A STUDY OF THE HEAT TRANSFER CHARACTERISTICS IN A FIN TYPE EVAPORATOR

A THESIS

Submitted for the Degree

of

Master of Science in Mechanical Engineering

by

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The authors take this means of expressing their appreciation to Professor R. S. King, who suggested the research, and to Professor A. D. Holland, whose practical advice and helpful suggestions facilitated its accomplishment. The authors also wish to thank Mr. Willis D. Ludwig and Mr. Wisser, who originally laid out the equipment.

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PREFACE

COMPLEXITY OF PROBLEM

A large number of variables affecting coil performance have been previously reported.*

The following table shows some variables encountered:

AIR:

1. Temperatures: Dry-bulb, wet bulb, dew-point.
2. Velocity: Linear velocity through free area or face area.
3. Turbulence.

REFRIGERANT:

1. Type: Chemical composition, liquid or direct expansion.
2. Operating Range: Pressures and temperatures.
3. Velocity.
4. Turbulence.

COIL:

1. Surface Ratio: Ratio of air-side surface area to refrigerant side surface.

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* See Bibliography in Appendix.

2. Type of fins: Round, square, or continuous.

3. Shape of fins: Plain, crimped, ribbon or wedge-shape.

4. Fin bond: Integral, dipped, expanded, pressed, etc.

5. Material: Copper, Aluminum, steel, cast-iron, brass.

6. Depth and piping: Depth of fins, number of tube rows, tube spacing, counter flow, parallel flow, cross flow, mixed flow, etc.

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I NATURE OF THE STUDY

A. Purpose.

During the last few years there has been very rapid development in the air-conditioning industry. This development has given rise to a number of manufacturers of accessories necessary for a complete unit. As yet, very little of the equipment has been standardized. There has been no accurate basis on which a coil might be rated. The selection of a coil has been at the best a rough estimate. Each manufacturer has given his own specifications. Since this method of rating and selecting a coil for a particular job has not been the desire of both the sales engineer and the installation contractor, it would indeed be practical to have standard, accurate information on which evaporator coils might be rated and selected.

Necessary tables and charts for complete information regarding all types of coils would require an unlimited amount of research, as well as a most elaborate outlay of expensive instruments and equipment. That a single manufacturer or even a group of men interested in research should undertake such a comprehensive problem would be an expensive and impractical project.

Since there has been considerable interest shown in this problem, the authors think it quite feasible that even so complete a situation might be attacked by any number interested and the final results correlated and made available to the general engineering public.

Our aim, then, has been to endeavor to fulfill partially the hopes of so many engineers for a standardization, with the hopes that the result obtained might be checked independently, verified or corrected and made available to those desirous of the information.

Sufficient data were taken to provide overall heat transfer coefficient with a combination of three variables, viz., quantity of air across coil, temperature of air entering, and temperature of ammonia. A schedule sheet of the runs made is shown on page 25

Thermocouples were attached at various points over the tubes and fins to determine the characteristics of the temperature gradient.

B. Applications.

The scientific value and practical advantages are apparent:

1. Complete information would provide manufacturers and installation engineers with an accurate and precise method of rating and selecting standard coils.

2. With any one type as a basis, comparative data could easily be acquired.

3. Reliable results concerning temperature gradients would tend to insure more efficient and economical design.

4. These data would be of great value to the independent manufacturers who do not have elaborate research and testing departments.

5. In many cases there is occasion for special equipment, depending on the nature of the job. With sufficient data available, the coil could be designed quickly and accurately with a knowledge that the product would perform as specified.

C. Previous Work.

After a search of practically all recent engineering information available, it was found that considerable work has been done with fin type evaporator coils.

The most recent research similar to this was by G. L. Tuve and C. A. McKeeman.[9]

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9. <u>Performance of Fin-Tube Units for Air Cooling</u>

and Dehumidifying, Heating, Piping and Air Conditioning, June, 1937.

*****

The most comprehensive mathematical analysis has been done by William Goodman[8]. Mr. Goodman also carried out sufficient experiments to verify his mathematical conclusions.

Pownall[1] of York Corporation and W. J. King and W. L. Knaus[5] of General Electric Corporation have conducted experiments similar to those mentioned.

As was definitely shown at the delivery of Mr. Goodman's paper, there are still numbers of conflicting opinions as to coil performance.

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8. Dehumidification of Air with Coils, Refrigerating Engineering, October, 1936.

1. Rational Development and Rating of Extended Air Cooling Surface, October, 1935.

5. Heat Transfer Rates in Refrigerating and Air Cooling Apparatus, May, 1934.

Without exception, the above mentioned authorities are of the opinion that so far there have not been accurate fin surface temperatures determined by the use of theremocouples.

The investigation of previous experiments did not disclose work done on the type coil used under the different variables employed herein.

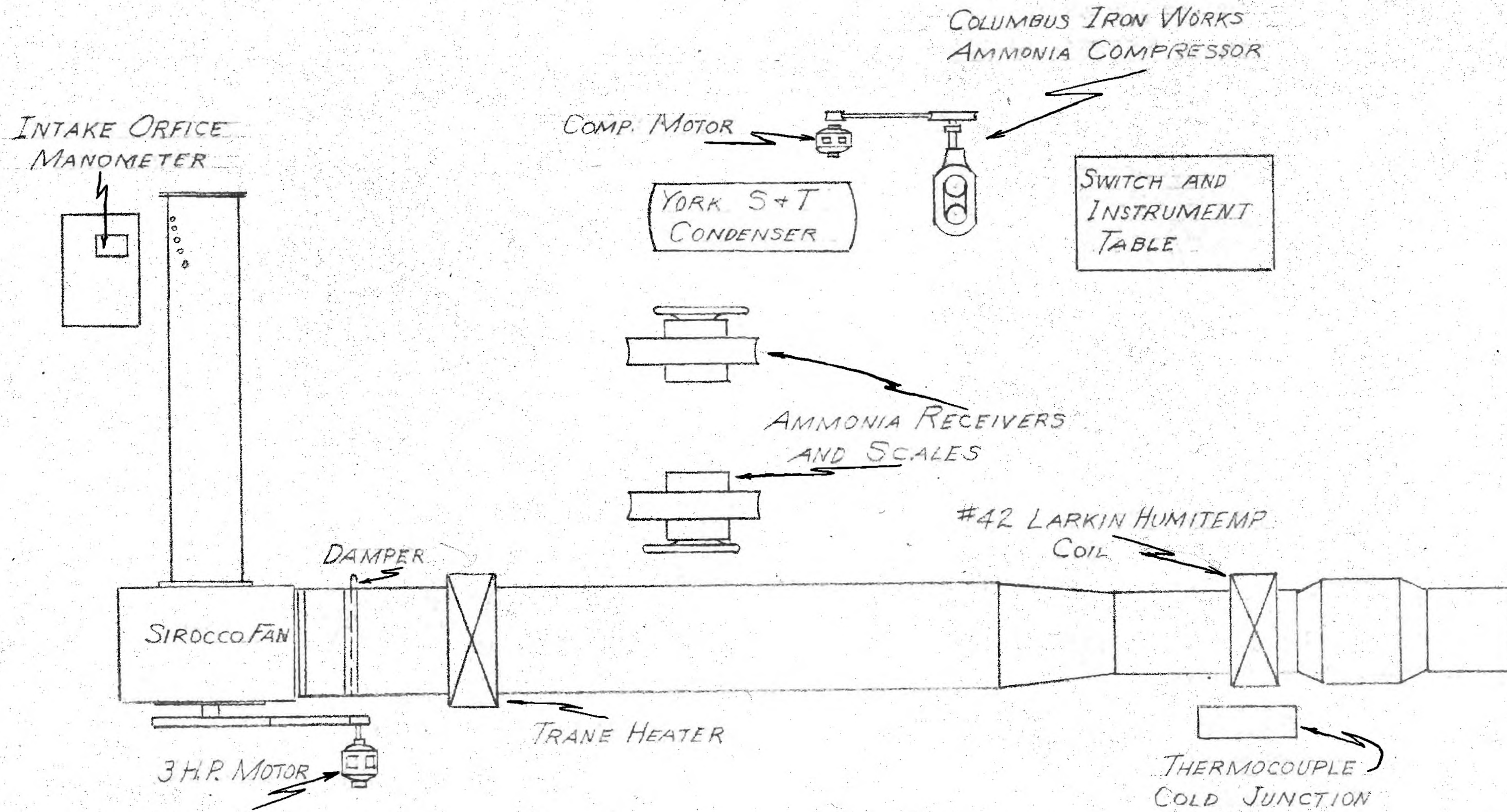
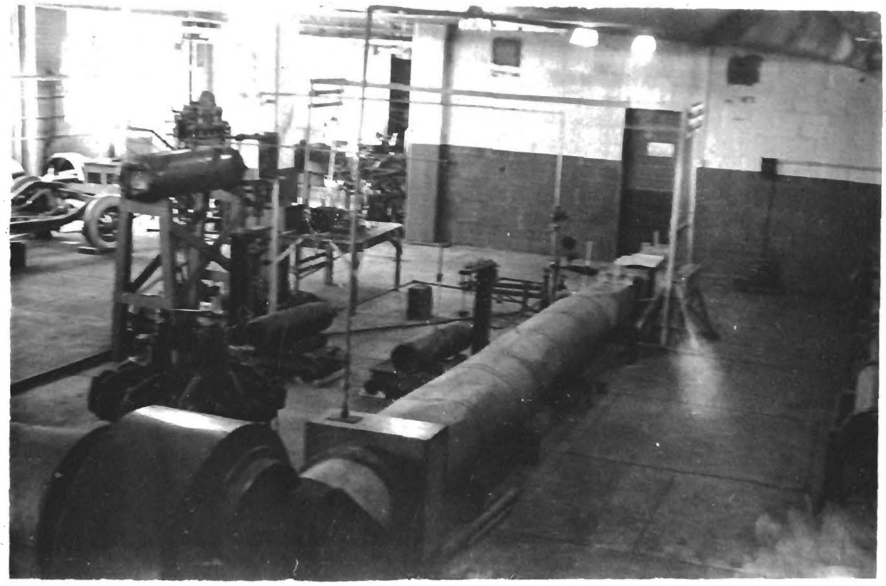
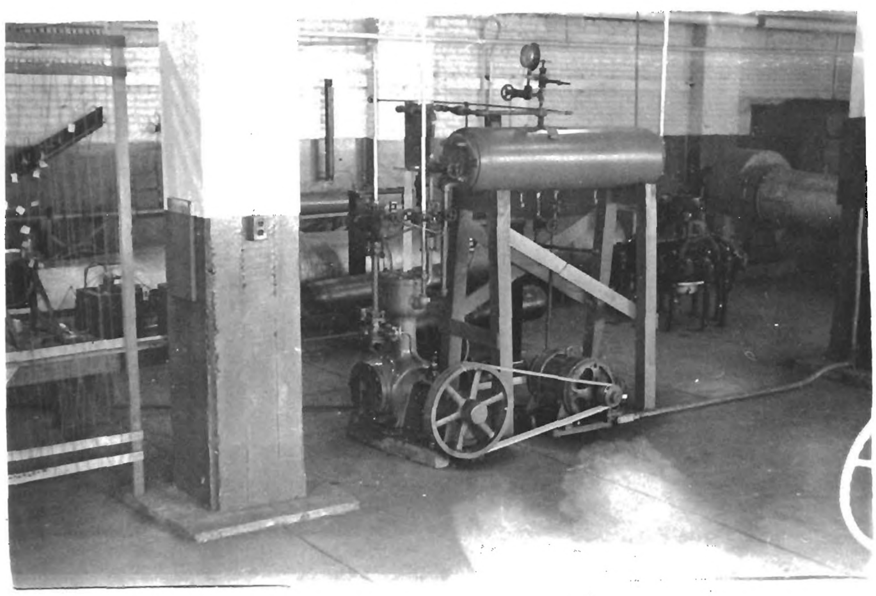


DIAGRAM OF APPARATUS



GENERAL VIEW OF EQUIPMENT



AMMONIA COMPRESSOR AND CONDENSER

II INSTRUMENTS AND EQUIPMENT

A. General Layout.

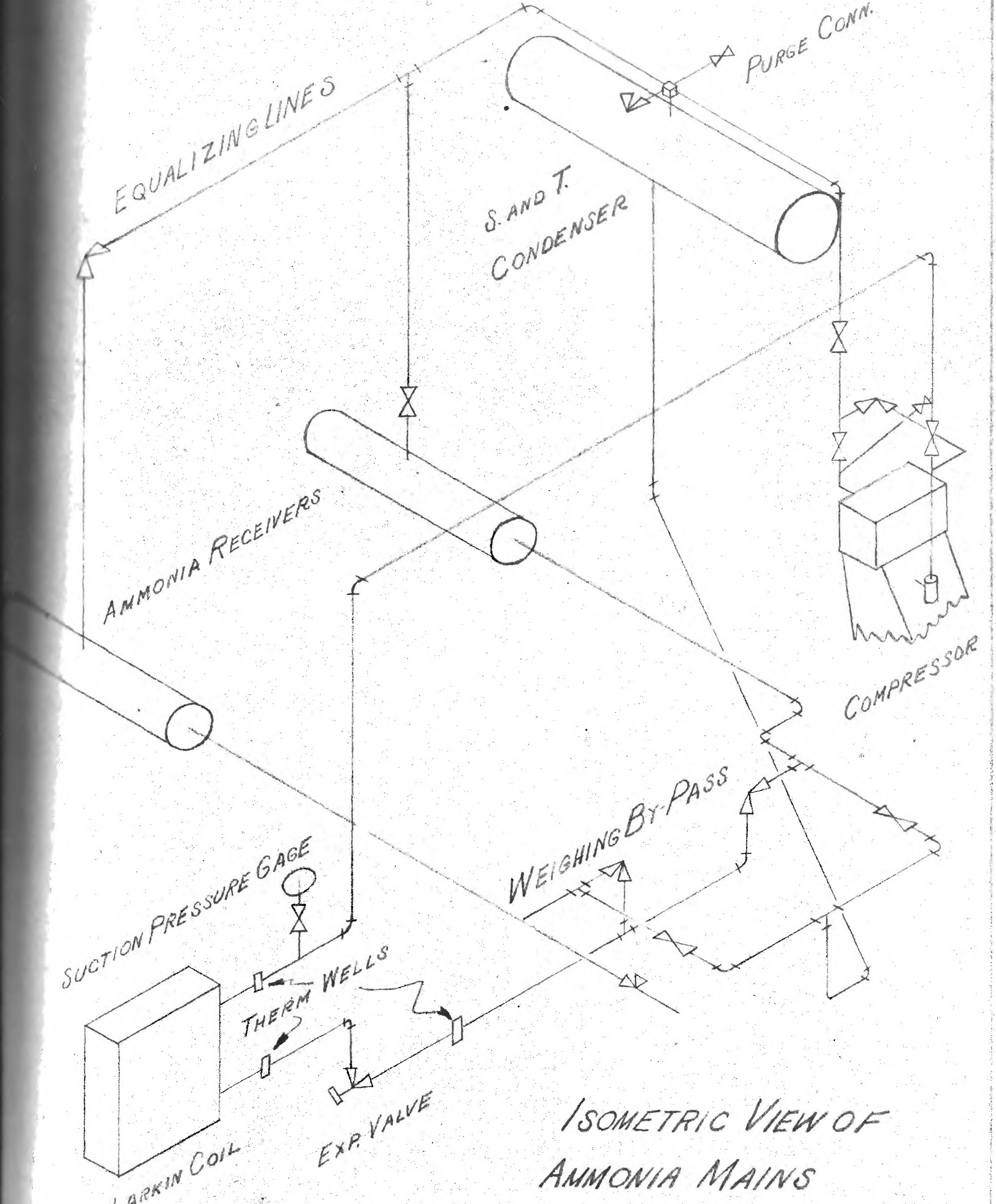
The plan on page 8 and photographs on page 9 show the arrangement and relative size of the equipment. It was set up in the basement of the Mechanical Engineering Building of the Georgia School of Technology in the spring of 1934. The metering element on the extreme end of the suction side of the blower was set up for an investigation of the intake pipe orifice. A Pitot tube traverse was taken for the orifices used and the quantity of air checked by a heat balance on air and ammonia.

