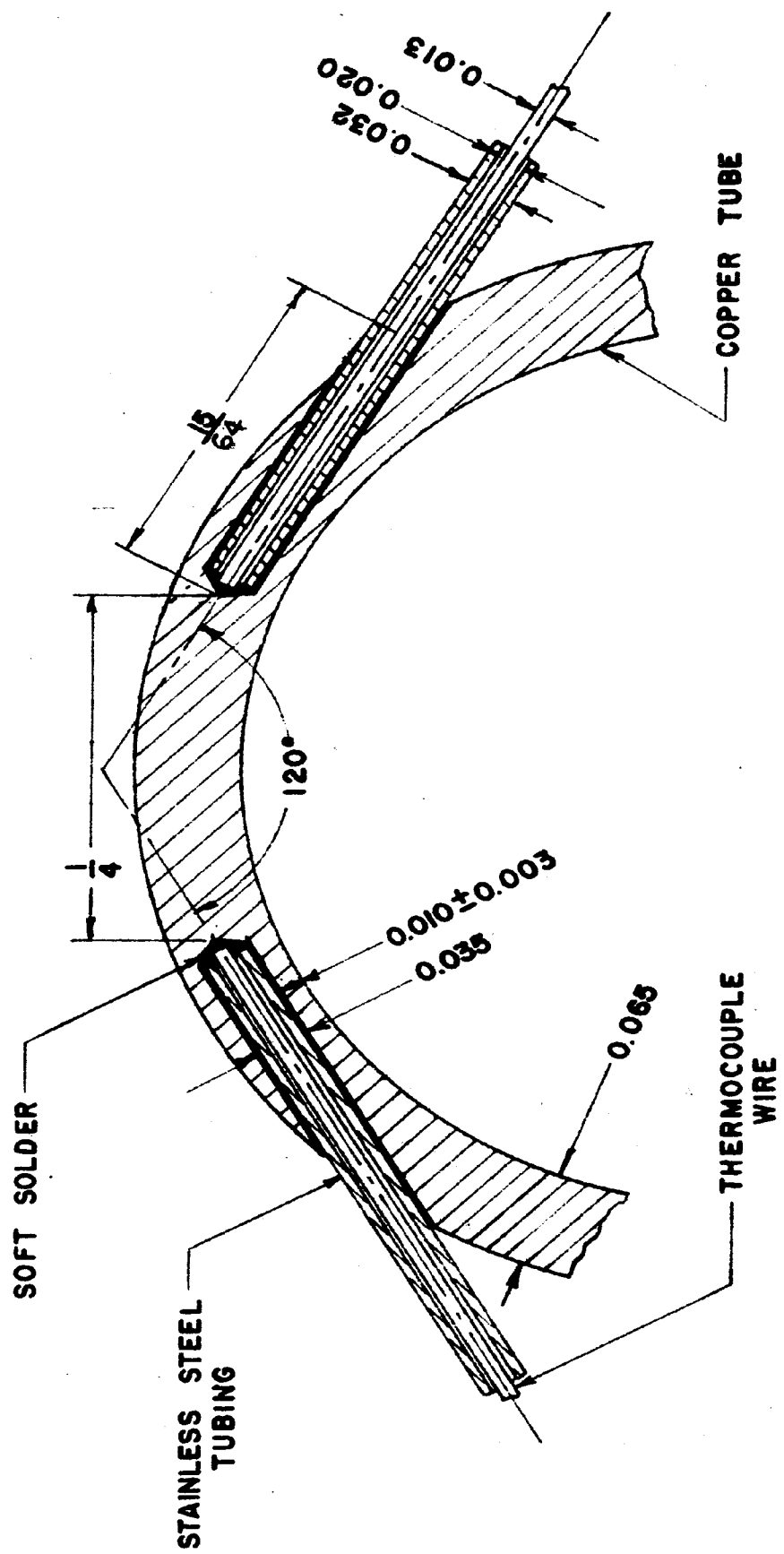


FIG. 23 THERMOCOUPLE ASSEMBLY

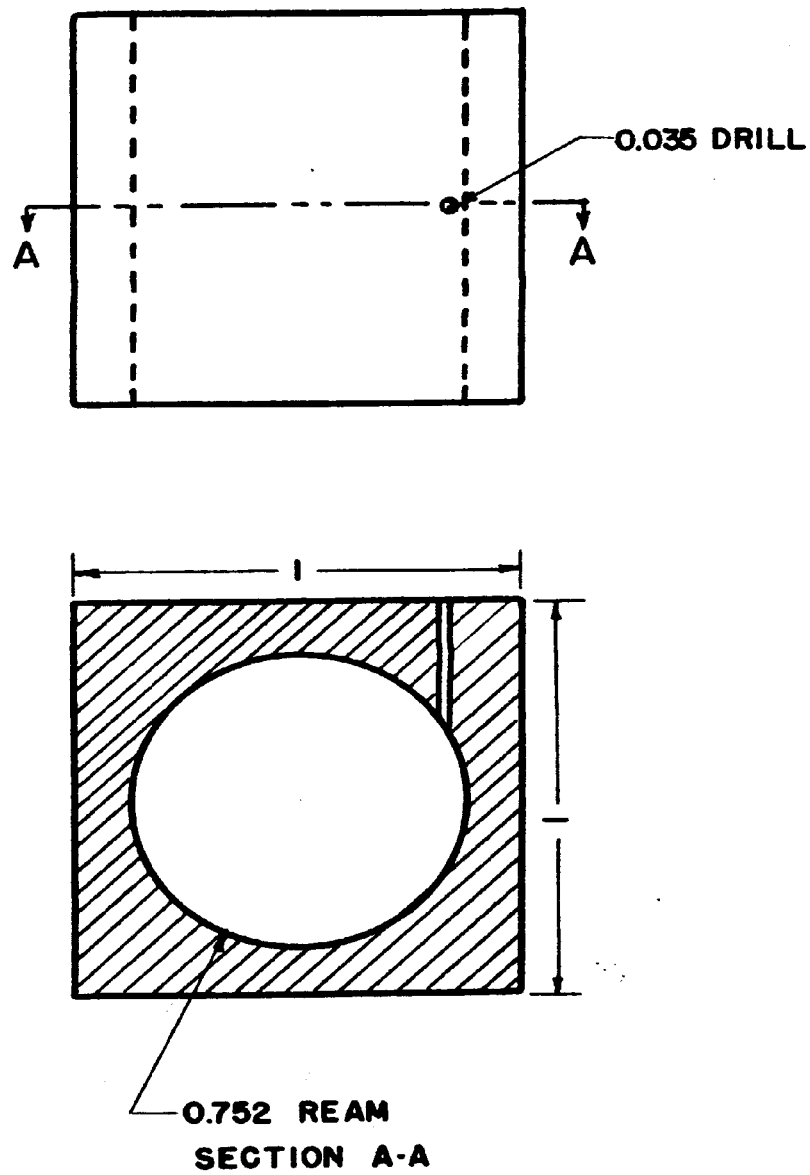


FIG. 24 JIG FOR DRILLING TUBE WALL

go in the tube wall. This procedure was repeated until the desired position was found. Lard oil was found successful as a drilling fluid. The tubes were annealed copper which had a strong tendency to tear and clog the drill with resultant breakage of the drill and ruin of the tube and the hole in the jig.

The holes in the bottom tube were located so that the measured temperature at one end would be on the bottom, in the middle it would be on the side, and on the other end it would be on the top.

2. ASSEMBLY OF THERMAL ELEMENTS

Each thermal element consisted of a single wire inside a stainless steel hypodermic needle tube. The stainless steel tubes were 0.032 in O.D. and 0.020 in I.D. The length of the tubes necessary ranged from 17-1/2 in. to 50 in. A single No. 30 thermocouple wire was pushed through each of these tubes. The wires were covered with baked enamel and silk and had an overall O.D. of about 0.013 in. It was impossible to push untreated wires through the tubes without the insulation unravelling at the end. Accordingly, twelve wires each of constantan and copper were cut to length and the ends given a thin coat of red Glyptal No. 1201 enamel to hold the silk tightly in place during the pushing operation. It was absolutely necessary that no bumps should appear on the surface of this enamel and that no kinks of any kind occur in the wire in order that it could be pushed through the tube. Each wire was pushed through until about 2 in. protruded at one end. They were

long enough so that about 3 ft. were left at the other end to make connections to the cold junction or switch.

After being pushed through, the long ends of the wires were covered with two coats of Glyptal No. 1201 enamel to prevent short circuits developing during the remainder of the operations, and to give the wires a certain amount of rigidity. All assemblies were tested for short circuits between the wire and the tube but none was found.

3. ASSEMBLY OF THE ELEMENTS AND THE TUBES

The drilled holes in the tube wall were cleaned out with carbon tetrachloride to remove any of the cutting fluid before the thermal elements were soldered in. Each hole was then filled with very small chips of tin solder mixed with some resin flux. The tube was then mounted in clamps with the holes in the bottom so that the solder would not run out when it was heated.

The short ends of the thermocouple wires were cut off with a pair of scissors about 1/32 in. from the end of the stainless tube. None of the insulation was interrupted at any other place so that the only place the wire was bare and able to make electric contact was over a cross section at its very tip. The end of the wire was then tinned with a needle point soldering iron. The end of the stainless tube was also tinned up to about half an inch from the end. The flux for this was a mixture of hydrochloric and acetic acid.

The two thermal elements, one copper and one constantan, which were the same correct length for the position

in question in the copper tube were then taken, one in each hand, and held ready to insert in the holes. The tube was then heated with a small flame until the solder melted in the holes. While it was molten, the ends of the elements were plunged into the holes and the flame removed immediately. The circuit was then checked for electrical contact. In one such case, where no contact was made, the element had to be removed. This gave an opportunity to examine its condition after heating, in fact after excess heating. No apparent change had taken place in the enamel insulation, although the silk had charred. The tip of the wire was the only place where solder was sticking.

The long stainless tubes were then bent up and over so they would be in the right position to go through the composite packing gland in the end of the evaporator. The copper tubes were then thoroughly cleaned with a very dilute solution of potassium dichromate in sulphuric acid. Any excess solder near the junction of the stainless steel with the copper tube was carefully scraped away. The tubes were then rolled into the tubesheet with the stainless steel tubing containing the thermocouple lead wires protruding through two $3/4$ in. holes drilled near the top of the tubesheet. The tubes containing the copper wires were all brought out through one hole and those containing the constantan wires at the other.