The manometer for the orifice was located as near as possible to the point of pressure measurement, so that the connecting length would be a minimum.

An isometric view of the ammonia mains and weighing bypass is shown on page 11.

B. Fan, Motor and Ductwork.

The fan used was a Sirocco #4 multiblade blower built by American Blower Company of Detroit. The diameter of the impeller was twenty-four inches with sixty-four blades ten inches wide. The scroll casing was sixteen inches wide and the diameter of the



ISOMETRIC VIEW OF AMMONIA MAINS

intake was twenty-five inches. The discharge was twenty inches square. It was belt driven at 700 R.P.M. by a 220 volt, 60 cycle, 3 phase, 5 H.P. induction motor.

The duct work was of twenty-two gage galvanized iron pipe 22.75 inches inside diameter. All joints were soldered air-tight, and all rough projections on the inside of the joints were removed. Steel reinforcing hoops were placed around the pipe to insure a round cross section.

C. Control of Flow of Air.

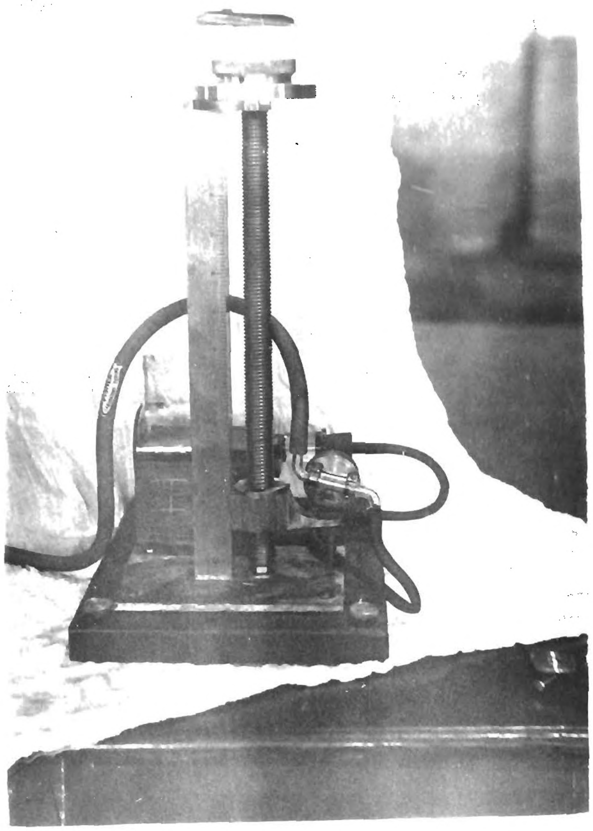
Since a constant speed inducation motor was used to drive the fan, other means had to be provided to vary the flow. This was accomplished by two methods:

1. By varying the size of the orifice on the suction side of the fan.

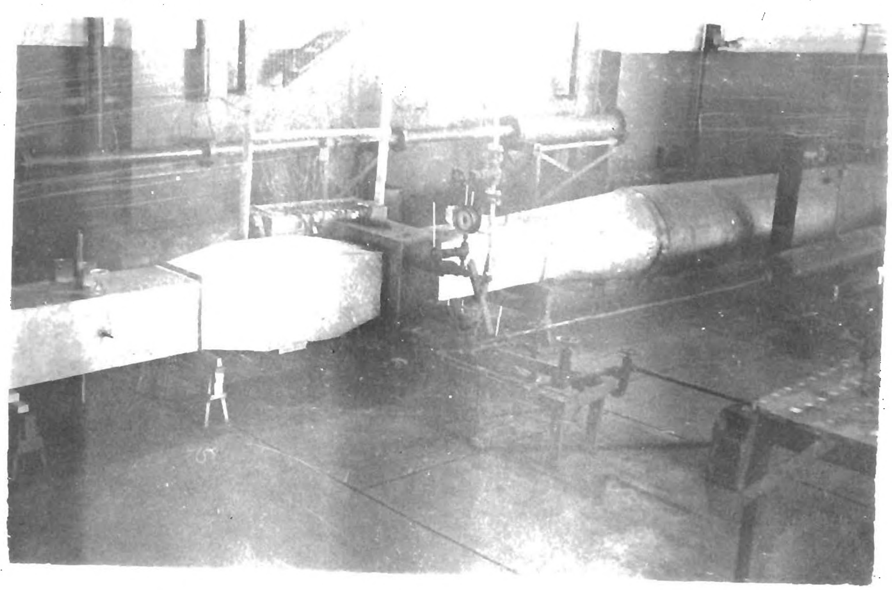
2. By a sliding shutter at the fan discharge.

D. Manometer.

Previously, during the investigation of the intake pipe orifice as a metering device for air, it was necessary to construct a micro-manometer to measure accurately slight pressure differentials. A photograph of this instrument may be found on page 13.



MICROMANOMETER



AMMONIA TEST COIL AND DIFFUSER

The design of the instrument was worked out at Georgia Tech, but it was based on a similar instrument used at the University of Toronto[11]. It is read directly to one thousandth of an inch of alcohol and may be estimated to .0005 inch of alcohol approximating .0004 inch of water. A full description of the manometer may be found in the appendix.

E. Pitot Tube.

The Pitot tube used to check the coefficient for the two orifices used was made especially for the series of tests on Investigation of the Intake Orifice. The tube was proportioned as recommended by the American Society of Mechanical Engineers.[10]

F. Intake Orifice Meter.

The orifice plates were of the thin-plate, square-edge type with circular openings concentrically located with respect to the pipe. They were turned on

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11. E. Owen - Measurement of Air Flow - Chapman & Hall, Ltd., London, 1933.

10. American Society of Mechanical Engineers Research Publications, Third Edition - 1931 - Fluid Meters, Their Theory and Application.

a lathe from galvanized sheet iron with an average thickness of .64 inches. Rough edges were removed by a fine file. Only two orifices were used:

1. 15.25 inch diameter or 45% of pipe area.
2. 10.2 inch diameter or 20% of pipe area.

A pressure tap located 40% of the pipe diameter downstream from the intake orifice was found to be the position of the vena contracta or maximum pressure differential. The tap was ground with valve compound, and the hose connection was of new, heavy tubing.

G. Heating Coil.

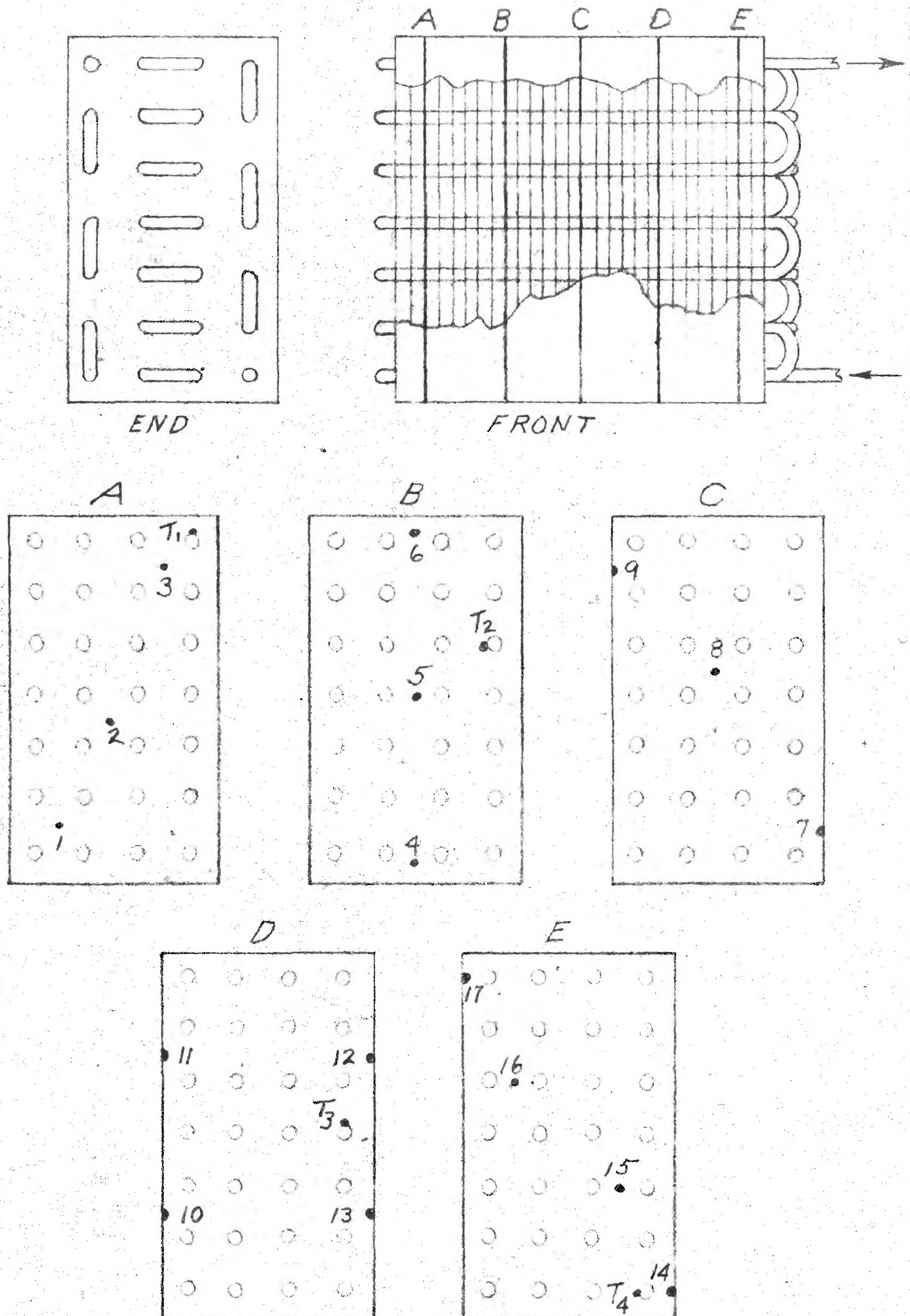
A steam heating coil manufactured by Trane Company was placed between the blower and the cooling coil in order to secure the desired dry-bulb temperature of the air before the cooling unit.

H. Cooling Coil.

The coil used was a special Larkin <u>Humi - Temp</u> of aluminum, cross fin type. Tubes were 5/8" in diameter and spaced on $1\frac{3}{4}$" center lines.

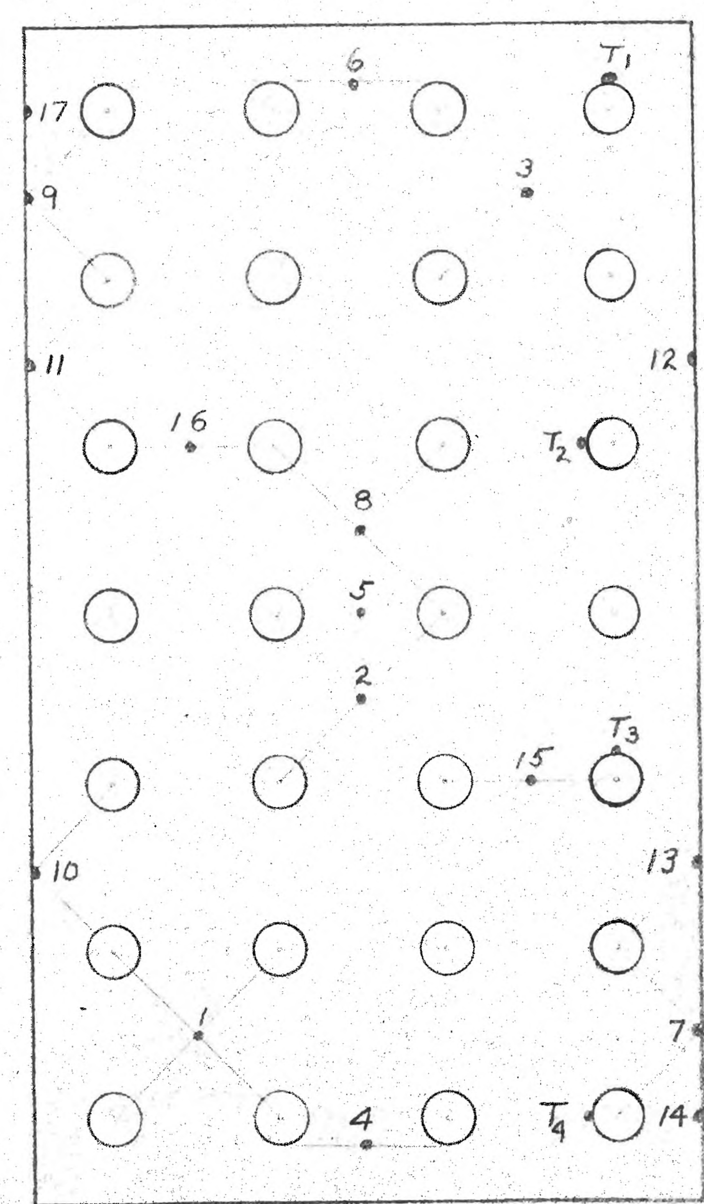
The unit was housed in an aluminum housing to give the maximum amount of refrigeration in the minimum amount of space, and with the minimum amount of weight.

THERMOCOUPLE POSITIONS



DOTS INDICATE THERMOCOUPLE POSITIONS

LOCATION AS IF ALL THERMOCOUPLES WERE ON ONE FIN



DOTS INDICATE THERMOCOUPLE POSITIONS

Specifications:

Width	Height	Depth	Cooling Surface
$12\frac{1}{4}$"	$12\frac{1}{4}$"	7"	33.25 sq. ft.

Fins

27 - 7" x $12\frac{1}{4}$" — Free Frontal Area

1 - $9\frac{1}{4}$" x $12\frac{1}{4}$" — 0.602 sq. ft.

Thickness - .028" — Fin Spacing

28 Holes .5" Diameter — 7/16"

Tubes:

4 rows of 7 each $12\frac{1}{4}$" long

0.627" outside diameter

0.525" inside diameter.

Ratio of Fin to Tube Area 7.09.

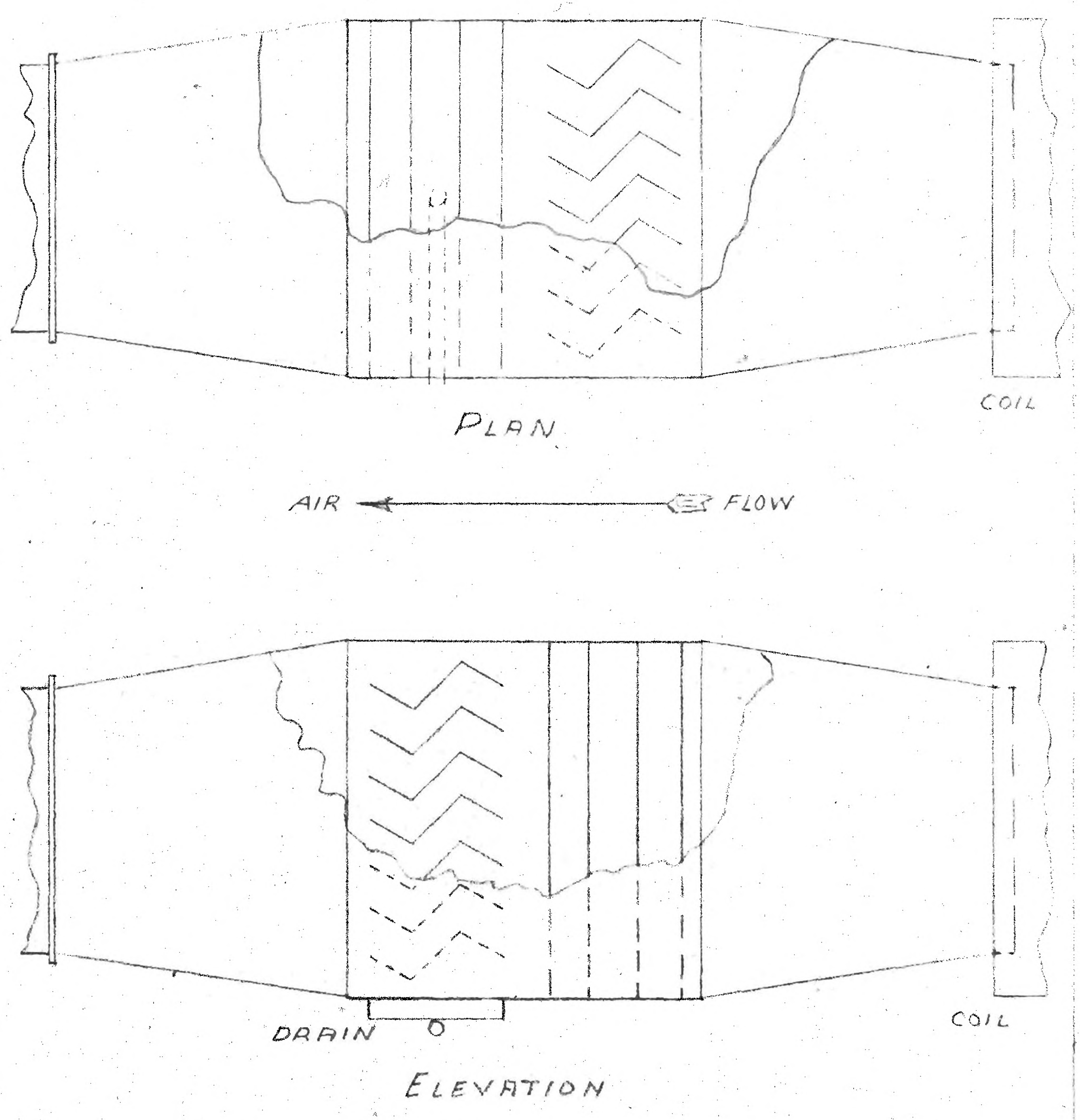
Refrigerant - Ammonia.

A sketch of the coil may be seen on page 16. A full description of the thermocouples and their positions may be found in the appendix.

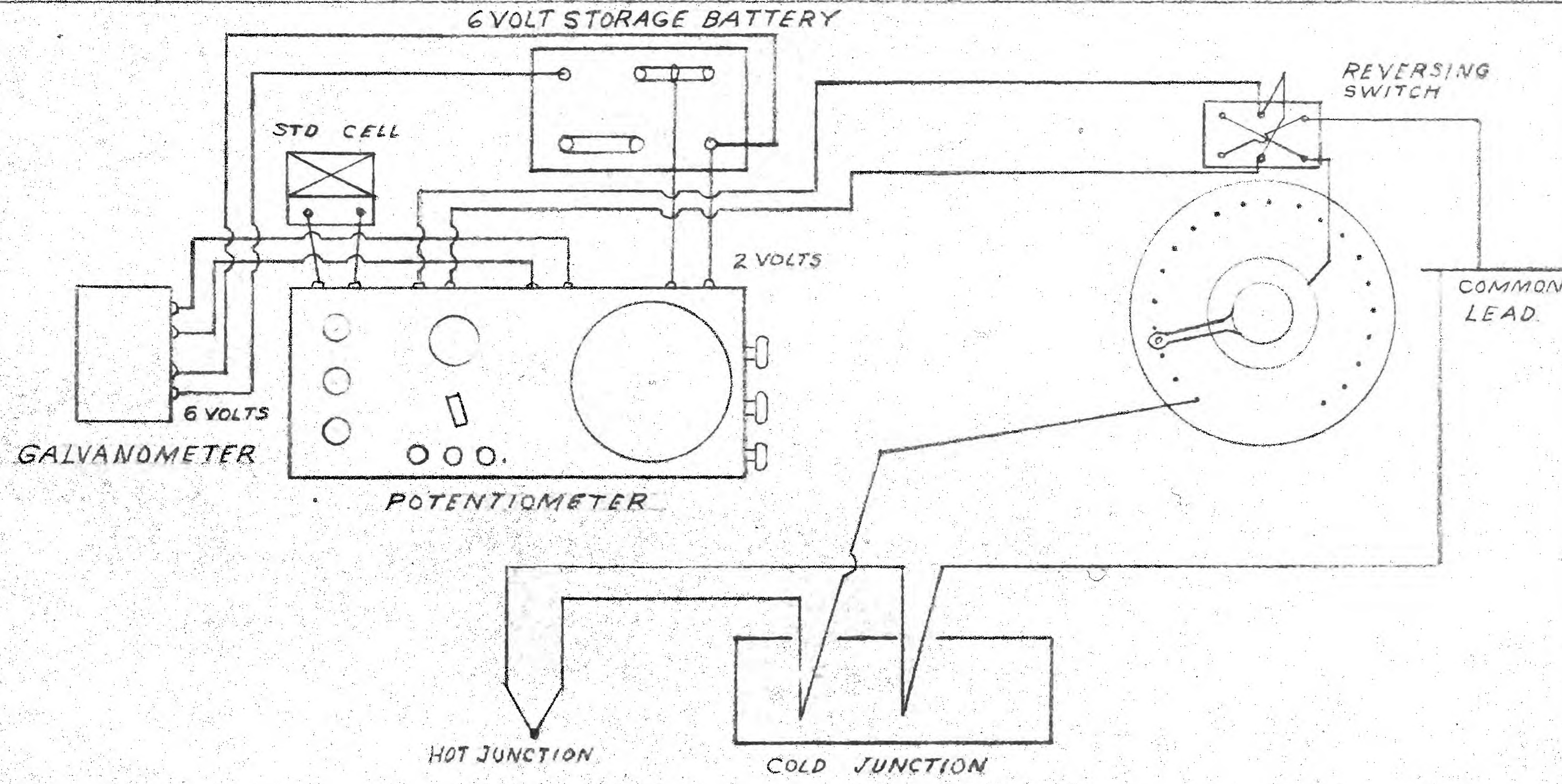
I. Eliminator and Diffuser.

It was found necessary to devise some means of completely mixing the exit air from the ammonia coil and also to eliminate all moisture particles in order that a representative temperature measurement might be made. A combined diffuser and eliminator was designed and built by Moncrief Furnace Company, Atlanta, Georgia. A sketch of this piece of apparatus

DIAGRAM OF
ELIMINATOR AND DIFFUSER

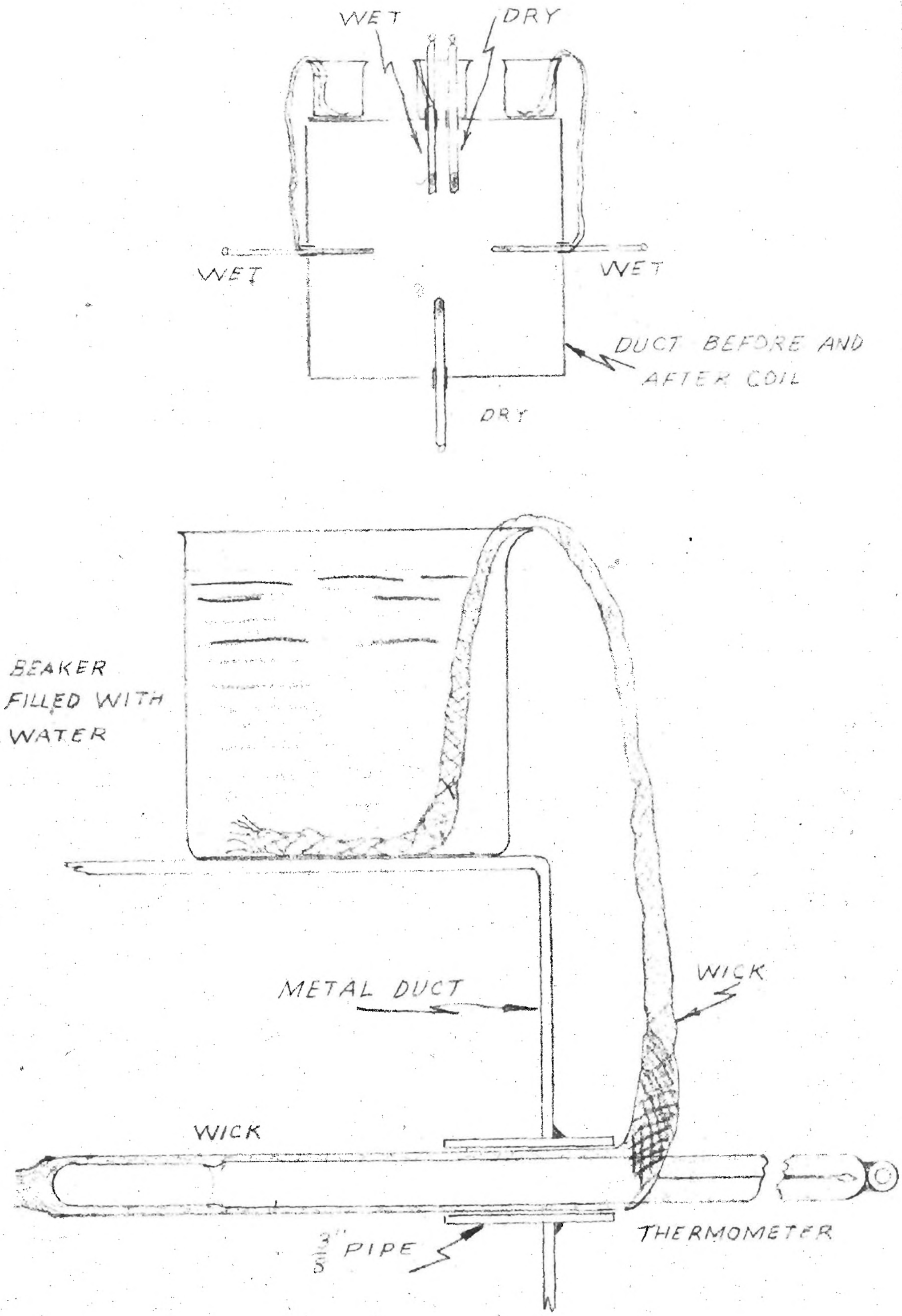


SCALE = $\frac{1}{8}$



THERMOCOUPLE WIRING DIAGRAM

WET BULB ARRANGEMENT



may be seen on page 19.

K. Thermocouples.

Copper, constantan thermocouples were used. The potentiometer was a Leeds & Northrup, K-2. Their arrangement and hook-up may be seen from figures on page 23.

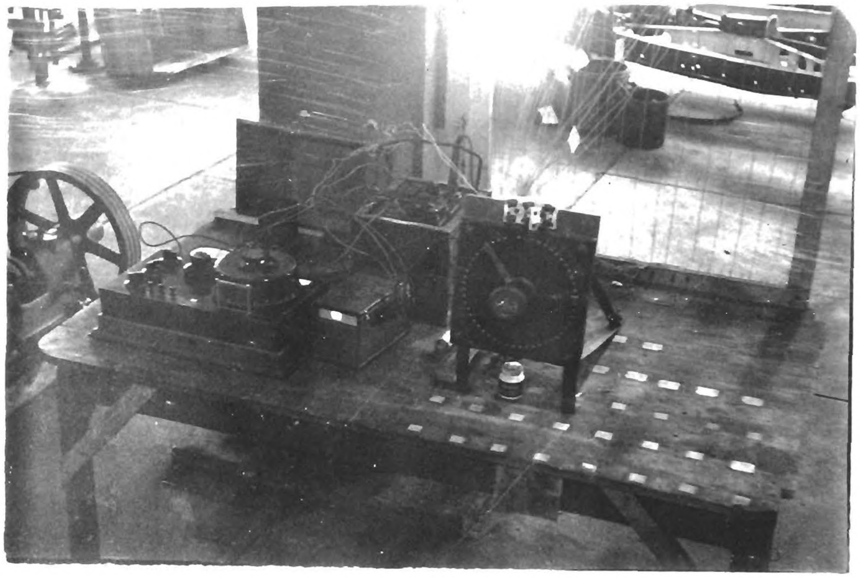
A complete discussion of apparatus and method of attaching the thermocouples may be found in the appendix.