4. PACKING OF THE LEAD WIRES

Fig. 25 illustrates in detail the packing gland assembly in the cover plate on the unpacked end. A $3/4$

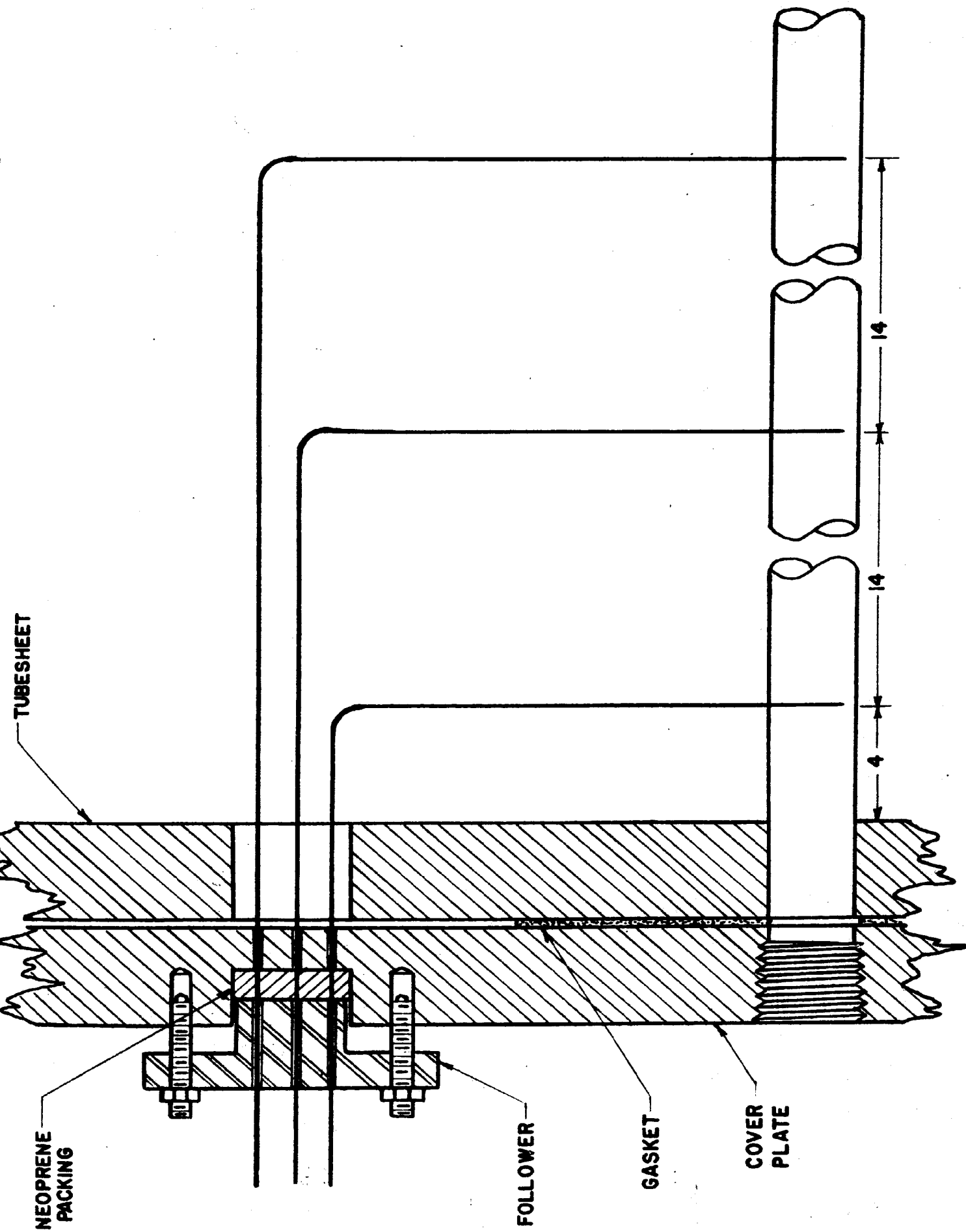


FIG. 25 PACKING GLAND ASSEMBLY

in. hole was drilled $1/2$ in. deep at two places in the front of the cover plate, and the bottom cleaned out with a flat-bottomed drill. Thirteen holes $1/16$ in. in diameter were then drilled all the rest of the way through the plate to take the individual tubes. The tubes containing the copper wires were all brought out at one gland and the tubes containing the constantan wires at the other.

A packing follower for each hole was made from a block of brass so that it would just fit into the hole in the cover plate. Thirteen holes which exactly matched the small holes in the cover plate were drilled all the way through as shown in Fig. 26. The flange was designed to take four $1/4$ in. stud bolts. Companion holes were drilled and tapped in the cover plate.

A neoprene rubber plug for packing in each $3/4$ in. hole was then cut from a rubber stopper and it likewise was drilled with 13 holes in an identical pattern.

The assembly of this packing gland was extremely tedious. The tube bundle was inserted in the evaporator with all the stainless tubes protruding about 4 inches through the $3/4$ in. holes in the tubesheet. The cover plate was then suspended in front of the evaporator and the wires and tubes all pulled through the individual holes in the cover plate, which was then bolted in place. The follower and packing were then suspended in front of the cover plate as in the photograph of Fig. 27. The wires were threaded through the holes in the rubber plug and follower corresponding to those in the cover plate. The method of

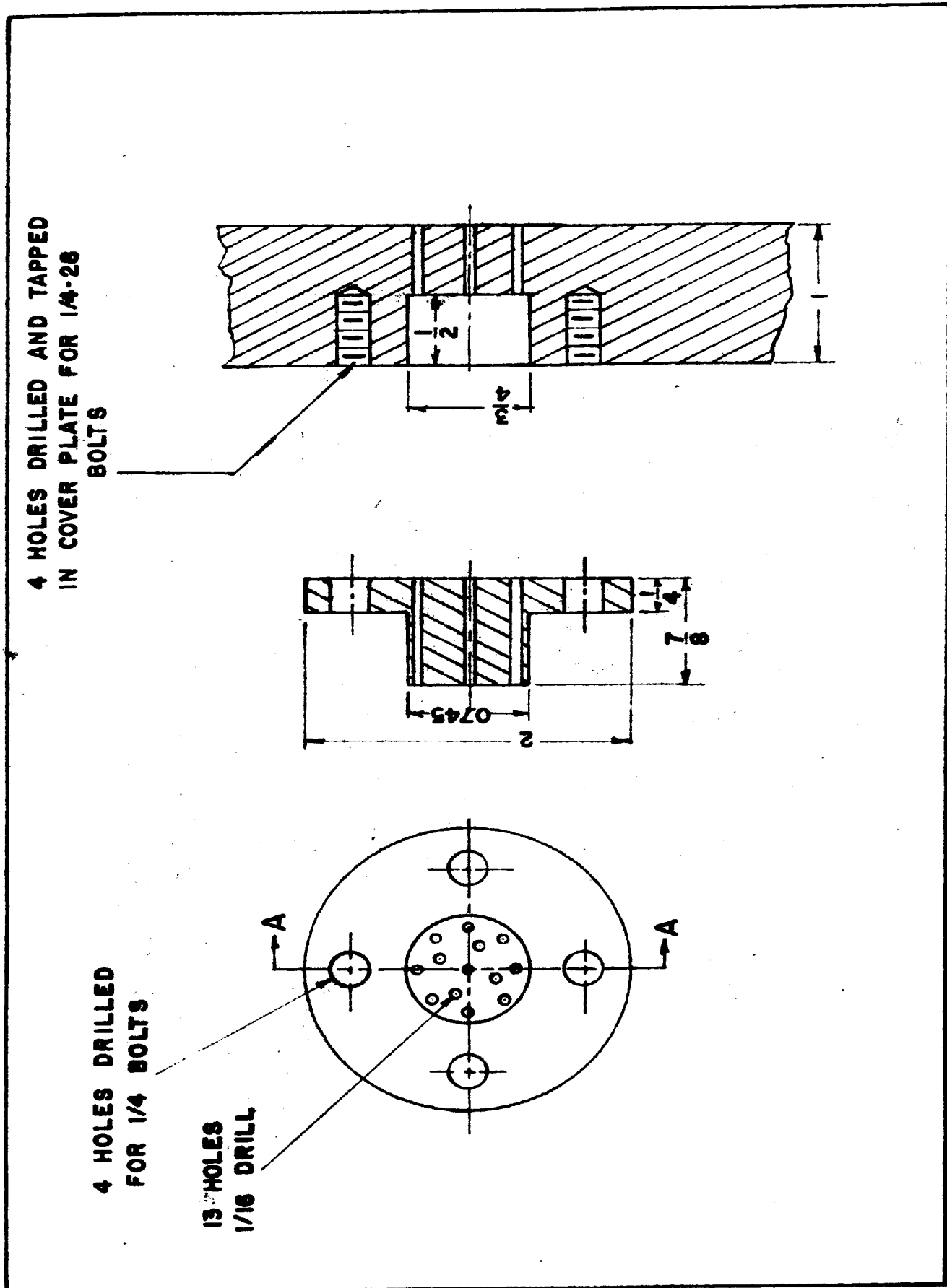


FIG. 26 DETAIL OF PACKING GLAND AND FOLLOWER

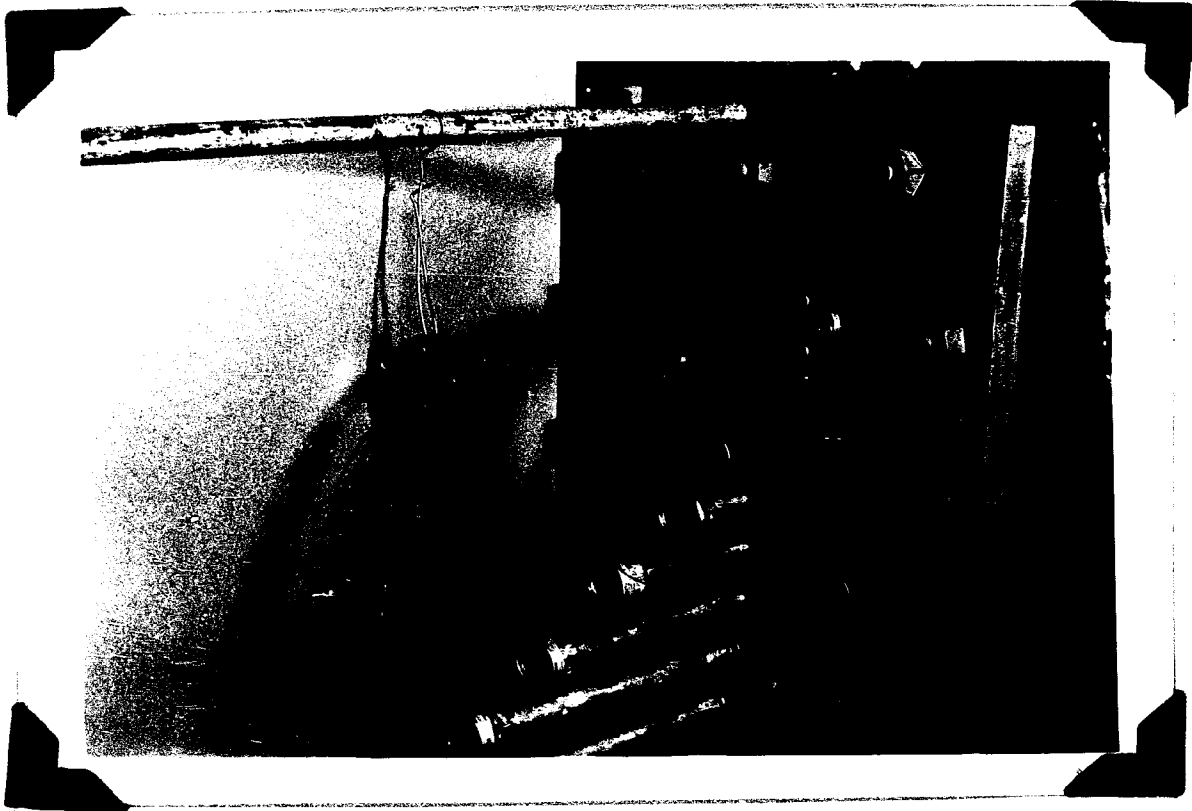


FIG. 27 PACKING GLAND INSTALLATION

inserting the wire through the rubber was to insert a short length of stainless tubing 0.042 O.D. by 0.027 I.D. into the hole. The wire was pushed through the tubing, the tubing then pulled the rest of the way through the rubber, and all the way off the end of the wire. The packing and follower were pushed on over the ends of the small stainless tubes and fastened in place.

The assembly was tightened up once and never gave any trouble with leaks.