L. Thermometers.

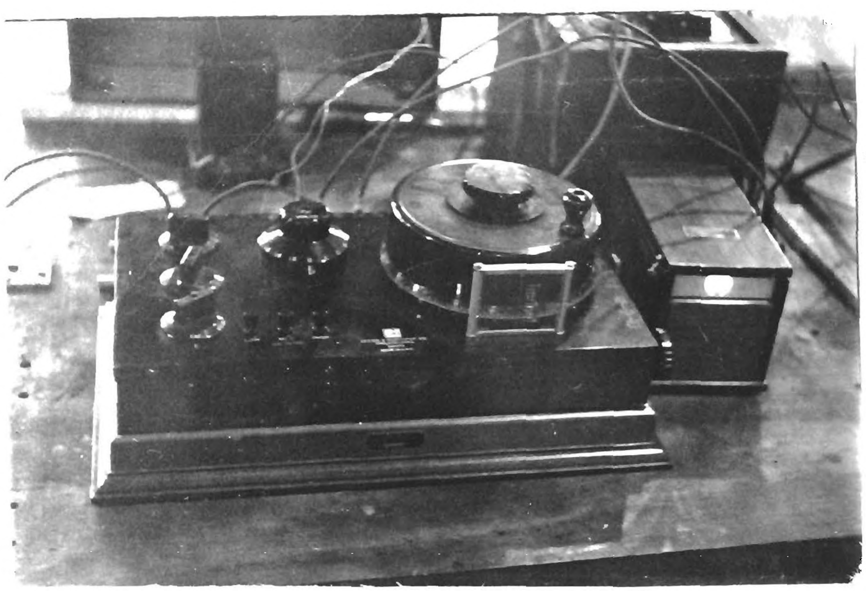
Calibrated thermometers capable of being read to one-half degree were used to measure the temperatures of the air and ammonia. Difficulty was encountered in making accurately calibrated thermometers give consistent wet bulb readings. It was necessary to shield them from any source of radiation, as well as to devise a suitable moisture supply through wicking.

One wet and dry bulb thermometer was placed in the air entering the duct. Three wet bulb and two dry bulb thermometers were placed before and after the cooling coil.

A sketch showing the wet bulb arrangment may be seen on page 21.



POTENTIOMETER ARRANGEMENT



POTENTIOMETER AND GALVANOMETER

M. Compressor, Condenser, Motor, Ammonia By-Pass.

The ammonia compressor used was manufactured by the Columbus Iron Works, Columbus, Georgia.

It is a two-cylinder vertical enclosed machine $3\frac{1}{2}$" bore, $3\frac{1}{2}$" stroke, maximum speed recommended 375 R.P.M. Capacity at 357 R.P.M. with 20# suction 185# discharge slightly in excess of three tons.

The compressor was driven through a V-Belt drive by a 10 H.P., 220 volt, 60 cycle, 3 phase, 1160 R.P.M., induction motor manufactured by Westinghouse.

The condenser used was a shell and tube type manufactured by York Ice Machinery Corporation.

The ammonia weighing by-pass was assembled at Georgia Tech. An isometric view showing the detail may be seen on page 11. This by-pass consisted of a system of pipes and valves arranged in such a manner that the two ammonia drums could be used alternately as source and receiver. Having these drums placed on scales provided an accurate method of weighing and checking the amount of ammonia circulated.

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TABLE OF RUNS MADE

AIR TEMPERATURE IN °F AMMONIA SUCTION PRESSURE IN #/□" GAGE

	SERIES 1			SERIES 2			SERIES 3		
	FLOW	AIR TEMP.	SUCTION PRESS.	FLOW	AIR TEMP.	SUCTION PRESS.	FLOW	AIR TEMP.	SUCTION PRESS.
A	1	80	25	1	90	25	1	100	25
	2	80	25	2	90	25	2	100	25
	3	80	25	3	90	25	3	100	25
B	1	80	35	1	90	35	1	100	35
	2	80	35	2	90	35	2	100	35
	3	80	35	3	90	35	3	100	35
C	1	80	45	1	90	45	1	100	45
	2	80	45	2	90	45	2	100	45
	3	80	45	3	90	45	3	100	45
D	1	80	60	1	90	60	1	100	60
	2	80	60	2	90	60	2	100	60
	3	80	60	3	90	60	3	100	60

FLOW LEGEND

1 .21 IN. ALCOHOL-15.25 IN. ORIFICE
2 .10 IN. ALCOHOL-15.25 IN. ORIFICE
3 .20 IN. ALCOHOL-10.20 IN. ORIFICE

1 INCH ALCOHOL = 0.80 INCHES WATER

METHOD OF CONDUCTING TEST

With a determined quantity of air, the proper orifice was bolted to the intake end of the duct. The manometer was checked to remove all air bubbles, the base of the instrument leveled, and the zero reading accurately set. After the fan was started, the slide valve at the discharge of the blower was set to give the desired flow.

The steam tempering coil was regulated to produce the desired dry bulb temperature before the coil.

The thermocouple cold junction was packed with a water ice mixture and the potentiometer connected up. The instruments were set to zero against a standard cell and a set of check readings taken to insure stable operation. In so doing, any thermocouple found to give pulsating readings was checked and corrected, usually at the cold junction.

The compressor was operated until the desired pressure and temperature of the evaporator coil was reached. It was necessary to use two expansion valves, one before the coil and one after it, in order to reach the desired capacity with any degree of rapidity. Since the system was of small capacity and "pump-downs" frequent, it was found advisable to purge regularly. A few degrees of superheat were carried on the suction side of the coil in order that a heat balance might be made between the ammonia and the air. After all conditions became constant, the valves of the weighing by-pass were changed in order that the full drum might be used as the source of ammonia. Readings were taken for a period of one-half hour. The time required for an overall set of readings averaged roughly four minutes. Time required between runsvaried, due to a number of reasons:

1. Atmospheric conditions naturally varied from day to day.

2. The necessity of keeping constant a number of conditions without sensitive, automatic controls required no set length of time.

In view of the fact that the outside air conditions changed from day to day, trial runs were made frequently and a heat balance calculated.

Repeat runs were made on at least one-fourth of the work. It was not practical to try to average the two runs, since the wet and dry bulb readings were necessarily different.

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RESULTS

The tabulated results are shown on pages 34 and 35.

On comparing the saturation temperature after the expansion value with the temperature and pressure, immediately after the coil when carrying no superheat, an average pressure drop of one (1) pound was considered a reliable value.

The coil temperature was taken from ammonia tables[13], asthe saturation temperature corresponding to the average pressure throughout the coil.

Calculations for the quantity of air flowing, using pressure drop across an intake orifice[14], are as shown.

Run 1 Series 1A

Entering Air	$T_d = 76^\circ$	Ammonia Data	
	$T_w = 50^\circ$	Temp. before exp. valve	$= 65^\circ$
Air Before Coil	$T_d = 79^\circ$	Temp. after exp. valve	$= 11^\circ$

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13. Circular of the Bureau of Standards #142, Tables of Thermodynamics Properties of Ammonia.

14. Transactions of American Society of Mechanical Engineering, Dec., 1934, Vol. 56, No. 12, The Intake Orifice and a Proposed Method For Testing Exhaust Fans, by N. C. Ebaugh and R. Whitfield.

$T_w = 51.7^\circ F$ Temp. out of Coil = $14^\circ F$

Air After Coil $T_d = 68^\circ F$ Pres. out of Coil = 25#/□" gage

$T_w = 46.4^\circ F$ $\#NH_3$/30 Min. = 16.5#

Barometer = 29.2 "Hg.

Orifice = 15.25" Manometer Reading = .21 "Alcohol

<u>Air-By Orifice</u>

$$\text{"air/min.} = 835\sqrt{hd}$$

$$h = .21 \times .8 = .168\text{" } H_2O$$

From Psychrometric chart the volume at T_d = 76 and T_w = 50 is 13.54 at 14.7#/□" pressure.

Correcting for Pressure

$$P_2V_2 = P_1\,V_1$$

$$P_2 = 29.2\text{"Hg} = 14.35\#/\square\text{"}$$

$$P_1 = 14.7\#/\square\text{"}$$

$$V_1 = 13.54 \text{ cu. ft.}$$

$$V_2 = \frac{14.7}{14.35} \times 13.54 = 13.88$$

$$\frac{1}{V_2} = d = \frac{1}{13.88} = .072$$

$$\#\text{air/min} = 835\sqrt{.168 \times .0721}$$

$$= 92\ \#/\text{min.}$$

A simplification of the formula used is shown in the appendix. Ammonia weights were recorded in order that a heat-balance check could be made on the quantity of air flowing. The difference in pounds of air by heat balance as compared to pounds of air by intake orifices drop was limited to five per cent. The calculation for this check is shown in the appendix.

As shown on the result sheet, a constant quantity of air was taken for a given orifice with a constant pressure drop. The maximum deviation due to a density change of entrance air was found to be 2.28%, as shown in the appendix.

Differences in total heat per pound of dry air were taken directly from a General Electric Psychrometric Chart. A number of readings were checked against a general psychrometric[15] chart, which took into account pressures under 14.7 pounds per square inch. The differences found were insignificant.

The log mean temperature differences and the overall heat transfer coefficients were calculated as follows:

$$H = KA\Delta T$$

H = Heat transfer in BTU's per hour.

K = Coefficient of heat transfer BTU's per sq. ft. per $^{\circ}$F per hr.

A = Area of transfer surface.

ΔT = Log mean temperature difference between

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15. Psychrometric chart with Barometric Pressure as a Variable. J. S. Chandler, Heating and Ventilating, Vol. 33, March, 1936, P. 36.

air entering and ammonia.

$$K = \frac{H}{A \Delta T}$$

Air temperature entering = $79^{o}F$

Air temperature leaving = $68^{o}F$

considering an average pressure drop of one pound through the coil.

Average pressure in coil = 39.35 + .5 = 39.85#/□" abs.

Saturation temperature at 39.85 #/□" abs. = $11.5^{o}F$

$$LMTD = \frac{T_1 - T_2}{\log_e \frac{T_s - T_1}{T_s - T_2}}$$

T_1 = Temp air in

T_2 = Temp air out

T_s = Sat. Temp of NH_3

$$LMTD = \frac{79 - 68}{\log_e \frac{11.5-79}{11.5-68}} = \frac{11}{\log_e \frac{67.5}{56.5}} = \frac{11}{\log_e 1.195}$$

$$LMTD = \frac{11}{.178}$$

$$LMTD = 61.8^{o}F$$

H = #air/hr x ΔH

= 92 x 60 x 3 = 16,560 BTU's/hr

$$K = \frac{H}{A \Delta T}$$

$$K = \frac{16,560}{33.25 \times 61.8}$$

K = 8.06 BTU's per sq. ft. per ^{o}F per hr.

The air velocity was calculated using the temp. before the coil and the free area of a cross section of the coil.

Calculation for free area of coil is shown in the appendix.

$$Q = AV$$

Q = Quantity of air - cu. ft. per. min.

A = Free Frontal Area

V = Velocity of Air - ft. per. min.

Area = .602 sq. ft.

Q = # air per. min. x specific value

Volume of air before coil $= 13.61 \times \frac{14.7}{14.35}$

Vol. = 13.96

$Q = 13.96 \times 92 = 1283$ cu. ft. per. min.

Velocity Entering Coil

$$V = \frac{Q}{A}$$

$$V = \frac{1283}{.602}$$

Velocity = 2130 ft. per. min.

On the result sheet is shown an overall coefficient based on an arithmetic temperature difference between the entering dry bulb and coil. Investigation showed that such a calculation was not used as a basis in engineering work. This value was used, however, as a rough check on the coefficient found by the use of the log mean temperature difference.

Although the entering dry bulb temperature was kept constant, the relative humidity changed with atmospheric conditions. With an increase in relative humidity, a decrease in sensible cooling was recorded for the same series. This fact affected the L.M.T.D., which in turn gave irregularity to the curves using velocity and the overall coefficient. Curves most representative of the performance are those using BTU per sq. ft. against the free velocity.

Curves were also plotted using coil temperature as the abscissa and "K" as the ordinate. Again the same irregularity was noticed.

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SAMPLE DATA SHEET

RUN #1 SERIES 1A

PRESSURES			TEMPERATURES															NH_3 WT.			
D	AC	MM	AIR IN		BEFORE COIL					AFTER COIL					NH_3			D1	D2	BAR	
			DRY	WET	DRY	DRY	WET	WET	WET	DRY	DRY	WET	WET	WET	B.E.	A.E.	O.C.				
#/□"	#/□"	"AL	°F	°F	°F	°F	°F	°F	°F	°F	°F	°F	°F	°F	°F	°F	°F	#	#	"HG	
105	25	.21	76	50	78	78.2	51	51.8	51.8	67	67.5	46	46.2	46	65	11	14.5	137.5	184	29.26	
105	25	.21	76	50	79	79.5	51.9	51.5	52	68	68.2	46.5	47	46.8	65	11.2	14			29.26	
103	25	.21	76	50	77.9	78.6	51	51.2	51.8	66.9	67	46	46.2	46	65	11	13.2			29.26	
103	24.5	.21	76	50	81	80	52	52.2	53	68.5	69	47	47	47	65	11	14.5	154.0	168	29.26	
104	25	.21	76	50	— 79 —		51.7			68		— 46.4 —			65	11	14.05	16.5	16	29.26	AVERAGE

LEGEND

D	DISCHARGE
AC	AFTER COIL
M.M.	MICROMANOMETER
BE	BEFORE EXPANSION VALVE
AE	AFTER EXPANSION VALVE
OC	OUT OF COIL
D1	DRUM ONE
D2	DRUM TWO

Tabulated Data and Results

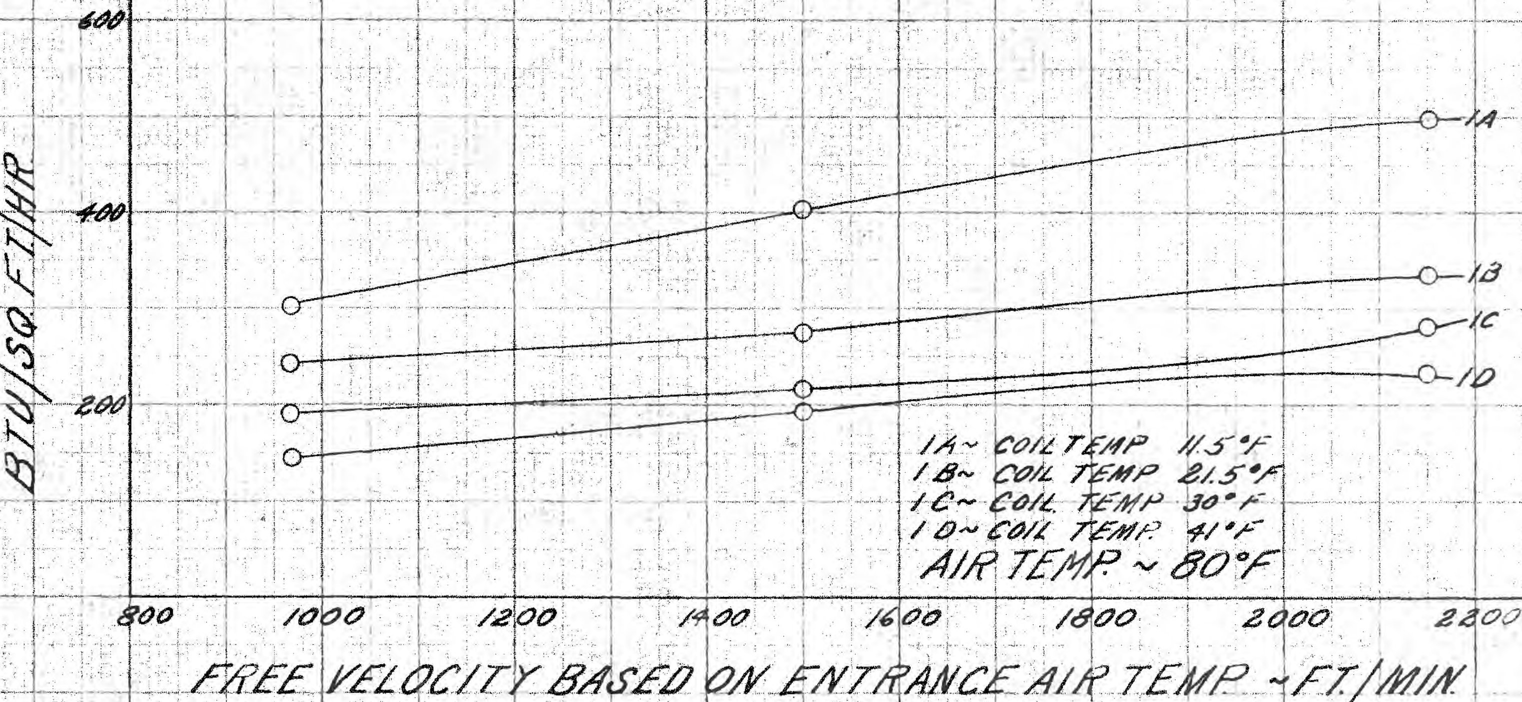
Series		Units	1A			1B			1C			1D		
Run			1	2	3	1	2	3	1	2	3	1	2	3
Length of Run		Min.	30	30.	30	30	30	30	30	30	30	30	30	30
Barometer		In. Hg.	29.2	29.12	29.1	29.16	29.16	29.1	29.0	29.0	29.0	29.0	29.0	29.0
Air Temp. In-Wet		°F	50	56	64.8	52.5	51.5	64.8	59.5	59.5	64.5	60	60	65
Air Temp. In-Dry		°F	76	77	76	79	77	76	78	78	76	78	78	76.7
Air Temp. Before Coil-Wet		°F	51.7	57	65.15	53.3	52.8	66	60.2	60.2	65.6	61	60.5	65.9
Air Temp. Before Coil-Dry		°F	79	79.8	79.1	81	80	79.5	80.2	80.2	79.6	80.4	79.7	79.9
Air Temp. After Coil-Wet		°F	46.4	51.2	60	50	48.5	61.5	57.75	57.3	62	58.8	57.9	63.2
Air Temp. After Coil-Dry		°F	68	66	68.4	74	71	70.4	73.8	73	71.1	74.7	73.4	72.6
Size Orifice		Inches	15.25	15.25	10.2	15.25	15.25	10.2	15.25	15.25	10.2	15.25	15.25	10.2
Orifice Manometer Reading		In. Al.*	.21	.10	.20	.21	.10	.20	.21	.10	.20	.21	.10	.20
Ammonia	Temp. Before Exp. Valve	°F	65	77	79.5	77	77	78.1	76	75.3	80	69.4	68	78
	Temp. After Exp. Valve	°F	11	11.5	11.5	21	21	21.5	30.6	30.7	31	40.5	41.3	41.6
	Temp. Out of Coil	°F	14	13	15.6	33.5	29.7	23.3	34.6	34	32.4	43.4	43.2	42.9
	Press. Out of Coil Abs.	#/□"	39.35	39.31	39.30	48.83	49.33	49.3	59.25	59.25	59.25	73.95	74.55	74.25
	Discharge Press. Abs.	#/□"	118.35	194.31	174.3	172.33	174.33	169.3	144.25	144.25	169.25	151.25	148.25	168.25
	Weights per 30 min	Lbs	16.5	13.5	10.5	11.0	9.5	8.5	9.0	7.5	6.5	8.0	6.5	5.0
	Average Pressure Abs.	#/□"	39.85	39.85	39.8	49.33	49.83	49.8	59.75	59.75	59.75	74.45	75.05	74.75
	Saturation Temp.	°F	11.5	11.5	11.44	21.05	21.5	21.48	30.0	30.0	30.0	40.76	41.16	40.96
Pounds of Air per Min.		#/min	92	63.5	40.8	92	63.5	40.8	92	63.5	40.8	92	63.5	40.8
ΔH per Pound of Air		BTU's/#	3.0	3.5	4.1	2.0	2.4	3.3	1.7	1.9	2.6	1.4	1.7	2.0
B.T.U.'s per Hour		BTU's/hr	16,560	13,340	10,040	11,040	9,144	8,078	9,384	7,239	6,365	7,728	6,477	4,896
L.M.T.D.** Air-NH_3		°F	61.8	61.3	61.7	56.1	53.8	52.8	46.3	46.1	45.4	36.5	35.2	34.9
Coef. of Heat Transfer		BTU/°F □' hr	8.06	6.54	4.89	5.92	5.11	4.60	6.10	4.72	4.22	6.37	5.53	4.22
Air Velocity Entering Coil		Ft/min	2130	1495	965	2155	1485	965	2170	1508	967	2170	1502	967
ΔT of Air In and NH_3		°F	67.5	68.3	67.7	59.9	58.5	58.0	50.2	50.2	49.6	39.6	38.5	39.0
Coef. Based on ΔT of Air In and NH_3		BTU/°F □' hr	7.38	5.87	4.46	5.54	4.70	4.19	5.62	4.34	3.86	5.87	5.06	3.78

Series		Units	2A			2B			2C			2D		
Run			1	2	3	1	2	3	1	2	3	1	2	3
Length of Run		Min.	30	30	30	30	30	30	30	30	30	30	30	30
Barometer		In. Hg.	29.3	29.3	29.0	29.2	29.2	29.0	29.12	29.12	29.0	29.12	29.12	29.12
Air Temp. In-Wet		°F	52	52	69.5	53	52.5	70	57.5	56	69	57	57	60
Air Temp. In-Dry		°F	74	74	80.5	76	76	82	79	77	83	78	78	84
Air Temp. Before Coil-Wet		°F	58.3	58.3	72.1	58.5	58	72.5	61	61	71	61	61.2	61.8
Air Temp. Before Coil-Dry		°F	90.2	90.2	90	90.2	90.3	90.8	89	90.2	90	89	90	89
Air Temp. After Coil-Wet		°F	52.4	52	65.35	53.9	53.1	67.9	57.8	57	67.3	58.1	57.8	58
Air Temp. After Coil-Dry		°F	75.2	75.6	79.9	80.5	79.5	79.3	80	80.2	79.5	81	81.1	79.1
Size Orifice		Inches	15.25	15.25	10.2	15.25	15.25	10.2	15.25	15.25	10.2	15.25	15.25	10.2
Orifice Manometer Reading		In. Al.*	.21	.10	.20	.21	.10	.20	.21	.10	.20	.21	.10	.20
Ammonia	Temp. Before Exp. Valve	°F	76.9	78	84	72.3	73	83	75	75	80	76.5	77	72
	Temp. After Exp. Valve	°F	11	12	12.5	22	21.3	22.3	32	31	31.1	42	41.5	42
	Temp. Out of Coil	°F	16	14.1	16.8	27.1	27.1	24	34	33	33.4	45	42	42
	Press. Out of Coil Abs.	#/□"	39.1	39.4	39.35	49.85	49.35	49.75	59.31	59.31	59.35	74.31	74.31	74.31
	Discharge Press. Abs.	#/□"	162.4	155.4	170.25	137.35	136.35	164.25	154.31	189.31	163.25	154.31	154.31	159.31
	Weights per 30 min	Lbs	20.5	15	14	16.5	11.5	10	12	10.5	8.0	10.5	8.5	6.5
	Average Pressure Abs.	#/□"	39.6	39.9	39.85	50.35	49.85	50.25	59.81	59.81	59.85	74.81	74.81	74.81
	Saturation Temp.	°F	11.21	11.55	11.49	21.99	21.53	21.9	30.06	30.06	30.09	41.0	41.0	41.0
Pounds of Air per Min.		#/min	92	63.5	40.8	92	63.5	40.2	92	63.5	40.2	92	63.5	40.8
ΔH per Pound of Air		BTU's/#	3.7	4.0	5.4	2.9	3.1	3.9	2.1	2.6	3.1	1.9	2.2	2.7
B.T.U.'s per Hour		BTU's/hr	20,420	15,240	13,220	16,010	11,810	9,407	11,590	9,906	7477	10,490	8,382	6,610
L.M.T.D.** Air-NH_3		°F	71.3	70.7	72.7	63.2	62.7	63.0	54.4	54.8	54.9	43.9	43.2	42.2
Coef. of Heat Transfer		BTU/°F □' hr	8.61	6.48	5.47	7.62	5.66	4.49	6.41	5.44	4.09	7.19	5.84	4.71
Air Velocity Entering Coil		Ft/min	2174	1510	990	2180	1515	990	2191	1525	988	2191	1525	975
ΔT of Air In and NH_3		°F	79.0	78.6	78.5	68.2	68.8	68.9	58.9	60.1	59.9	48.0	49.0	48.0
Coef. Based on ΔT of Air In and NH_3		BTU/°F □' hr	7.77	5.84	5.06	7.06	5.16	4.11	5.92	4.96	3.75	6.57	5.14	4.14