5. SWITCHING ARRANGEMENT AND COLD JUNCTION

Each thermocouple had a separate cold junction. The constantan lead wires were cut to the right length and fastened to a piece of copper wire by twisting and soldering. The two copper wires which resulted, one from the hot and one from the cold junction, were then soldered to the posts of two 2-gang switches. The cold junctions were thoroughly coated with Glyptal No. 1201 enamel. They were placed in a thermos bottle full of cracked ice while readings were taken. One switch was used for the bottom two tubes and lower liquid temperature and one for the rest. Switch A, on the left of the panel, and switch B, on the right of the panel had the following arrangement:

Switch A		<u>Position</u>	
<u>Tube 1</u>	Upstream	4	(bottom)
	Middle	3	(side)
	Downstream	2	(top)
<u>Tube 2</u>	Upstream	7	
	Middle	6	
	Downstream	5	
<u>Freon</u>		8	

Switch B.		<u>Position</u>
<u>Tube 3</u>	Upstream	6
	Middle	7
	Downstream	8
<u>Tube 4</u>	Upstream	3
	Middle	4
	Downstream	5
<u>Freon</u>		2

6. FREON LIQUID COUPLES

The Freon liquid couples were made by pushing one constantan and one copper No. 30 insulated wire through a stainless steel tube 0.042 in. O.D. and 0.027 I.D., and soldering completely over the end where the tips of the separate wires were bared. Otherwise the method of installation was the same.

One liquid thermocouple was located beside tube 2 and the other beside tube 4.

C. CALIBRATION

Mechanical difficulties precluded the possibility of calibrating the couples before the installation was complete. It was impossible to arrange the cold junction and switches before the tubes were inserted in the packing gland.

The method used to calibrate was to pass city water through the tubes at a known temperature for a period of about 24 hours, with nothing in the evaporator, and to take readings at the end of that time. The readings were assumed to be those corresponding to the water temperature. All couples read the same within the accuracy of measurement which was 0.002 mv. or $1/10^{\circ}\text{F}$ on a Leeds and Northrup port-

able precision potentiometer. The average deviation from the standard Leeds and Northrup 38 calibration was -0.2°F . The curves used for getting temperatures from potentiometer readings are shown in Fig. 34 of Appendix C.

Although the couples were only calibrated over a limited range of temperature, it is not felt that the e.m.f. per degree would change significantly over the range for which the couples were used, 32°F to 80°F .

XII. APPENDIX B

AUXILIARY OPERATION CONNECTED WITH THE FREON CIRCUIT.

A. CHARGING LIQUID TO THE EVAPORATOR

Liquid is charged to the storage tank from a commercial cylinder by connecting the cylinder to the tank at valve O, Fig. 1, elevating the bottom end of the cylinder, opening valves O, P, and A, and turning on the compressor so the tank is under reduced pressure. Liquid then flows into the tank from the cylinder.

Liquid is charged to the evaporator by adjusting valves N, M, P, and B. Valve P is closed tight and valves N, M, and B are opened slightly. With the compressor on, a small amount of liquid will flow into the evaporator. Valve B is then shut off and the compressor left on until the evaporator gets cold. Valve B is then reopened, admitting the liquid from the tank to the evaporator as a result of the pressure difference in the two vessels.

B. REMOVING FREON FROM THE SYSTEM

If oil should get into the Freon in the evaporator, or if for any other reason the Freon has to be removed, the following procedure is suitable.

A 45 gal. drum is placed on a scale near the compressor with a commercial Freon cylinder standing in it. The drum is partially filled with cold water at a definite level and the scale balanced. The cylinder is connected to the Freon condenser at the lead-off for gauge IV, Fig. 1. Valves C and D are closed tightly and the compressor started. Liquid

will be forced from the condenser into the cylinder. It may be necessary to adjust the automatically controlled water valve so that the condenser pressure is high enough to force the liquid into the cylinder. When about 125 lb. of Freon have been collected the cylinder should be changed. A steam hose used on the storage tank enables the Freon to be removed more rapidly.

The last bit of Freon is removed with the small 1/3 h.p. compressor using its air condenser and discharging liquid to a small commercial Freon cylinder.

Precaution must be taken in all these operations to purge air from all connecting lines.

The oil can be drained if necessary from valve O or at the sight glass in the storage tank when the Freon has been removed.

C. CHARGING OIL TO THE COMPRESSOR

The compressor may be isolated from the rest of the system by closing valves Q, G, F, and E, Fig. 1. With these valves closed the Freon gas can be removed from the main compressor using the small 1/3 h.p. compressor connected to the main compressor valves. When the pressure has reached atmospheric, the small compressor is stopped, the oil line to the top of the differential oil pressure switch disconnected, and put beneath the surface of the oil in a container. The small compressor is then restarted and oil drawn into the crankcase by suction to the desired level. The compressor is then stopped and the oil line left in the container until atmospheric pressure is again

reached. It may then be reconnected to the differential switch.

D. ADJUSTMENT OF COMPRESSOR OIL PRESSURE

This procedure is outlined fully in the Carrier instruction sheets.

XIII. APPENDIX C

EXPERIMENTAL AND CALIBRATION DATA

TABLE II
SUMMARY OF EXPERIMENTAL DATA

Date	Run No.	Tube No.	Water Temp. In of	(4)	Water Temp. Drop of	(5)	Water Rate lb/hr.	(6)	Heat Flux BTU/hr sq.ft.	(7)	Tube Wall Temp. of	(8)	Freon Temp. of	(9)	Freon Pressure p.s.i.a.	ΔT_{cu} of	(11)	ΔT_b of	(12)	h BTU/hr sq.ft. of	(13)	ΔT_i of	(14)	h_i BTU/hr sq.ft. of	(15)
June 17	100	1*	67.25	3.530	3350	19950	49.23	37.73	35.2	0.49	11.25	1775	16.15	1235											
		2	67.20	3.607	3370	20500	49.77	37.53		0.49	11.99	1711	15.38	1332											
		3	67.20	3.523	3380	20800	49.26	37.37		0.49	11.66	1723	15.32	1306											
		4	67.20	3.524	3380	20800	49.17	37.37		0.49	11.55	1737	16.03	1296											
17	101	1	67.80	3.973	2470	16520	48.45	37.35	34.9	0.40	10.30	1503	17.15	964											
		2	67.80	4.081	2460	16930	48.77	37.30		0.40	11.27	1507	16.70	1013											
		3	67.80	4.019	2470	16740	48.37	37.37		0.40	10.80	1552	17.22	972											
		4	67.80	4.007	2470	16700	48.27	37.25		0.40	10.32	1545	17.33	964											
17	102	1	67.35	1.708	4700	13570	58.35	48.30	45.0	0.33	10.39	1308	7.67	1769											
		2	67.40	1.732	4675	14090	59.00	48.58		0.35	10.24	1373	7.34	1918											
		3	67.45	1.825	4675	14430	58.33	48.37		0.36	9.78	1476	8.03	1797											
		4	67.45	1.827	4710	14500	58.47	48.25		0.36	10.04	1443	8.07	1797											
19	103	3	66.90	3.514	5430	32200	47.33	32.80	31.0	0.79	14.53	2220	17.41	1850											
		4	66.95	3.560	5370	32300	47.30	32.80		0.79	14.50	2230	17.47	1850											
19	104	1	64.05	2.129	911	3270	54.85	47.05	44.0	0.08	7.76	422	8.12	402											
		2	64.05	2.333	911	3590	53.70	47.05		0.09	6.65	542	9.15	393											
		3	64.05	2.354	911	3630	53.45	47.07		0.09	6.34	572	9.38	387											
		4	64.05	2.441	911	3755	52.50	47.05		0.09	6.41	588	9.28	404											
19	105	1	64.00	2.590	911	3980	53.03	44.55	41.7	0.10	8.23	485	9.62	413											
		2	64.00	2.821	911	4330	51.65	44.55		0.11	6.85	634	10.88	399											
		3	64.00	2.833	911	4350	51.55	44.55		0.11	6.75	646	11.04	394											
		4	64.00	2.889	911	4440	51.75	44.65		0.11	6.85	649	10.76	412											

* bottom tube

(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)	(12)	(13)	(14)	(15)
20	106	1	64.90	3.361	2280	12960	48.59	37.35	34.8	0.32	11.08	1168	14.47	897
		2	64.90	3.494	2280	13440	46.30	37.35		0.33	10.79	1243	14.68	916
		3	64.90	3.499	2290	13520	47.95	37.25		0.33	10.54	1282	15.03	900
		4	64.85	3.431	2275	13200	48.17	37.30		0.33	10.64	1237	14.80	893
20	107	1	64.00	2.817	876	4160	52.25	44.45	41.4	0.10	7.75	538	10.97	379
		2	64.05	3.053	876	4520	51.18	44.20		0.11	6.93	653	12.02	377
		3	64.00	3.087	876	4570	50.70	44.28		0.11	6.37	717	12.50	366
		4	64.00	3.132	876	4620	50.80	44.28		0.11	6.47	715	12.33	374
20	108	3	67.20	3.514	5540	32900	48.06	32.65	31.0	0.81	14.89	2210	16.91	1945
		4	67.20	3.523	5500	32800	48.00	32.65		0.81	14.83	2210	16.97	1934
21	109	1	63.35	2.171	1582	5800	53.30	44.75	42.0	0.14	8.98	647	8.90	652
		2	63.85	2.374	1582	6350	52.80	44.75		0.15	7.98	786	9.80	648
		3	63.35	2.432	1591	6520	52.47	44.75		0.16	7.64	855	10.08	648
		4	63.85	2.506	1579	6680	52.63	44.75		0.16	7.30	847	9.89	676
21	110	1	65.15	2.673	915	4110	54.00	46.10	43.3	0.10	7.85	525	9.77	421
		2	65.15	2.840	913	4370	52.95	46.10		0.11	6.80	642	10.73	406
		3	65.15	2.923	915	4500	52.52	46.10		0.11	6.37	707	11.12	405
		4	65.15	3.004	913	4740	52.50	46.10		0.11	6.35	732	11.09	428
21	111	1	65.75	3.425	2730	15800	48.45	37.15	34.9	0.39	11.10	1420	15.40	1025
		2	65.75	3.494	2740	16150	48.70	37.10		0.40	11.20	1440	15.11	1068
		3	65.75	3.497	2750	16220	48.32	37.00		0.40	11.12	1458	15.49	1048
		4	65.75	3.488	2730	16080	48.50	37.00		0.40	11.20	1433	15.31	1050
21	112	1	65.80	3.767	1842	11730	47.95	37.20	35.0	0.29	10.60	1106	15.93	738
		2	65.80	3.888	1846	12130	47.70	37.25		0.30	10.30	1176	16.01	758
		3	65.80	3.922	1846	12220	47.28	37.15		0.30	9.98	1222	16.41	744
		4	65.77	3.942	1842	12280	47.50	37.20		0.30	10.15	1210	16.25	755
23	113	1	62.00	1.532	947	2450	55.40	48.25	45.3	0.06	7.11	344	5.84	419
		2	61.98	1.701	944	2710	54.40	48.25		0.06	6.12	443	5.73	403
		3	61.93	1.932	947	2950	54.00	48.30		0.06	5.67	525	7.01	421
		4	61.93	1.818	947	2900	54.00	48.30		0.06	5.67	513	7.02	413