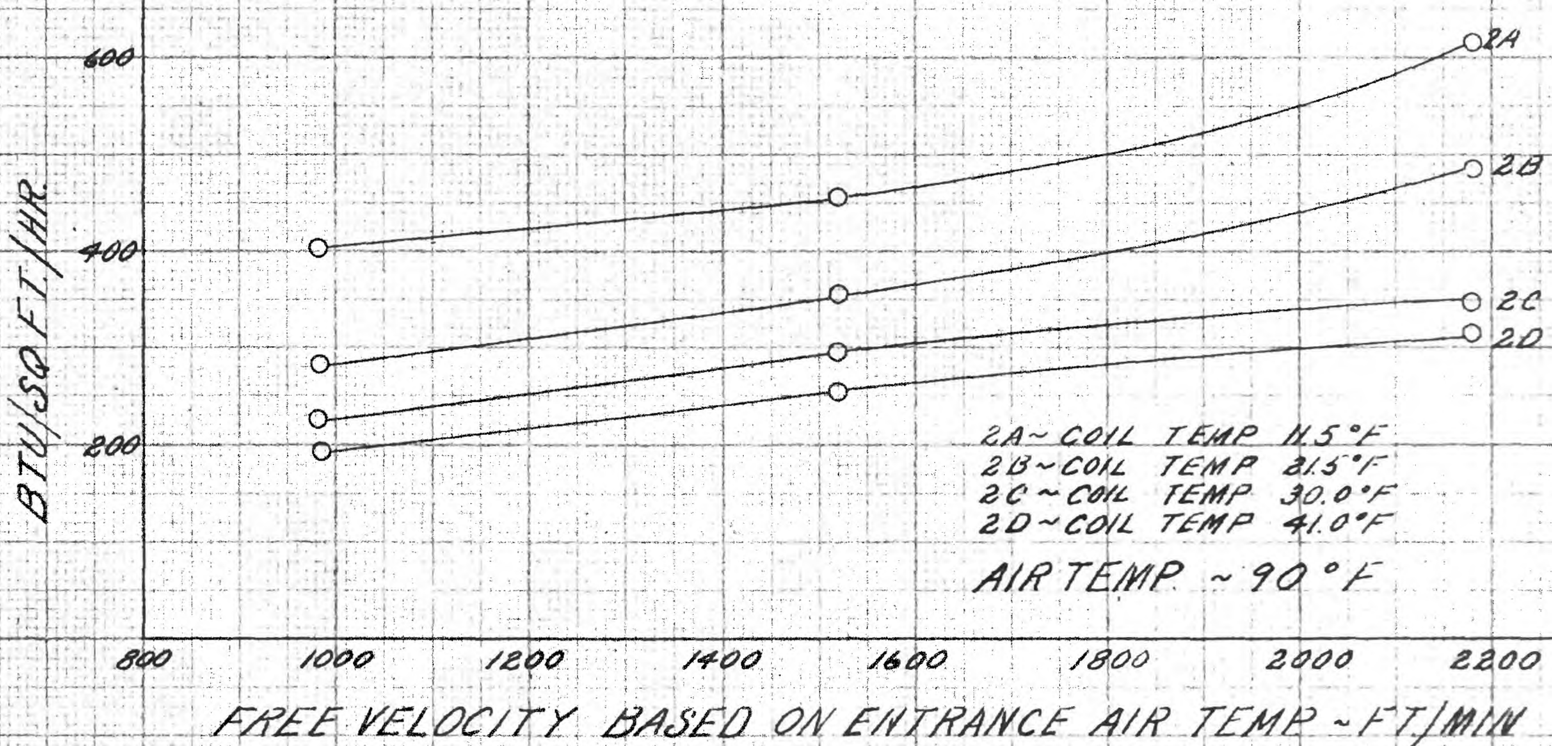
Series		Units	3A			3B			3C			3D		
Run			1	2	3	1	2	3	1	2	3	1	2	3
Length of Run		Min.	30	30	30	30	30	30	30	30	30	30	30	30
Barometer		In. Hg.	29.0	29.0	29.1	29.14	29.14	29.12	29.12	29.1	29.0	29.12	29.0	29.0
Air Temp. In-Wet		°F	57.5	56	70.5	56	56	60	58	55	67.3	59	57	66.5
Air Temp. In-Dry		°F	78	75	90	77	77	84	80	78	85	81	83	84
Air Temp. Before Coil-Wet		°F	66	65.3	73.25	64.8	63.9	65.8	65.3	63	71.8	65.5	63.2	71.5
Air Temp. Before Coil-Dry		°F	102.1	101	99.8	100.9	101	100.5	100.5	100	99.7	99.8	100.6	99.8
Air Temp. After Coil-Wet		°F	58.95	57.5	65.6	59.25	57.9	59	61.5	58.3	67.2	62.6	59.85	68
Air Temp. After Coil-Dry		°F	83.1	80.9	81.5	86.4	84.7	82	89.5	86.9	86.6	91	91.7	88.3
Size Orifice		Inches	15.25	15.25	10.2	15.25	15.25	10.2	15.25	15.25	10.2	15.25	15.25	10.2
Orifice Manometer Reading		In. Al.*	.21	.10	.20	.21	.10	.20	.21	.10	.20	.21	.10	.20
Ammonia	Temp. Before Exp. Valve	°F	85	79.4	87.2	72.1	79.5	73	75	80	73.5	75	79.8	82.6
	Temp. After Exp. Valve	°F	12.7	12.3	12.8	22.1	25	22	31	30.8	30.8	42	41.9	41.5
	Temp. Out of Coil	°F	37.1	12.6	17.3	23	23	23	35.8	32.9	32.1	44	45.6	43.8
	Press. Out of Coil Abs.	#/□"	38.95	39.25	39.3	49.32	49.32	49.31	59.31	59.3	59.35	74.31	74.75	74.25
	Discharge Press. Abs.	#/□"	154.25	204.25	191.3	166.32	166.32	159.31	154.31	162.3	169.25	154.31	157.25	168.25
	Weights per 30 min	Lbs	27.5	22	16.5	21	16.5	12	15	12	9.5	12	9.5	7.5
	Average Pressure Abs.	#/□"	39.45	39.75	39.8	49.82	49.82	49.81	59.81	59.8	59.85	74.81	75.25	74.75
	Saturation Temp.	°F	11.05	11.38	11.44	21.5	21.5	21.49	30.06	30.05	30.09	41.0	41.3	40.96
Pounds of Air per Min.		#/min	92	63.5	40.8	92	63.5	40.8	92	63.5	40.8	92	63.5	40.8
ΔH per Pound of Air		BTU's/#	5.0	5.5	6.4	3.8	4.2	4.8	2.7	3.2	3.8	2.1	2.3	2.9
B.T.U.'s per Hour		BTU's/hr	27,600	20,960	15,670	20,780	16,000	11,750	14,890	12,190	9,302	11,590	8,763	7,099
L.M.T.D.** Air-NH_3		°F	81.0	79.0	79.3	72.1	70.7	69.3	64.9	63.3	63.0	54.3	54.6	52.9
Coef. of Heat Transfer		BTU/°F □' hr	10.2	7.98	5.94	8.75	6.81	5.10	6.90	5.79	4.44	6.42	4.83	4.04
Air Velocity Entering Coil		Ft/min	2250	1560	1008	2235	1551	994	2238	1549	1010	2235	1555	1010
ΔT of Air In and NH_3		°F	91.0	89.6	88.4	79.4	79.5	79.0	70.4	69.9	69.6	58.8	59.3	58.8
Coef. Based on ΔT of Air In and NH_3		BTU/°F □' hr	9.12	7.04	5.33	7.95	6.05	4.47	6.36	5.24	4.02	5.93	4.44	3.63

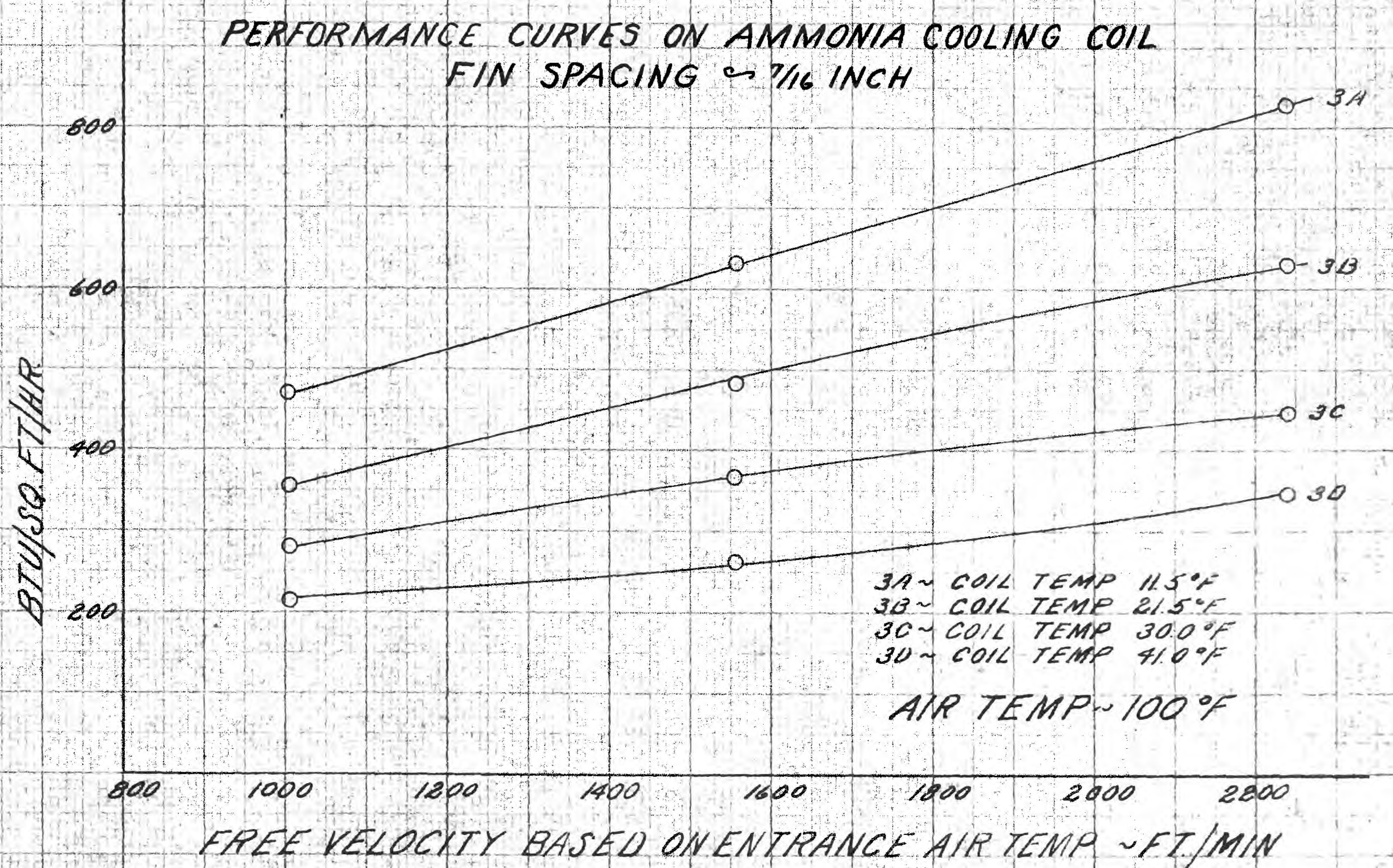
Note * In. Al. = Inches Alcohol ** L.M.T.D. = Log Mean Temperature Difference