(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)	(12)	(13)	(14)	(15)
June														
23	114	1	67.76	3.272	5080	28000	50.75	37.17	34.75	0.69	13.23	2110	15.03	1863
		2	67.75	3.278	5020	27780	51.50	37.17		0.68	13.99	1861	15.27	1820
		4	67.75	3.136	5060	26760	51.25	37.25		0.66	13.57	1857	14.60	1832
23	115	1	63.10	3.422	1445	8350	47.80	37.35	34.9	0.20	10.35	806	13.79	606
		2	63.10	3.653	1422	8760	46.95	37.45		0.21	9.40	934	14.22	616
		3	63.00	3.642	1422	8450	46.35	34.45		0.21	8.80	951	14.72	573
		4	63.00	3.740	1422	8960	46.30	37.26		0.22	8.93	1007	14.72	608
23	116	1	63.40	3.431	1460	8450	47.85	37.30	35.0	0.21	10.44	808	13.73	616
		2	63.45	3.689	1446	8920	47.02	37.20		0.22	9.71	918	14.50	615
		3	63.50	3.679	1470	9120	46.65	37.20		0.22	9.34	978	14.90	612
		4	63.50	3.735	1470	9260	46.92	37.25		0.22	9.56	947	14.63	632
23	117	1	62.05	2.338	5080	20020	49.38	37.80	35.0	0.49	11.83	1696	11.16	1795
		2	62.05	2.426	5000	20440	49.95	37.30		0.49	12.40	1660	10.68	1911
		3	62.00	2.383	5010	19960	49.40	37.25		0.49	11.90	1677	11.17	1790
		4	62.00	2.351	4970	19700	49.55	37.20		0.49	12.10	1650	11.02	1790
23	118	1	61.85	1.823	2480	7610	52.80	42.82	40.0	0.19	9.68	772	8.05	945
		2	61.85	2.061	2450	8520	52.15	42.82		0.21	9.23	924	8.56	995
		3	61.85	2.072	2450	8580	51.60	42.82		0.21	8.68	986	9.11	940
		4	61.85	2.101	2430	8630	51.70	42.82		0.21	8.78	982	9.00	958
25	119	1	66.95	1.559	4720	12420	58.90	48.07	45.0	0.30	10.66	1163	7.12	1746
		2	66.95	1.699	4720	13520	58.42	48.02		0.33	10.23	1323	7.52	1798
		3	67.00	1.706	4700	13690	58.05	47.95		0.33	9.93	1369	7.94	1725
		4	66.95	1.712	4700	13620	58.20	47.90		0.33	10.13	1348	7.74	1760
25	120	1	68.80	3.697	5460	34000	48.60	33.10	31.5	0.84	15.08	2250	17.95	1695
		4	68.80	3.622	5440	33300	48.83	33.10		0.82	15.32	2170	17.75	1675
25	121	2	68.80	3.715	5490	34100	49.15	33.10	31.5	0.84	15.53	2180	17.38	1961
		4	68.80	3.821	5440	35100	48.95	33.10		0.84	15.45	2270	17.52	2000
25	122	3	68.68	3.613	5490	33500	48.83	33.10	31.5	0.83	15.31	2185	17.84	1900
		4	68.68	3.632	5440	33400	49.02	33.10		0.83	15.50	2150	17.44	1914

(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)	(12)	(13)	(14)	(15)
June														
25	123	3	60.07	2.066	5650	19420	49.30	37.25	35.0	0.48	11.81	1641	9.51	2040
		4	60.07	2.065	5620	19560	49.55	37.25		0.48	12.06	1622	9.25	2110
25	124	3	61.23	2.218	4850	18100	49.35	37.30	35.0	0.48	11.81	1530	10.53	1718
		4	61.25	2.327	4820	18960	49.55	37.30		0.48	12.01	1577	10.30	1841
25	125	3	65.20	3.040	3620	18550	49.67	37.30	35.0	0.48	12.13	1530	13.77	1349
		4	65.20	3.101	3600	18840	49.85	37.30		0.48	12.31	1530	13.56	1390
25	126	3	69.08	4.027	2585	17530	49.55	37.30	35.0	0.48	12.01	1462	17.26	1018
		4	69.00	4.136	2570	17910	49.70	37.30		0.48	12.16	1475	16.99	1054
27	127	1	44.73	0.394	2390	1430	43.00	36.90	34.4	0.04	6.08	261	1.51	947
		2	44.78	0.481	2380	1742	42.53	36.90		0.05	5.61	345	1.99	876
		3	44.80	0.527	2380	1910	41.95	36.90		0.05	5.03	421	2.57	742
		4	44.84	0.562	2365	2025	41.95	36.95		0.05	4.98	452	2.59	782
28	128	1	45.95	1.264	2100	4040	39.25	32.63	31.0	0.09	6.37	705	6.03	672
		2	45.92	1.120	2080	3540	40.20	32.63		0.09	7.32	538	5.04	702
		3	45.92	1.141	2080	3590	39.34	32.63		0.09	6.97	574	5.40	665
		4	45.92	1.129	2080	3570	39.02	32.63		0.09	7.05	563	5.36	665
28	129	1	45.94	1.102	2100	3510	39.93	32.63	31.0	0.09	7.06	552	5.42	648
		2	45.94	1.069	2080	3380	40.38	32.63		0.09	7.51	502	4.99	677
		3	45.94	1.141	2080	3445	39.75	32.63		0.09	6.98	582	5.57	618
		4	45.94	1.163	2080	3670	39.75	32.63		0.09	6.88	594	5.56	648
28	130	1	45.94	1.309	2100	4170	38.83	32.63	31.0	0.10	5.95	780	6.41	650
		2	45.97	1.157	2080	3660	39.75	32.63		0.10	6.87	592	5.59	655
		3	45.94	1.155	2080	3640	39.70	32.63		0.10	6.82	595	5.61	649
		4	45.94	1.127	2080	3570	39.90	32.63		0.09	7.02	565	5.43	657
28	131	1	45.98	0.932	2100	3300	40.90	32.80	31.0	0.08	8.06	411	4.58	720
		2	45.98	1.157	2080	4060	39.60	32.80		0.10	6.75	603	6.01	676
		3	45.98	1.217	2080	4270	39.38	32.80		0.10	6.53	655	6.21	687
		4	45.98	1.226	2080	4300	39.50	32.80		0.10	6.65	647	6.09	706

(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)	(12)	(13)	(14)	(15)
June														
29	132	1	47.72	1.168	2075	4080	41.40	32.70	31.0	0.10	8.65	472	5.69	718
		2	47.68	1.377	2080	4320	40.55	32.70		0.12	7.79	620	6.39	754
		3	47.68	1.431	2080	5020	40.13	32.70		0.12	7.37	683	6.78	740
		4	47.68	1.427	2085	5020	40.35	32.75		0.12	7.54	668	6.54	768
29	133	1	47.65	1.220	2075	4270	41.05	32.70	31.0	0.11	8.30	515	5.93	720
		2	47.65	1.307	2080	4580	40.84	32.65		0.12	8.13	564	6.10	752
		3	47.65	1.379	2080	4840	40.15	32.65		0.12	7.44	652	6.76	715
		4	47.65	1.402	2085	4920	40.25	32.65		0.12	7.54	654	6.64	741
29	134	1	47.70	1.352	2075	4730	40.38	32.67	31.0	0.12	7.71	613	6.59	718
		2	47.70	1.310	2080	4610	40.85	32.67		0.12	8.12	567	6.14	751
		3	47.70	1.359	2080	4760	40.33	32.65		0.12	7.62	626	6.68	713
		4	47.70	1.368	2085	4800	40.38	32.65		0.12	7.67	627	6.58	729
29	135	1	47.68	1.435	2075	5030	39.98	32.65	31.0	0.13	7.27	684	6.92	727
		2	47.68	1.328	2080	4650	40.30	32.70		0.12	8.04	580	6.16	754
		3	47.68	1.354	2080	4750	40.25	32.65		0.12	7.54	631	6.70	709
		4	47.70	1.348	2085	4730	40.20	32.60		0.12	7.54	629	6.77	717
29	136	1	47.65	1.494	2075	5240	39.65	32.70	31.0	0.13	6.89	762	7.19	729
		2	47.65	1.341	2080	4700	40.30	32.70		0.12	8.04	586	6.12	768
		3	47.68	1.348	2080	4720	40.35	32.65		0.12	7.64	618	6.60	715
		4	47.68	1.318	2085	4630	40.35	32.60		0.12	7.69	604	6.61	701
29	137	1	47.71	1.490	2075	5220	39.70	32.65	31.0	0.13	6.99	750	7.19	727
		2	47.71	1.344	2080	4720	40.30	32.65		0.12	8.09	584	6.18	764
		3	47.71	1.341	2080	4700	40.37	32.65		0.12	7.66	615	6.61	712
		4	47.71	1.305	2085	4570	40.60	32.65		0.12	7.89	580	6.40	714
30	138	1	45.93	1.247	2060	4330	38.67	32.65	31.0	0.10	6.19	702	6.27	692
		2	45.93	1.100	2060	3820	39.84	32.65		0.09	7.14	536	5.50	694
		3	45.93	1.121	2060	3890	39.50	32.65		0.09	6.80	573	5.83	668
		4	45.93	1.114	2060	3880	39.50	32.65		0.09	6.80	572	5.84	665

(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)	(12)	(13)	(14)	(15)
June														
30	139	1	45.93	1.316	2060	4570	38.30	32.70	31.0	0.10	5.85	732	6.63	689
		2	45.95	1.143	2060	3980	39.34	32.70		0.09	7.09	562	5.50	723
		3	45.95	1.130	2060	3930	39.50	32.70		0.09	6.75	583	5.84	672
		4	45.95	1.100	2060	3830	39.65	32.70		0.09	6.90	556	5.71	671
July														
2	140	1	72.42	2.020	4080	13900	62.10	48.96	45.5	0.34	12.97	1073	9.14	1538
		2	72.42	2.200	4060	15050	61.35	49.01		0.37	12.66	1191	9.30	1620
		3	72.45	2.245	4070	15400	61.25	49.01		0.38	12.05	1283	9.89	1558
		4	72.45	2.264	4100	15600	61.25	48.96		0.38	12.10	1290	9.88	1578
2	141	1	72.42	2.167	4080	14930	61.40	49.00	45.5	0.37	12.22	1221	9.75	1533
		2	72.42	2.236	4060	15370	61.66	49.00		0.38	12.47	1232	9.45	1625
		3	72.45	2.223	4070	15280	61.07	49.00		0.38	11.38	1287	10.08	1522
		4	72.45	2.228	4100	15400	61.25	49.00		0.38	12.06	1280	9.90	1555
2	142	1	72.45	2.291	4080	15780	60.60	49.10	45.5	0.39	11.30	1395	10.48	1507
		2	72.45	2.293	4060	15730	61.45	49.00		0.39	12.35	1233	9.66	1638
		3	72.45	2.194	4070	15100	61.06	48.95		0.37	11.93	1267	10.11	1494
		4	72.45	2.196	4100	15200	61.20	48.95		0.37	12.07	1260	9.97	1530
2	143	1	72.45	2.356	4080	16220	60.26	48.92	45.5	0.40	11.14	1460	10.81	1500
		2	72.45	2.288	4060	15720	61.45	49.02		0.39	12.24	1286	9.67	1626
		3	72.45	2.178	4070	14930	61.04	48.97		0.37	11.89	1256	10.13	1475
		4	72.45	2.185	4100	15120	61.18	48.92		0.37	12.08	1252	9.99	1515
2	144	1	69.05	1.420	5130	12280	61.90	48.85	45.5	0.30	12.90	953	6.29	1953
		2	69.05	1.632	5110	14040	61.26	48.85		0.34	12.24	1150	6.81	2060
		3	69.00	1.658	5110	14270	60.82	48.85		0.35	11.80	1211	7.07	2010
		4	69.00	1.640	5120	14160	60.78	48.85		0.35	11.76	1206	7.22	1960
2	145	1	70.27	1.618	4300	11730	61.90	48.85	45.5	0.29	12.90	910	7.42	1580
		2	70.27	1.841	4280	13300	61.22	48.90		0.33	12.16	1098	7.96	1673
		3	70.27	1.924	4280	13900	60.55	48.90		0.34	11.48	1213	8.59	1608
		4	70.30	1.904	4300	13750	60.60	48.85		0.34	11.58	1192	8.58	1603