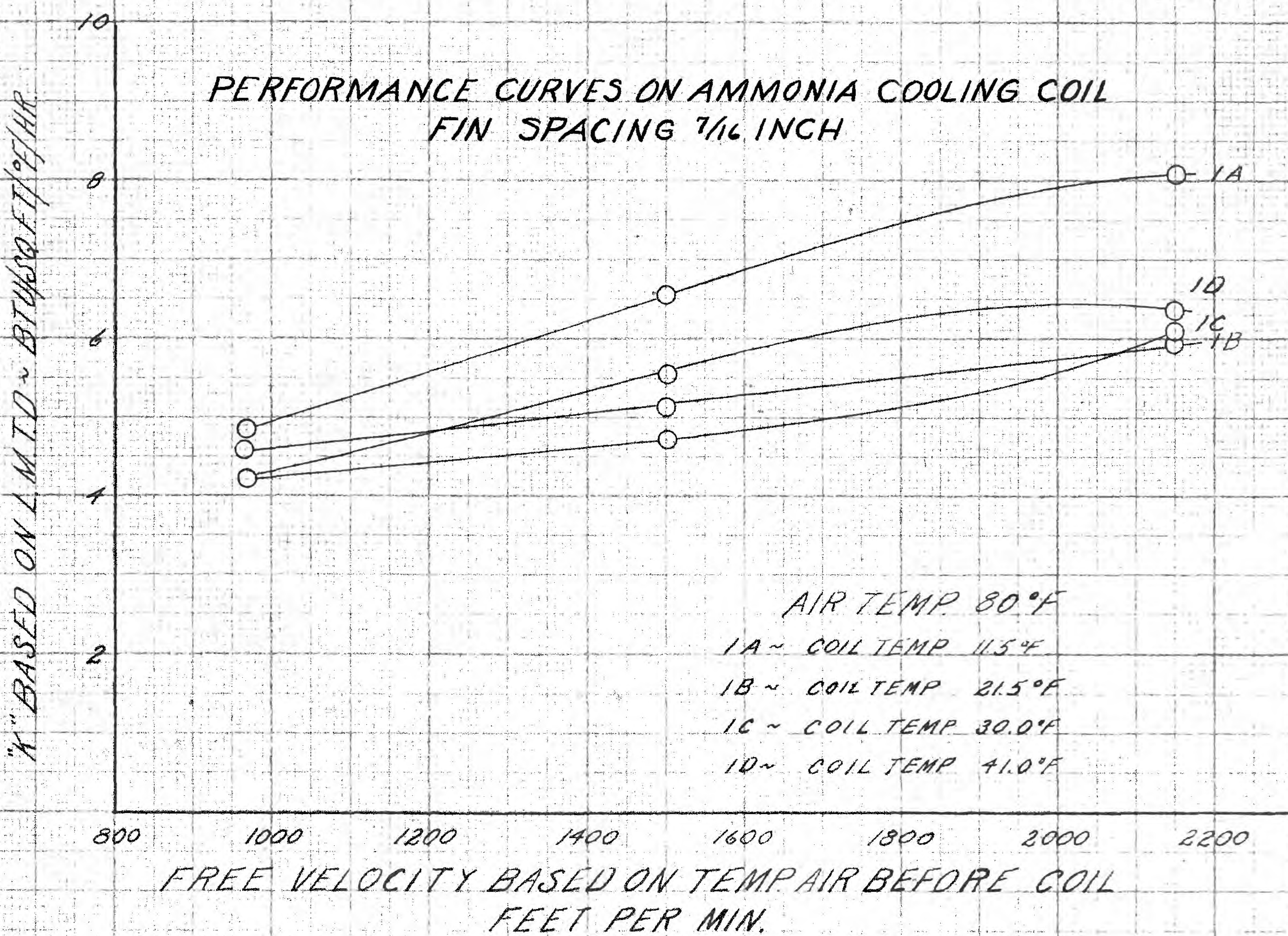
PERFORMANCE CURVES ON AMMONIA COOLING COIL
FIN SPACING ~ 7/16 INCH

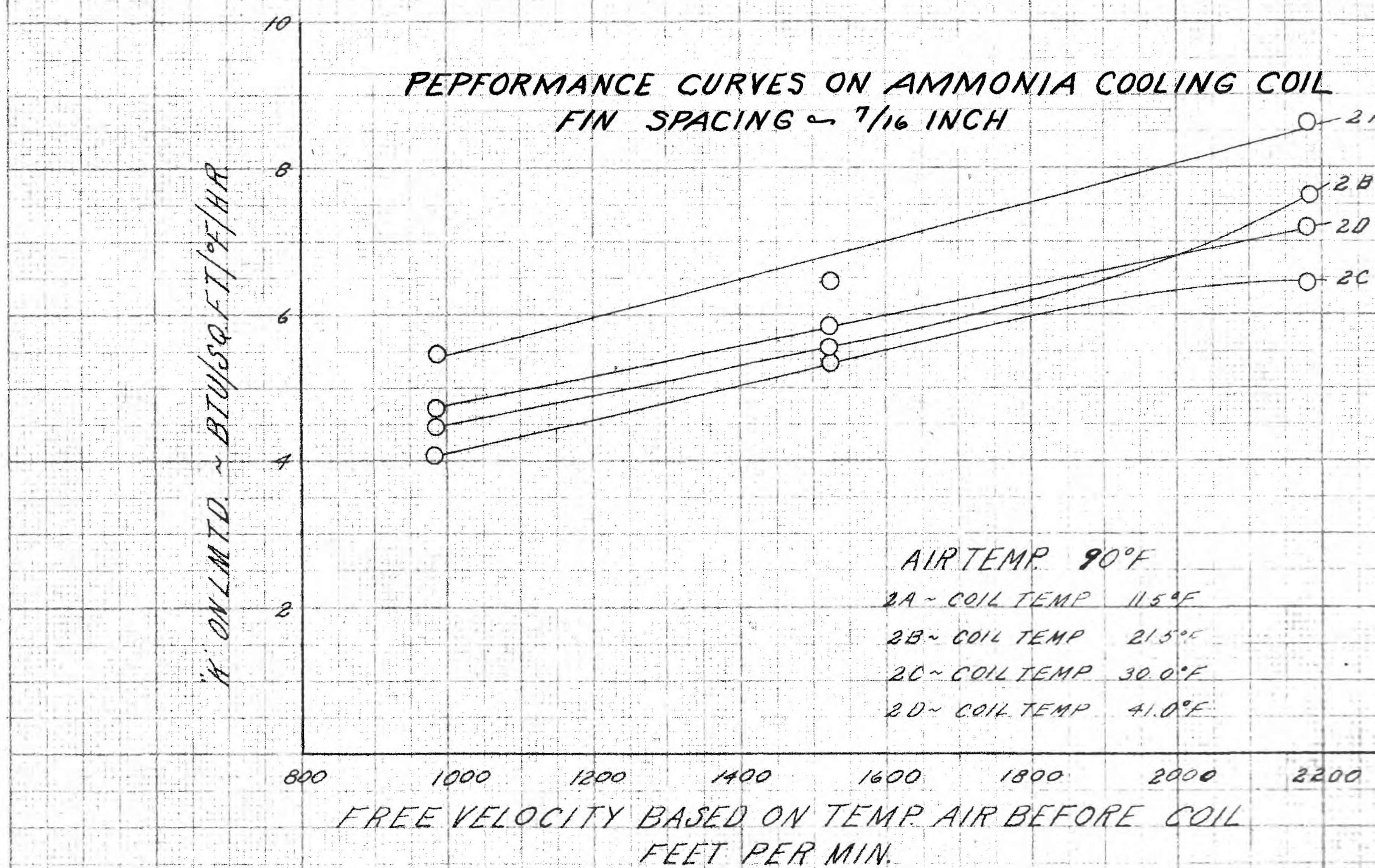


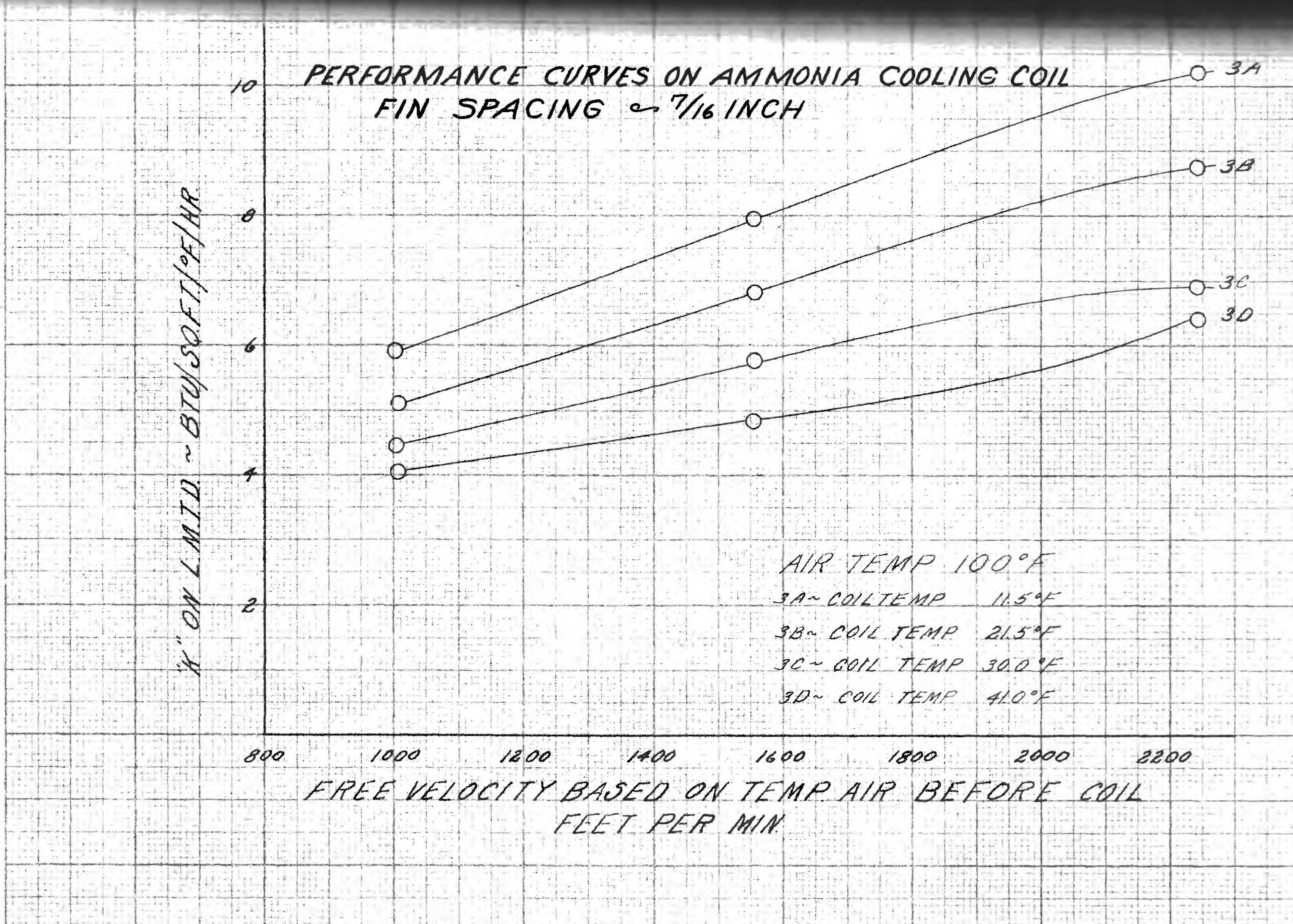
PERFORMANCE CURVES ON AMMONIA COOLING COIL
FIN SPACING ~ 7/16 INCH

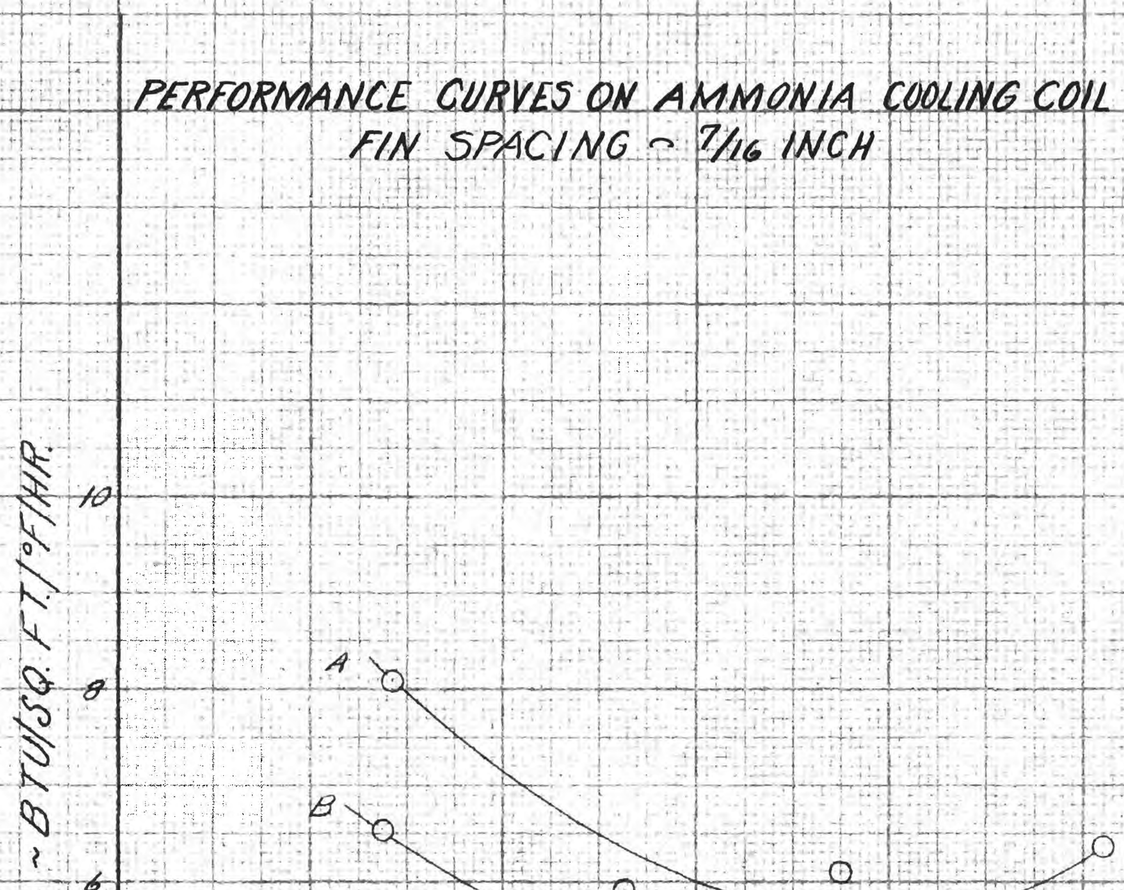


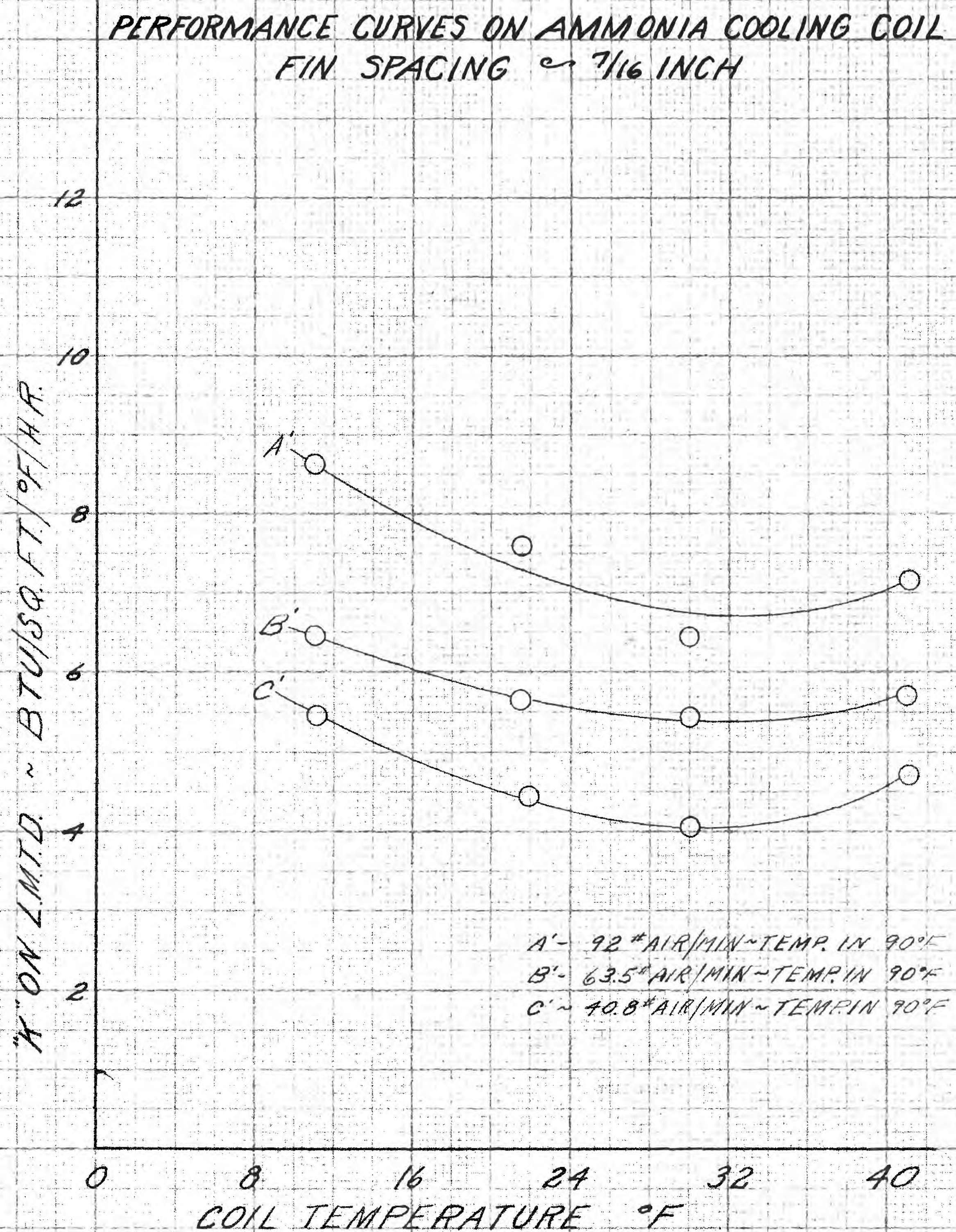

PERFORMANCE CURVES ON AMMONIA COOLING COIL
FIN SPACING ~ 7/16 INCH
BTU/SQ. FT/HR
800
600
400
200
800
1000
1200
1400
1600
1800
2000
2200
FREE VELOCITY BASED ON ENTRANCE AIR TEMP ~FT./MIN
3A
3B
3C
3D
3A ~ COIL TEMP 11.5°F
3B ~ COIL TEMP 21.5°F
3C ~ COIL TEMP 30.0°F
3D ~ COIL TEMP 41.0°F
AIR TEMP ~ 100°F