(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)	(12)	(13)	(14)	(15)
2	145	1	72.05	2.048	3230	11300	61.63	48.85	45.5	0.29	12.63	897	9.24	1225
		2	72.05	2.351	3220	12630	60.92	48.85		0.33	11.91	1062	9.81	1288
		3	72.05	2.351	3220	12740	60.24	48.80		0.34	11.27	1133	10.47	1218
		4	72.05	2.347	3230	12770	60.34	48.80		0.34	11.37	1125	10.37	1232
2	147	1	76.30	3.060	2165	11150	61.80	48.85	45.5	0.30	12.80	872	12.72	876
		2	76.30	3.388	2165	12330	60.95	48.85		0.33	11.93	1038	13.38	923
		3	76.15	3.484	2165	12500	60.33	48.85		0.34	11.31	1106	13.94	897
		4	76.15	3.413	2165	12440	60.33	48.85		0.34	11.31	1101	13.94	892
2	148	1	78.45	4.459	4850	36200	54.95	36.24	33.9	0.89	18.16	1997	20.84	1738
		4	78.45	4.464	4850	36400	54.40	36.07		0.89	17.88	2040	21.28	1708
2	149	1	78.45	4.649	4860	38100	53.95	34.00	33.9	0.94	19.48	1960	21.71	1753
		4	78.45	4.703	4850	38500	53.05	34.00		0.94	18.56	2075	22.60	1702
4	150	1	70.37	1.698	4510	12360	61.90	48.98	45.5	0.29	12.63	969	7.53	1653
		2	70.37	1.885	4290	13670	60.92	48.98		0.33	11.78	1160	8.41	1623
		3	70.37	1.859	4290	13530	61.00	49.03		0.34	11.78	1150	8.34	1623
		4	70.37	1.827	4320	13300	61.22	49.03		0.34	12.00	1109	8.14	1634
2	151	1	70.35	1.672	4270	12000	61.13	46.22	45.5	0.30	12.76	945	6.13	1494
		2	70.35	1.870	4250	13420	60.05	46.15		0.33	11.73	1143	6.09	1478
		3	70.35	1.854	4320	12500	60.10	46.22		0.33	11.71	1130	6.03	1493
		4	70.35	1.822	4260	12130	60.30	46.22		0.32	11.92	1105	6.88	1490
2	152	1	73.20	1.704	4300	19600	59.50	45.25	42.4	0.48	14.01	1400	12.71	1541
		2	73.20	1.837	4240	18860	59.55	45.25		0.46	13.87	1363	12.30	1536
		3	73.20	1.834	4300	18370	59.45	45.25		0.45	13.97	1317	12.25	1500
		4	73.20	1.813	4290	18170	59.65	45.25		0.45	14.17	1283	12.06	1507
2	153	1	70.35	1.119	4270	15230	59.45	45.80	43.0	0.40	13.65	1195	9.87	1545
		2	70.35	1.245	4250	16110	59.15	45.85		0.40	13.10	1230	10.38	1555
		3	70.35	1.135	4300	15820	59.15	45.85		0.39	13.10	1210	10.40	1520
		4	70.35	1.134	4260	15950	59.35	45.85		0.39	13.50	1200	10.23	1556

(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)	(12)	(13)	(14)	(15)
July														
13	154	1	78.48	3.649	5260	32400	58.30	39.65	37.4	0.80	18.25	1776	17.95	1805
13	155	1	78.48	3.647	5260	32400	58.40	39.65	37.4	0.80	18.35	1769	17.86	1814
13	156	1	78.48	3.665	5260	32550	58.40	39.65	37.4	0.80	18.35	1774	17.85	1823
13	157	1	78.48	3.689	5260	32700	58.35	39.65	37.4	0.80	18.30	1790	17.88	1829
13	158	1	70.19	2.871	2780	13470	55.70	40.90	38.5	0.33	14.63	919	12.88	1045
13	159	1	70.19	2.912	2780	13650	55.46	40.80	38.5	0.33	14.43	945	13.16	1037
13	160	1	70.19	2.957	2780	13860	55.06	40.85	38.5	0.34	14.04	985	13.48	1027
13	161	1	70.15	2.986	2780	14030	54.90	40.90	38.5	0.34	13.83	1030	13.59	1032
13	162	1	70.90	2.365	1771	7080	60.00	48.70	46.0	0.17	11.21	629	9.63	736
13	163	1	70.90	2.419	1771	7240	59.55	48.70	46.0	0.18	10.76	671	10.05	721
13	164	1	70.90	2.493	1771	7470	59.16	48.75	46.0	0.18	10.34	720	10.40	718
13	165	1	70.88	2.558	1771	7550	58.82	48.65	46.0	0.19	10.07	760	10.68	707
13	166	1	70.81	1.996	966	3250	62.54	54.24	52.0	0.08	8.25	394	7.23	449
13	167	1	70.81	2.174	946	3470	61.73	54.10	52.0	0.08	7.59	457	7.95	437
13	168	1	70.81	2.297	946	3660	61.10	54.15	52.0	0.09	6.90	530	8.51	430
13	169	1	70.80	2.416	966	3930	60.65	54.10	52.0	0.10	6.50	605	8.89	442
14	170	1	70.20	3.048	2782	14320	54.65	40.90	38.8	0.34	13.58	1055	13.86	1033
14	171	1	70.20	3.090	2782	14520	54.36	40.95	38.8	0.35	13.23	1094	14.13	1028

(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)	(12)	(13)	(14)	(15)
July														
14	172	1	70.94	2.640	1804	8030	58.58	48.50	46.0	0.20	9.78	821	10.94	724
14	173	1	70.94	2.708	1804	8250	58.19	48.55	46.0	0.21	9.33	883	11.29	721
14	174	1	70.80	2.541	966	4140	60.18	53.97	52.0	0.11	5.95	696	9.29	447
14	175	1	70.80	2.592	966	4230	59.76	53.97	52.0	0.12	5.53	766	9.68	438
19	176	1	78.48	3.508	5290	31300	59.55	39.74	37.4	0.77	19.42	1610	16.79	1865
19	177	1	70.21	2.806	2782	13160	56.01	40.92	38.5	0.31	14.38	881	12.64	1040
19	178	1	70.90	2.338	1778	7050	60.28	48.73	46.0	0.18	11.46	615	9.36	752
19	179	1	70.91	1.951	994	3270	62.94	54.25	52.0	0.08	8.65	378	6.95	470
June														
23	180	1	61.97	0.346	1493	874	60.45	55.00	52.7	0.02	5.45	160	1.34	653
		2	62.05	0.356	1475	884	60.50	55.27	52.8	0.02	5.12	173	1.47	601
		3	62.12	0.420	1485	1052	59.87	55.28	52.9	0.02	4.54	232	2.03	518
		4	62.13	0.490	1475	1219	59.72	55.31	60.0	0.02	4.40	257	2.16	563

TABLE III
GAS INJECTION CALCULATIONS

Run No.	Manometer Rdg. In Glycol	Rdg. In Hg.	Pressure at Orifice p.s.i.g.	Temp. at Orifice OF	Pressure in Evap. p.s.i.g.	Temp. in Evap. OF	Volume Saturated Vapor Cu.ft/hr.
128	17.45		32.8	103	31.0	32.65	103.5
129	5.59		32.1	106	31.0	32.65	61.0
130	27.55		34.0	106	31.0	32.65	130.5
133	2.02		31.3	93	31.0	32.65	38.4
134	8.10		32.0	97	31.0	32.65	73.0
135	14.58		33.0	95	31.0	32.65	96.8
136	26.64		34.4	98	31.0	32.65	131.0
137	18.41		78.0	108	31.0	32.65	157.2
138	28.03		34.4	99	31.0	32.65	131.4
139	25.91		79.6	104	31.0	32.65	186.2
141	4.79		46.4	103	45.5	49.00	50.7
142	14.25		47.3	110	45.5	49.00	87.9
143	26.97		48.8	115	45.5	49.00	106.0
152	17.50		44.8	103	42.4	59.48	159.0
153	4.16		43.6	110	43.0	59.32	81.8
155	3.53		37.8	104	37.4	39.65	46.9
156	10.06		38.5	110	37.4	39.65	78.5
157	21.66		40.0	115	37.4	39.65	110.0
159	3.55		38.8	100	38.5	40.80	46.1
160	10.68		39.9	105	38.5	40.85	77.6
161	21.88		41.0	108	38.5	40.90	113.2
163	3.92		46.4	105	46.0	48.70	47.1
164	12.22		47.8	105	46.0	48.75	78.4
165	25.31		49.3	107	46.0	48.65	110.0
167	4.65		52.3	107	52.0	54.10	47.3
168	13.66		53.9	107	52.0	54.15	78.2
169	28.43		55.9	100	52.0	54.10	106.1
170		4.60	45.0	110	38.8	40.90	173.5
171		7.74	52.3	114	38.8	40.95	234.0
172		5.17	53.6	100	46.0	48.70	172.5
173		9.03	63.0	103	46.0	48.75	250.4
174		6.02	61.0	98	52.0	54.17	176.5
175		10.17	70.5	105	52.0	54.17	217.0

TABLE IV
CROSS PLOT CALCULATIONS

ΔT_b OF	Rate of Injection Cu. ft./hr					
	0	47	78	110	175	235
6	h 245 % Increase	357 45.7	483 97.0	591 141.0	700 186.0	780 218.0
8	374	485 29.7	590 57.7	668 78.6	756 104.0	830 122.0
10	521	618 18.6	700 34.4	760 45.8	835 60.3	900 72.7
13	774	835 7.8	896 15.8	948 22.5	1030 33.1	1075 38.9
16	1120	1165 4.0	1230 9.8	1275 13.8	1350 20.6	1400 25.0
18	1600			1675 4.7	1720 7.5	1740 8.75

TABLE V
CALCULATION OF INSIDE COEFFICIENTS

Run No.	Water Velocity ft/sec.	Avg. Water Temp. OF	h_1 BTU/hr OF sq.ft. (inside area)	$\left(\frac{h_1 D}{k}\right)$	$\left(\frac{C \mu}{k}\right)^{0.3}$	$\left(\frac{h_1 D}{k Pr}\right)^{0.3}$	$\frac{DV \rho}{\mu}$
100	7.15	65.44	1561	237	1.820	130.1	33000
101	5.24	65.78	1184	179	1.818	98.5	24300
102	9.96	66.55	2210	334	1.812	184.3	46700
103	11.50	65.13	2240	339	1.820	186.2	52800
104	1.94	62.89	480	73	1.848	39.5	8550
106	4.84	63.26	1090	166	1.840	89.9	21700
107	1.86	62.50	452	69	1.847	37.2	8240
108	11.70	65.44	2350	356	1.818	195.6	54200
109	3.36	62.66	793	121	1.843	65.4	14980
110	1.94	63.72	502	76	1.835	41.5	8750
111	5.81	64.01	1267	192	1.832	104.6	26300
112	3.92	63.85	905	137	1.834	75.0	17720
113	2.01	61.10	501	76	1.864	40.8	8610
114	10.71	66.16	2220	335	1.812	185.0	50200
116	3.03	61.26	748	114	1.861	61.2	13820
117	10.63	61.97	2170	330	1.850	178.5	46800
118	5.21	60.84	1162	177	1.860	95.3	22600
119	10.00	66.12	2120	323	1.812	178.2	46600
120	11.57	66.97	2280	344	1.807	190.3	54400
123	11.93	59.04	2510	384	1.877	205.0	54400
124	10.26	60.10	2150	328	1.865	175.6	44000
125	7.66	63.63	1655	252	1.836	137.3	34500
126	5.47	67.00	1252	189	1.812	104.3	25600
127	5.06	44.74	1012	162	2.042	78.2	17000
128	4.43	45.33	818	128	2.032	62.8	15030
138	4.37	45.36	841	131	2.032	64.6	14820
140	8.61	71.83	1905	286	1.769	161.6	43500
144	10.86	68.22	2410	363	1.798	202.0	52100
145	9.12	65.37	1954	296	1.818	162.5	42200
146	6.85	76.91	1500	225	1.737	126.6	34100
147	4.59	74.52	1085	162	1.753	92.3	23900
148	10.30	76.22	2090	312	1.742	179.5	54800
150	9.16	69.36	1970	297	1.790	166.0	44600
151	9.08	69.32	1793	270	1.788	151.2	44300
152	9.12	71.90	1840	276	1.770	156.1	46000
153	9.08	69.75	1867	281	1.785	157.4	44500
154	11.15	76.66	2210	330	1.730	190.6	59500
158	5.91	68.74	1252	189	1.793	105.3	28600
162	3.76	69.70	870	131	1.785	73.4	18400
166	2.03	69.81	531	80	1.785	44.8	9940
170	5.91	68.68	1244	188	1.795	104.5	28500
172	3.83	69.60	873	131	1.786	73.6	18730
174	2.05	69.52	523	78	1.786	44.1	10030
176	11.21	76.73	2250	335	1.731	192.4	59900
177	5.91	68.81	1260	190	1.793	105.7	28600
178	3.76	69.73	913	138	1.785	76.9	18400
179	2.11	69.93	571	86	1.784	48.2	10340

TABLE VI

WILSON PLOT CALCULATIONS

Run No.	Tube No.	Average Water Temp. of	Freon Temp. of	ΔT of	Heat Flux BTU/hr. sq.ft.	Water Velocity ft./sec.	U BTU/hr. sq.ft. of	$V^{0.88}$	$\frac{1}{U} \times 10^3$	$\frac{1}{V^{0.88}}$
123	3	59.05	37.25	21.80	19420	11.36	891	8.88	1.123	0.1126
	4	59.04	37.25	21.79	19560	11.90	897	8.88	1.115	0.1126
124	3	60.12	37.30	22.82	18100	10.28	794	7.76	1.260	0.1290
	4	60.14	37.30	22.79	18960	10.23	832	7.76	1.203	0.1290
125	3	63.68	37.30	26.38	18550	7.67	703	6.00	1.422	0.1667
	4	63.65	37.30	26.35	18840	7.64	715	6.00	1.400	0.1667
126	3	67.07	37.30	29.77	17580	5.48	591	4.47	1.691	0.2235
	4	66.94	37.30	29.64	17910	5.46	605	4.47	1.653	0.2235
144	1	68.34	48.85	19.49	12280	10.90	630	8.20	1.587	0.1220
	2	68.23	48.85	19.38	14040	10.84	725	8.15	1.330	0.1226
	3	68.17	48.85	19.32	14270	10.84	738	8.15	1.356	0.1226
	4	68.18	48.85	19.33	14180	10.87	737	8.17	1.358	0.1223
145	1	69.46	48.85	20.61	11730	9.13	569	7.00	1.758	0.1430
	2	69.35	48.90	20.45	13300	9.10	651	6.97	1.536	0.1434
	3	69.31	48.90	20.41	13900	9.10	681	6.97	1.468	0.1434
	4	69.35	48.85	20.50	13750	9.13	671	7.00	1.490	0.1430
146	1	71.03	48.85	22.18	11300	6.87	510	5.45	1.960	0.1835
	2	70.89	48.85	22.04	12630	6.84	574	5.43	1.741	0.1840
	3	70.88	48.80	22.08	12740	6.84	577	5.43	1.734	0.1840
	4	70.88	48.80	22.08	12770	6.85	578	5.44	1.730	0.1838
147	1	74.67	48.85	25.82	11150	4.59	432	3.83	2.315	0.2610
	2	74.51	48.85	25.66	12350	4.59	481	3.83	2.080	0.2610
	3	74.43	48.85	25.58	12500	4.59	489	3.83	2.025	0.2610
	4	74.44	48.85	25.59	12440	4.59	487	3.83	2.050	0.2610

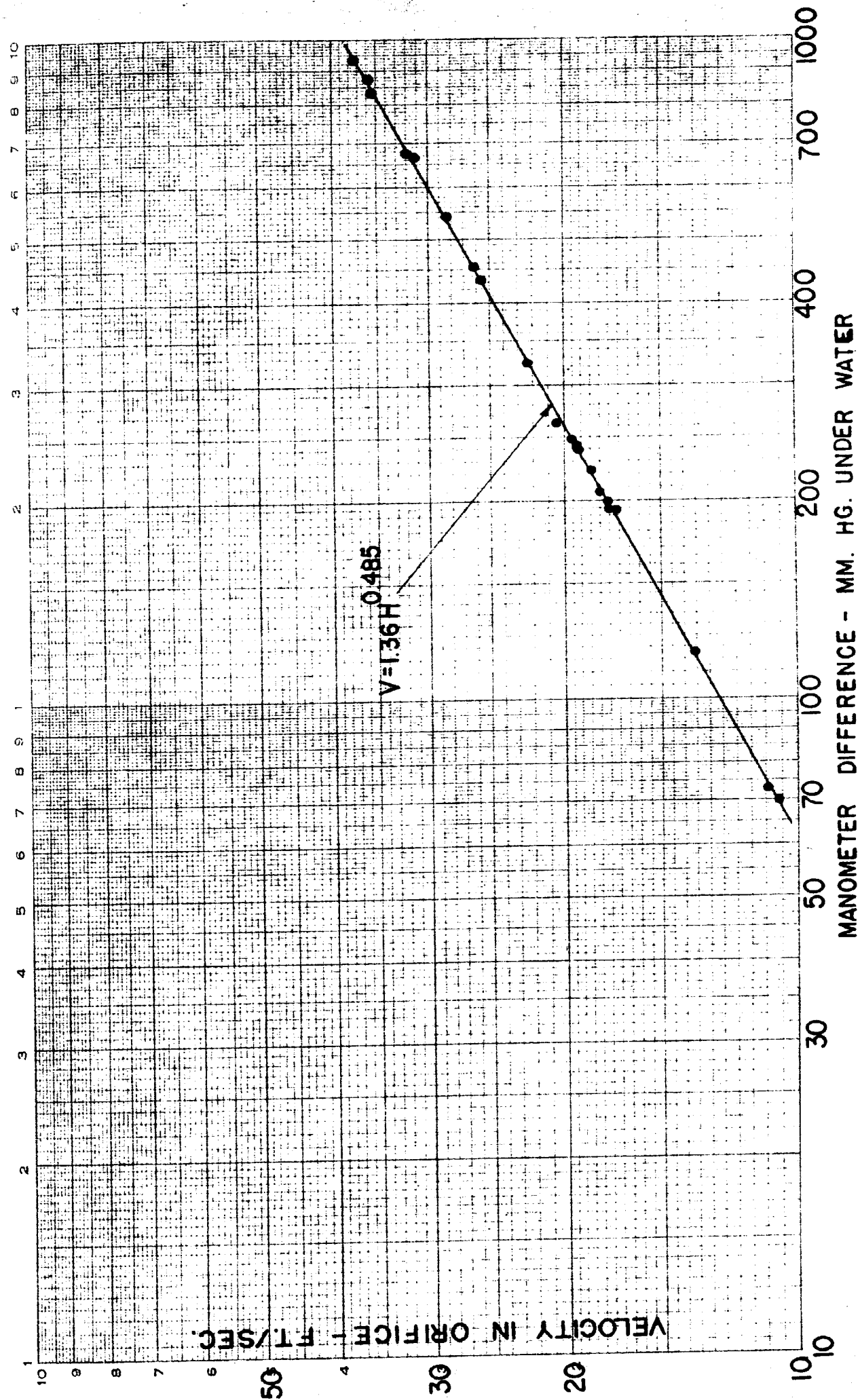


FIG. 28 WATER ORIFICE CALIBRATION

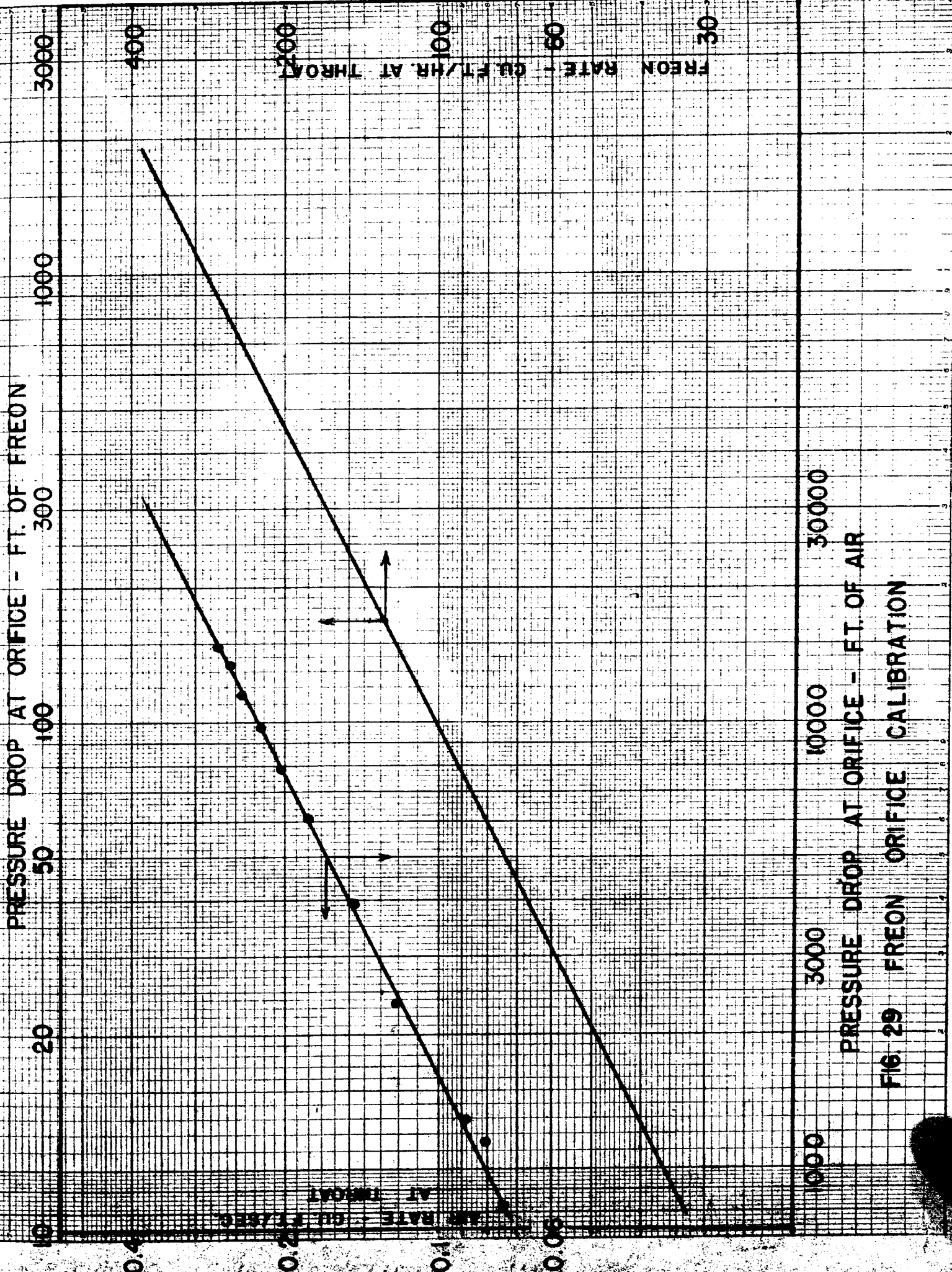


FIG 29 FREON ORIFICE CALIBRATION

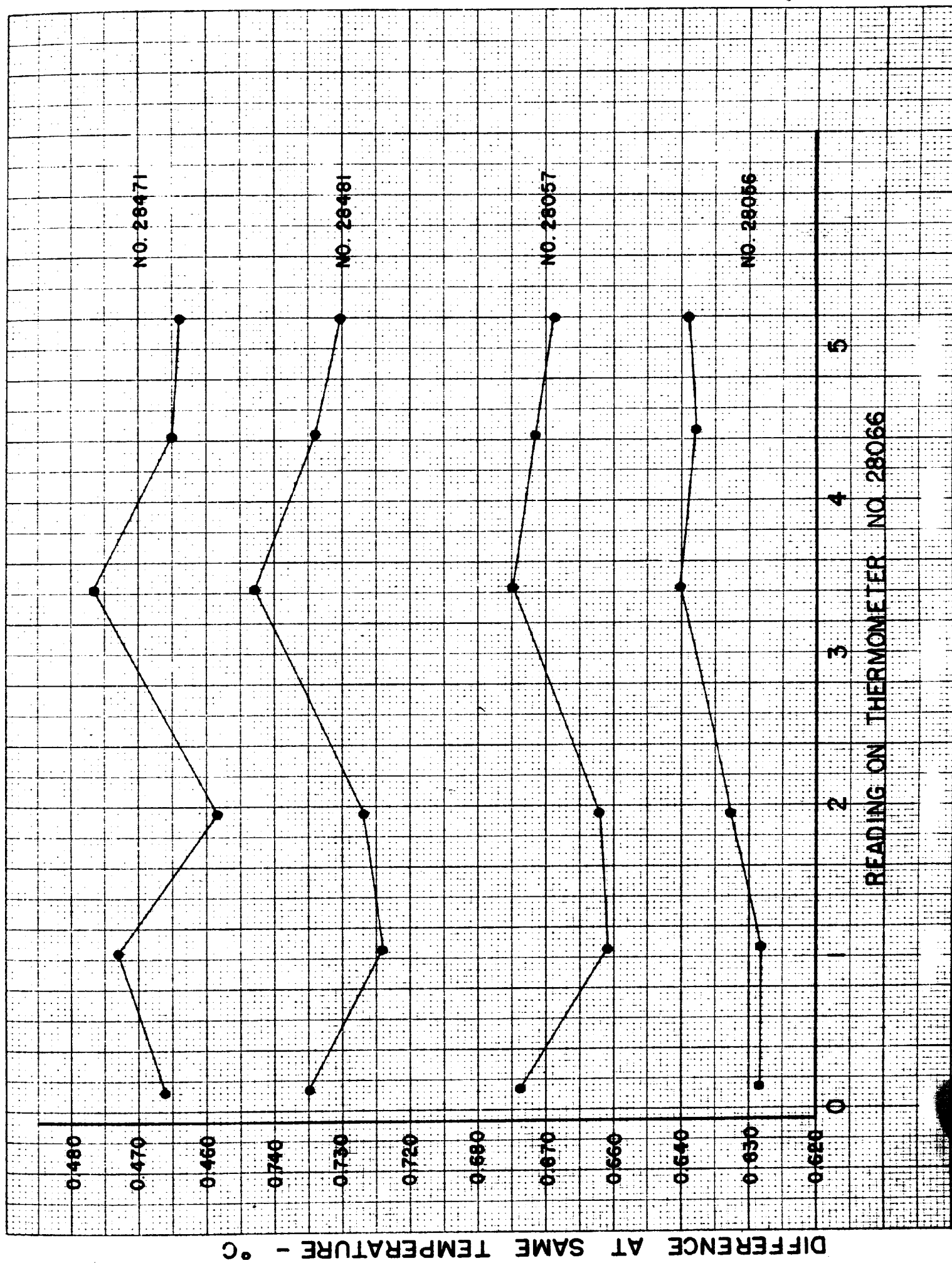


FIG. 30 EXAMPLE BECKMANN CALIBRATION

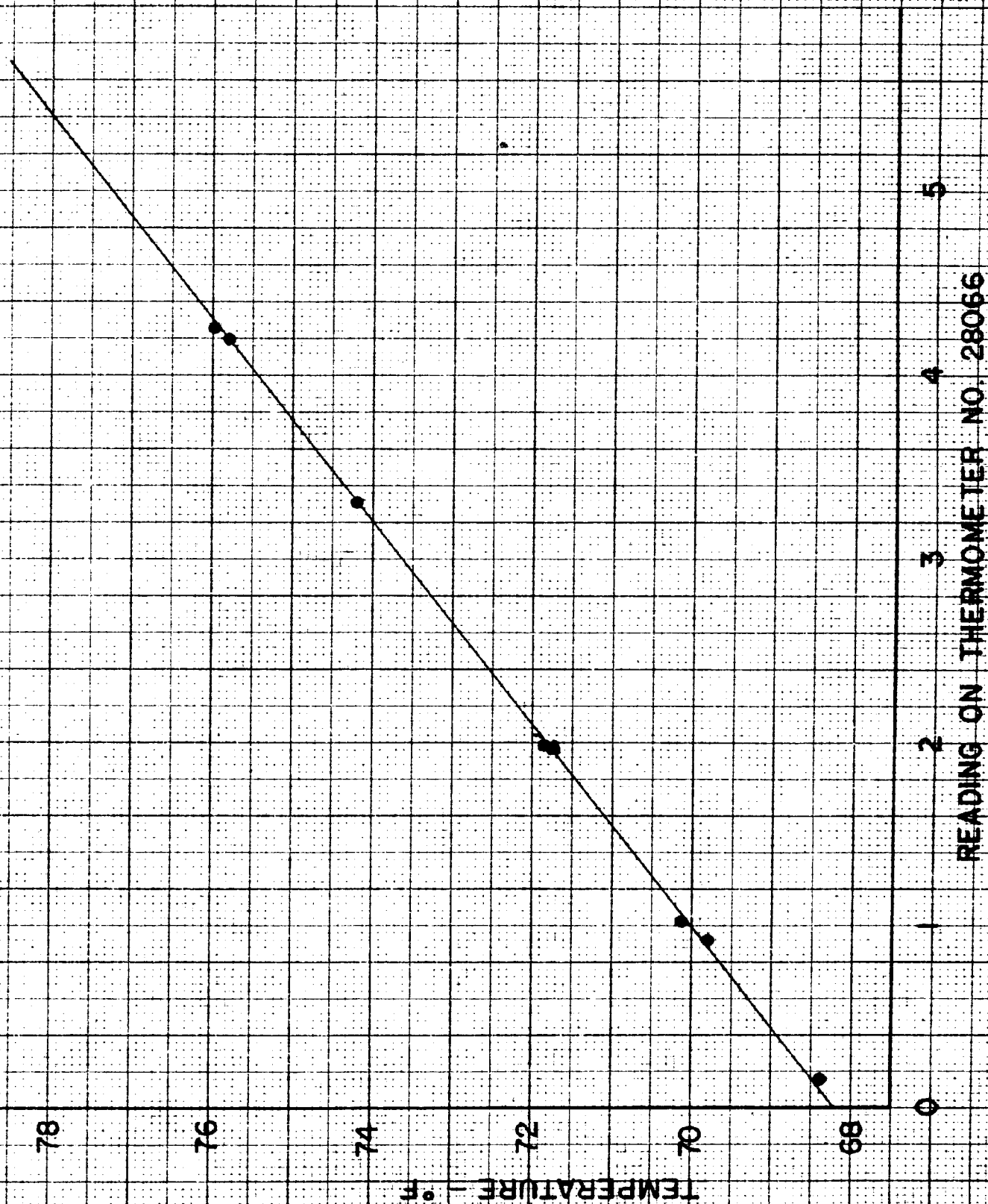


FIG. 31 TEMPERATURE CORRESPONDING TO BECKMANN READING

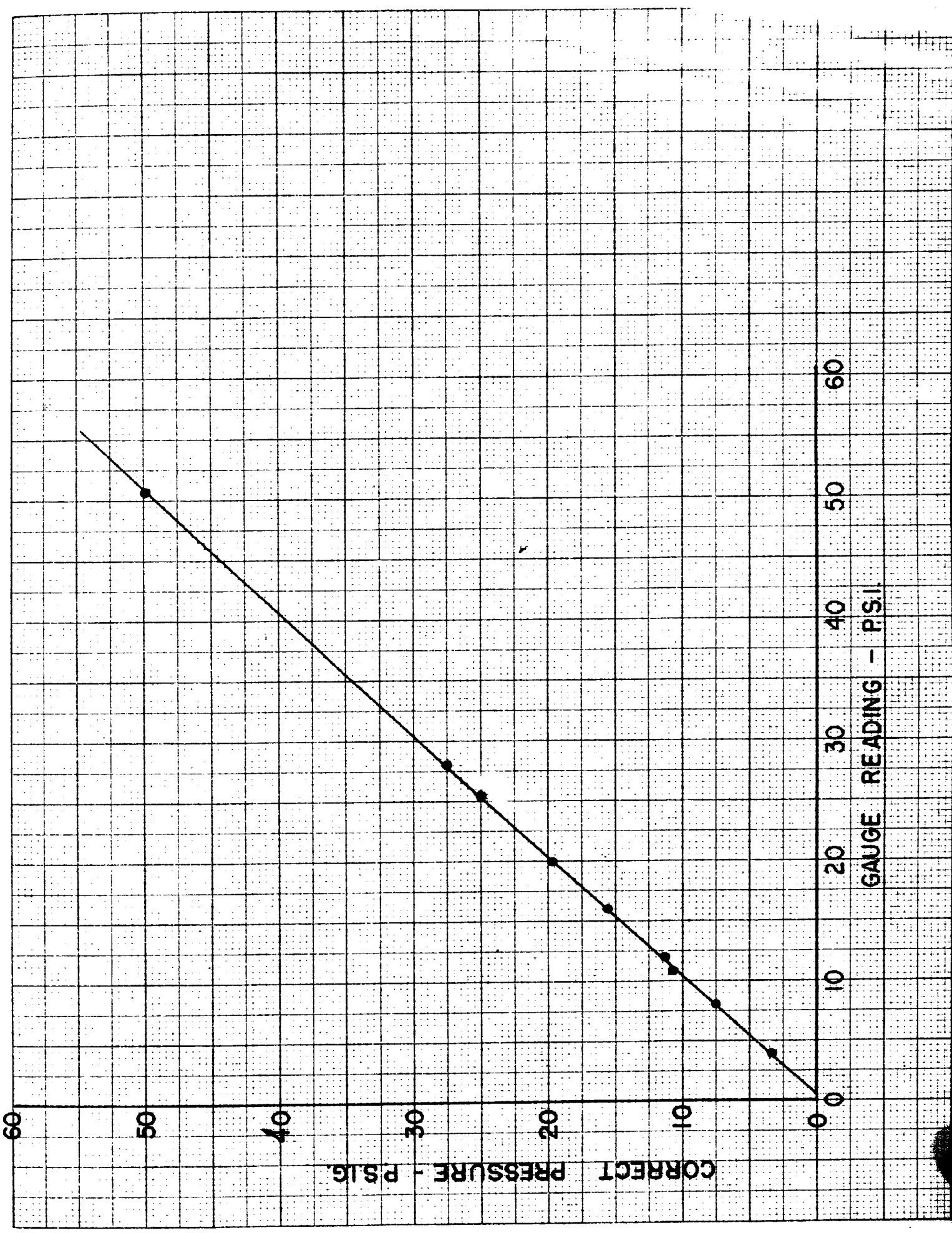


FIG. 32 CALIBRATION FOR GAUGE C2-108

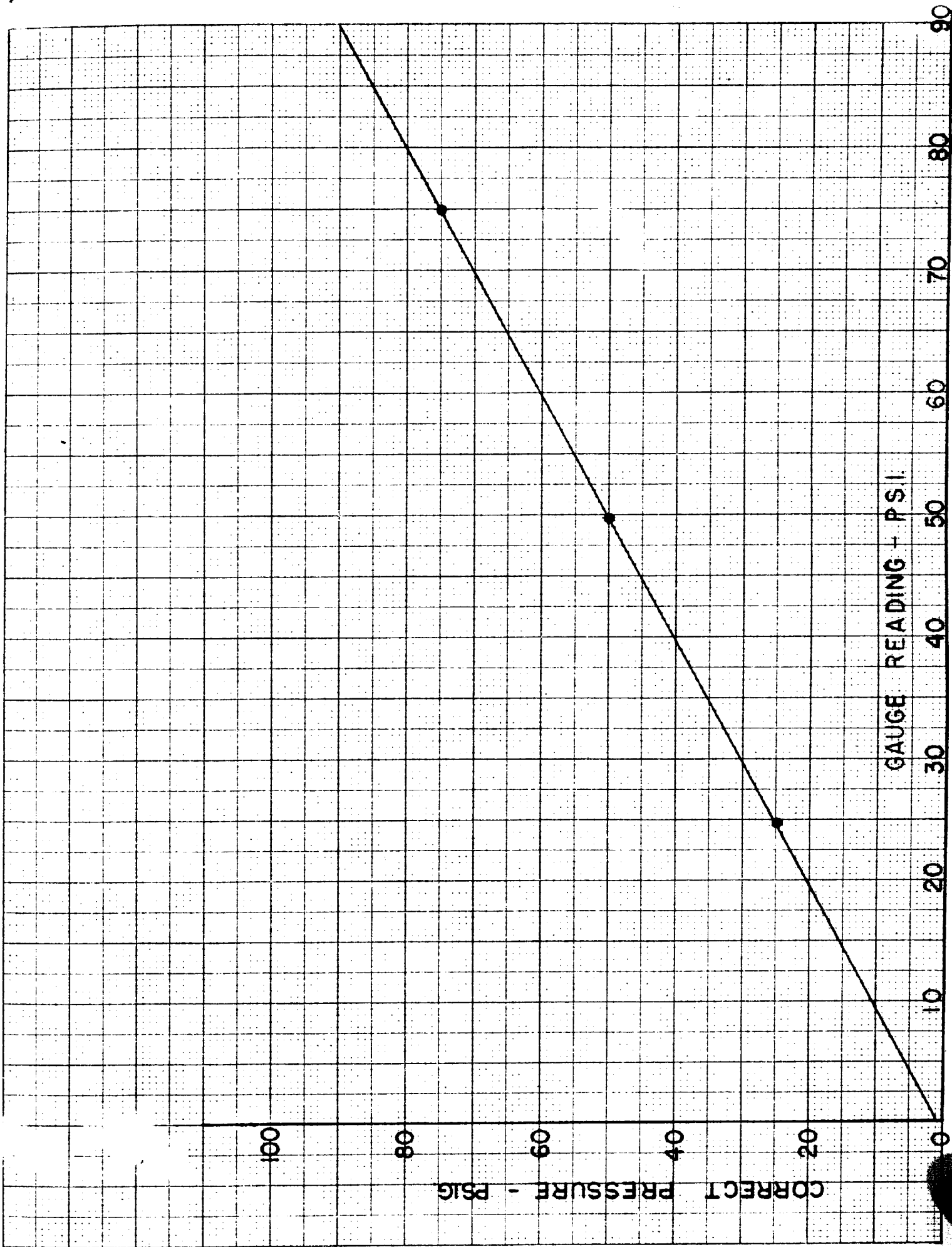


FIG. 33 CALIBRATION FOR GAUGE C2 - 378

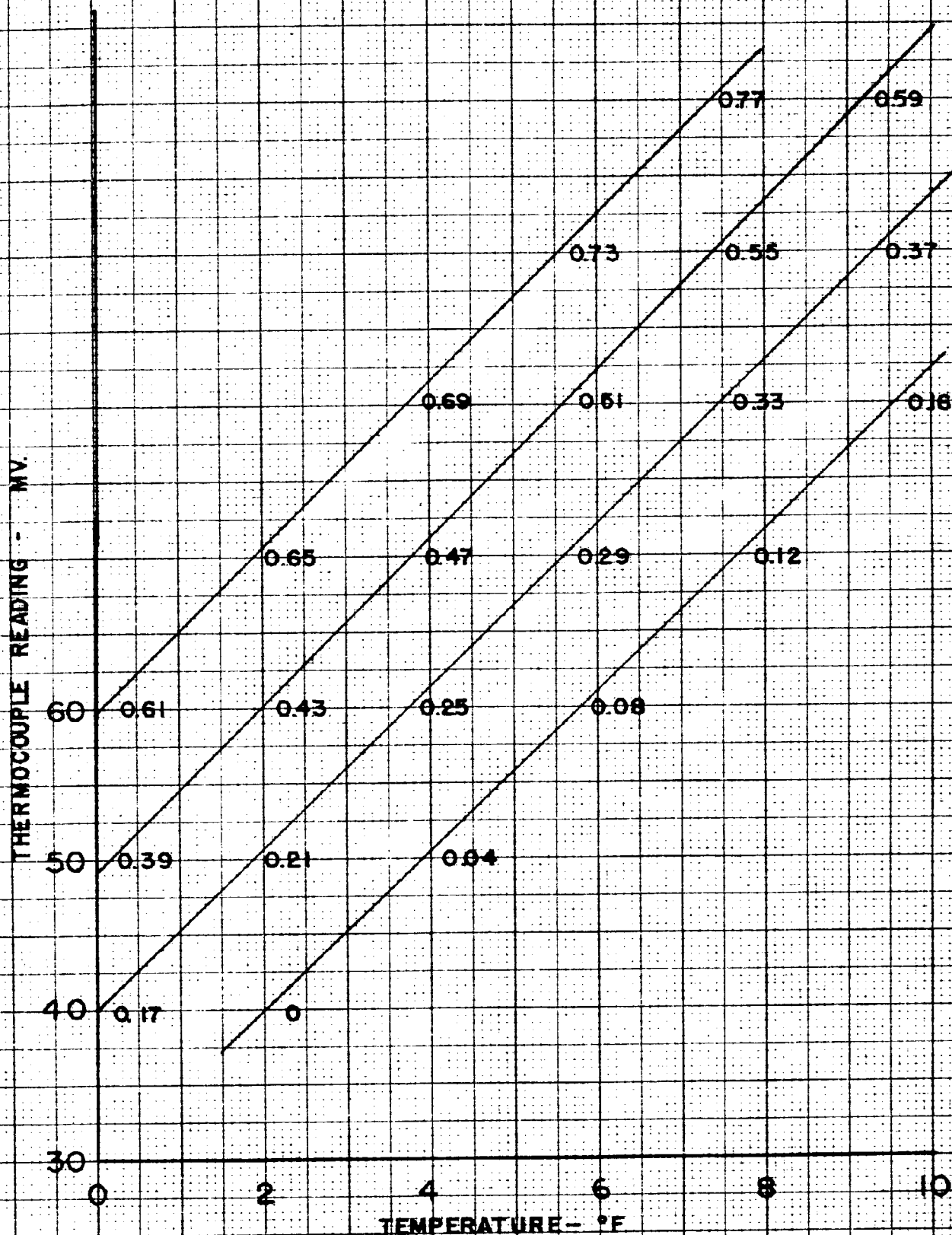


FIG.34 THERMOCOUPLE CONVERSION CHART

XIV. NOMENCLATURE

a	A numerical constant.
b	A numerical constant.
h	Boiling film coefficient, BTU/hr. sq. ft. °F.
h_i	Inside film coefficient, BTU/hr. sq. ft. °F.
k	Thermal conductivity, BTU/hr. sq. ft. °F/ft.
m	A numerical constant.
r_f	Resistant of fouling on tube walls, hr. sq. ft. °F/BTU.
x	Thickness of tube wall, ft.
A	Outside area of tube, sq. ft.
A_i	Inside area of tube, sq. ft.
A_m	Mean area of tube wall, sq. ft.
B	A numerical constant.
C	Specific heat, BTU/lb. °F.
D	Inside diameter of tube, ft.
E	A numerical constant.
Q	Quantity of heat transferred, BTU/hr.
T_1	Inlet temperature of fluid flowing in a tube, °F.
T_2	Outlet temperature of fluid flowing in a tube, °F.
ΔT	Bulk temperature difference between fluids inside and outside a tube, °F.
ΔT_b	Temperature drop across the boiling film, °F.
ΔT_i	Temperature drop across the inside film, °F.
U	Overall heat transfer coefficient, BTU/hr. sq. ft. °F.
V	Velocity of fluid flowing in a tube, ft/sec.
W	Weight rate of flow of fluid in a tube, lb/hr.
μ	Viscosity, lb/ft. hr.
ρ	Density, lb/cu. ft.

XV. BIBLIOGRAPHY CITED

1. Akin, G.A., and W. H. McAdams, A.I.Ch.E., 35, 137, (1939).
2. Ashley, C.M., Journal of the A.S.R.E., 43, No. 2, (1942).
3. Austin, L., Mitteilungen über Forschungsarbeiten auf dem Gebiete des Ingenieurwesens, 7, 75-86, (1903).
4. Baker, E.H., and A.C. Mueller, Ind. Eng. Chem., 29, 1065, (1937).
5. Bonilla, C.F., and C.W. Perry, A.I.Ch.E., 37, 685, (1941).
6. Cryder, D.S., and A.C. Finalborgo, A.I.Ch.E., 33, 346, (1937).
7. Cryder, D.S., and E.R. Gilliland, Ind. Eng. Chem., 24, 1382, (1932).
8. Dittus, F.W., and L. M.K. Boelter, Univ. Calif. Pub. in Eng., 2, 443, (1930).
9. Drew, T.P., and A.C. Mueller, A.I.Ch.E., 33, 449, (1937).
10. Hebbard, G.M., and W.L. Badger, Ind. Eng. Chem. Anal. Ed., 5, 359, (1933).
11. Insinger, T.H. Jr., and H. Bliss, A.I.Ch.E., 36, 491, (1940).
12. Jakob, M., Journal of the A.S.M.E., 58, 648, (1936).
13. Jones, W., Journal of the A.S.R.E., 41, 413, (1941).
14. Mohun, W.A., and W.S. Peterson, Can. Chem. and Proc. Ind., 31, 908, (1947).
15. Pridgeon, L.A., and W.L. Badger, Ind. Eng. Chem., 16, 474, (1924).
16. Wilson, E.E., Trans. Am. Soc. Mech. Eng., 37, 47, (1915).