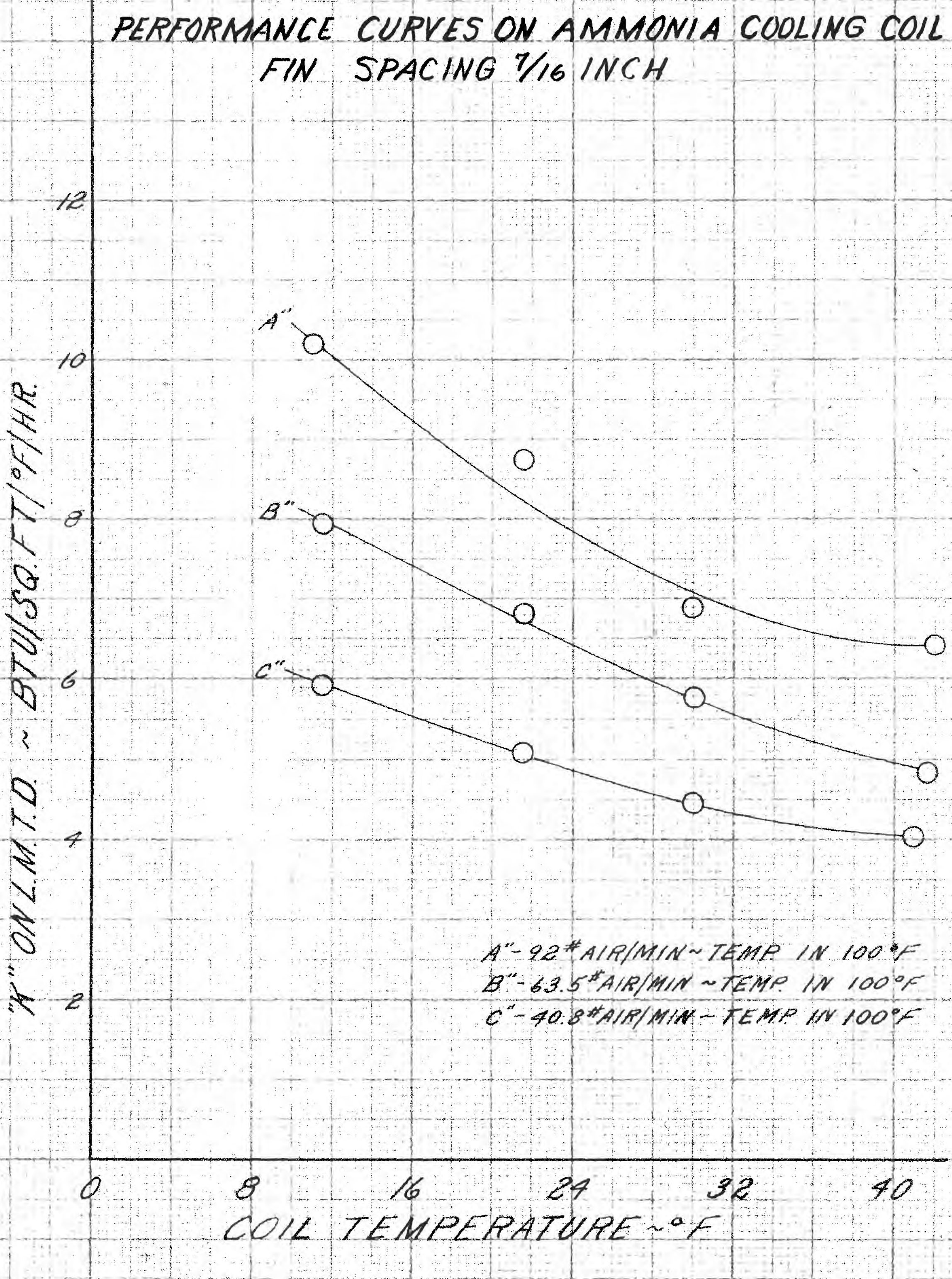

PERFORMANCE CURVES ON AMMONIA COOLING COIL
FIN SPACING 7/16 INCH
"K" BASED ON L.M.T.D. ~ BTU/SQ.FT/°F/HR
10
8
6
4
2
800
1000
1200
1400
1600
1800
2000
2200
FREE VELOCITY BASED ON TEMP AIR BEFORE COIL
FEET PER MIN.
1A
1D
1C
1B
AIR TEMP 80°F
1A ~ COIL TEMP 11.5°F
1B ~ COIL TEMP 21.5°F
1C ~ COIL TEMP 30.0°F
1D ~ COIL TEMP 41.0°F


PEPFORMANCE CURVES ON AMMONIA COOLING COIL
FIN SPACING ~ 7/16 INCH
'K' ON L.M.T.D. ~ BTU/SQ.FT/°F/HR.
10
8
6
4
2
2A
2B
2D
2C
AIR TEMP 90°F
2A ~ COIL TEMP 11.5°F
2B ~ COIL TEMP 21.5°F
2C ~ COIL TEMP 30.0°F
2D ~ COIL TEMP 41.0°F
800
1000
1200
1400
1600
1800
2000
2200
FREE VELOCITY BASED ON TEMP. AIR BEFORE COIL
FEET PER MIN.


PERFORMANCE CURVES ON AMMONIA COOLING COIL
FIN SPACING ~ 7/16 INCH
"K" ON L.M.T.D. ~ BTU/SQ.FT./°F/HR.
10
8
6
4
2
3A
3B
3C
3D
AIR TEMP 100°F
3A ~ COIL TEMP 11.5°F
3B ~ COIL TEMP 21.5°F
3C ~ COIL TEMP 30.0°F
3D ~ COIL TEMP 41.0°F
800
1000
1200
1400
1600
1800
2000
2200
FREE VELOCITY BASED ON TEMP. AIR BEFORE COIL
FEET PER MIN.


PERFORMANCE CURVES ON AMMONIA COOLING COIL
FIN SPACING ~ 7/16 INCH
"K" BASED ON L.M.T.D. ~ BTU/SQ. FT./°F/HR.
10
8
6
4
2
A
B
C
A ~ 92# AIR/MIN ~ TEMP. IN 80°F
B ~ 63.5# AIR/MIN ~ TEMP. IN 80°F
C ~ 40.8# AIR/MIN ~ TEMP. IN 80°F
0 4 8 12 16 20 24 28 32 36 40
COIL TEMPERATURE ~ °F


PERFORMANCE CURVES ON AMMONIA COOLING COIL
FIN SPACING ~ 7/16 INCH
"K" ON L.M.T.D. ~ BTU/SQ. FT./°F/HR.
12
10
8
6
4
2
A'
B'
C'
A' ~ 92 # AIR/MIN ~ TEMP. IN 90°F
B' ~ 63.5 # AIR/MIN ~ TEMP. IN 90°F
C' ~ 40.8 # AIR/MIN ~ TEMP. IN 90°F
0
8
16
24
32
40
COIL TEMPERATURE °F


PERFORMANCE CURVES ON AMMONIA COOLING COIL
FIN SPACING 7/16 INCH
"K" ON L.M.T.D. ~ BTU/SQ.FT/°F/HR.
COIL TEMPERATURE ~ °F
A"
B"
C"
A" - 92# AIR/MIN ~ TEMP. IN 100°F
B" - 63.5# AIR/MIN ~ TEMP. IN 100°F
C" - 40.8# AIR/MIN ~ TEMP. IN 100°F
0
8
16
24
32
40
2
4
6
8
10
12

CONCLUSIONS

1. With a limited velocity range, the heat transfer per square foot per degree F. per hour based on log M.T.D. between ammonia and air varies with some power of the velocity.

$$BTU/Sq.\ ft./^{o}F/hr. = CV^{n}$$

(a). This statement is thought to be true for total dry-coil or wet coil operation over a relatively small velocity range.

(b). From curves obtained, it is concluded that "n" also varies with varying coil temperatures and entering air temperatures.

(c). Values of "n" were checked and found to vary from .2 to .8.

3. The purpose previously described, has been partially fulfilled:

(a) For the particular coil, under operating conditions specified, curves giving the desired information are available.

(b) An analysis was made based on the thermocouple data.

4. How Should a Coil Be Rated?

The following schedule is proposed on testing and rating coils:

(1). Coils should be classified as to material used in construction.

(2). A second necessary consideration is to refrigerant used.

(3). With the material and refrigerant determined, the following tests should be made:

(a) Select one type fin.

(b) Vary conditions:

1. Entrance air temp. - 70, 80, 90, 100°F
2. Coil Temperature - 10, 20, 30, 40°F
3. Quantity of air flowing or velocity from 200 ft/min - 1500 ft/min in desired increments.

(c) Run tests with dry coil and wet coil.

(d) A straight, plat-type, continuous fin is recommended as a basis for comparison.

*****

RESULTS ON AMMONIA COIL PERFORMANCE BASED ON SURFACE TEMPERATURES OBTAINED BY THE USE OF THERMOCOUPLES

An average surface temperature was obtained by taking an average of the thermocouple readings. This average was made on a relative weight basis. By an examination of the thermocouple positions shown on pages 16 and 17, it was concluded that a direct arithmetic average would not give a representative surface temperature due to the fact that various thermocouples represented areas of different size. The readings were averaged according to the following weights:

Couple #	Weight	Couple #	Weight	Couple #	Weight
10	1	4	2	1	4
11	1	6	2	2	4
12	1	15	2	3	4
13	1	16	2	8	4
14	1				
17	1				
T_3	1				

Thermocouples representing the smallest areas were given a weight of unity. One tube temperature was taken since the tube area represented only 14.1% of the total cooling surface.

A tabulated result sheet is shown on page 54.

Families of curves showing the coil performance were included.

The logarithmic mean temperature difference between the surface and the air was calculated in the same manner as the coefficient based on the ammonia saturation temperature.

$$L.M.T.D. = \frac{(T_s - t_2) - (T_s - t_1)}{\log_e \dfrac{T_s - t_2}{T_s - t_1}}$$

$$= \frac{t_1 - t_2}{\log_e \dfrac{T_s - t_2}{T_s - t_1}}$$

T_s = Surface Temperature - °F

t_1 = Dry Bulb Temperature of Air before ammonia coil - °F

t_2 = Dry bulb temperature of air after ammonia coil - °F

$\log_e$ = Logarithm, Base "e"

The designation, free velocity, as appears on the curves means the velocity in feet per minute based on the temperature of the air entering the ammonia coil and the free frontal area.

The predicated curves shown dotted, were plotted by taking ratios between the space increments and velocity increments.

CONCLUSIONS REGARDING THERMOCOUPLES

1. It is evident from the result sheet that there is a general decrease in surface temperature with a decrease in air velocity with constant coil and entering air temperature.

2. The surface temperature decreases toward the downstream end of the coil.

3. There is a rise in surface temperature with an increase in entering air temperature.

4. The average temperature of the coil approaches the dew point of the entering air.

5. The variation in temperature recorded by the individual thermocouples was inconsistent from run to run.

An attempt at an explanation is made:

(a). The bond between tube and fin is a consideration.

1. Corrosion or looseness would tend toward a decrease in heat transfer rate.

(b). The fact that there is a pressure drop through the coil would affect the coil temperature.

(c). In successive points throughout the tubes varying degrees of turbulence in the refrigerant would tend to alter the heat transfer rate.

(d). Local eddies set up in the particular coil tested would vary from time to time and run to run, a fact which could easily lead to inconsistency.

6. The surface temperature cannot be taken as the refrigerant temperature.

7. From the curves, using surface temperature and BTU/D'/hr. as coordinates, it is readily seen that the heat transfer per square foot per hour increases as the velocity increases. This fact indicates that the resistance to heat transfer through the air film decreases with increased velocity.

*****

SAMPLE THERMOCOUPLE DATA SHEET

RUN #1 SERIES 1D

MICROVOLTS																				
#	1	2	3	4	5	6	8	9	10	11	12	13	14	15	16	17	T_1	T_2	T_3	T_4
3-45	697	720	697	767	778	770	762	790	809	862	758	764	821	781	726	845	723	897	784	803
	695	718	695	768	772	777	764	795	786	857	758	768	824	787	736	840	719	904	787	803
	685	709	679	773	783	777	766	793	800	863	762	761	814	787	734	836	705	882	777	785
	674	700	671	758	775	771	763	782	803	850	751	762	812	777	732	833	712	880	776	789
	670	697	667	749	760	751	750	776	784	849	748	755	811	774	723	831	715	872	749	764
	666	692	667	757	765	758	750	773	770	853	737	757	802	769	723	828	707	878	771	791
	671	696	685	760	773	765	758	779	805	857	745	754	810	779	719	830	703	873	769	781
4-15	673	694	673	754	771	772	753	778	794	850	738	758	800	774	726	828	711	874	773	781
AVG	679	703	679	761	772	768	758	783	794	855	750	759	810	779	727	834	712	882	773	787
TEMP	62.5	63.6	62.5	66.2	66.7	66.6	66.1	67.2	67.7	70.2	65.7	66.1	68.4	67.0	64.7	69.5	64.0	71.7	66.8	67.4

THERMOCOUPLE DATA AND RESULTS

EMF-MICROVOLTS TEMP °F 30 MINUTE AVERAGE

RUN #	1		2		3		1		2		3		1		2		3	
SERIES	1A		1A		1A		1B		1B		1B		1C		1C		1C	
COUPLE #	EMF	TEMP	EMF	TEMP	EMF	TEMP	EMF	TEMP	EMF	TEMP	EMF	TEMP	EMF	TEMP	EMF	TEMP	EMF	TEMP
1	390	49.5	347	47.6	426	51.1	592	58.6	506	54.7	510	54.9	601	59.0	540	56.3	548	56.6
2	286	44.9	328	46.6	305	45.7	615	59.6	496	54.3	458	52.6	621	59.9	549	56.7	513	55.0
3	305	45.8	330	46.9	370	48.6	629	60.3	495	54.3	418	50.8	573	57.8	510	54.9	467	53.0
4	507	54.8	453	52.5	530	55.8	691	63.0	589	58.5	589	58.5	710	63.9	640	60.8	620	59.9
5	409	51.2	459	52.6	416	50.7	723	64.5	605	59.2	550	56.7	708	63.8	647	61.1	588	58.5
6	480	53.6	521	55.4	477	53.5	777	66.9	611	59.5	523	55.5	707	63.8	634	60.5	546	56.6
8	41.5	50.7	455	52.4	402	50.1	722	64.5	597	58.8	523	55.5	707	63.8	639	60.7	549	56.7
9	458	52.6	519	55.3	545	56.5	803	68.1	677	62.7	574	57.8	737	65.1	695	63.3	632	60.4
10	554	56.9	576	57.9	—	—	768	66.5	614	62.3	—	—	766	66.5	701	63.5	—	—
11	563	57.3	608	59.3	600	59.0	859	70.6	749	65.7	649	61.2	823	69.0	772	66.7	705	63.7
12	374	48.8	430	51.3	412	50.5	723	64.5	581	58.4	500	54.5	676	62.4	619	59.8	545	56.5
13	483	53.6	485	53.8	487	53.9	715	64.2	609	59.4	565	57.4	712	64.0	653	61.3	599	58.9
14	627	60.2	579	58.0	554	56.9	767	66.5	687	62.9	596	58.8	755	65.9	697	63.3	628	60.3
15	505	54.7	500	54.5	435	51.5	717	64.3	629	60.3	527	55.7	702	63.5	650	61.2	558	57.1
16	364	48.4	385	48.3	255	43.5	679	62.6	553	56.8	385	49.3	646	61.0	577	57.9	434	51.5
17	647	61.1	630	60.3	581	58.1	843	69.9	745	65.5	611	59.5	792	67.6	739	65.2	632	60.4
T-1	418	50.8	479	53.5	431	51.4	698	63.4	583	58.2	444	51.9	642	60.9	576	57.9	486	53.9
T 2	646	61.0	672	62.2	604	59.1	882	71.6	804	68.1	680	62.6	851	70.3	821	68.9	731	64.9
T 3	462	52.8	496	54.3	470	53.1	728	64.8	634	60.5	579	58.0	723	64.5	677	62.5	609	59.4
T4	576	57.9	511	54.9	414	50.6	702	63.6	637	60.6	470	53.1	679	62.5	641	60.8	537	56.1

RUN #	1		2		3		1		2		3		1		2		3	
SERIES	1D		1D		1D		2A		2A		2A		2B		2B		2B	
COUPLE #	EMF	TEMP	EMF	TEMP	EMF	TEMP	EMF	TEMP	EMF	TEMP	EMF	TEMP	EMF	TEMP	EMF	TEMP	EMF	TEMP
1	679	62.5	614	59.6	613	59.6	393	49.6	444	52.0	425	51.1	623	60.0	592	58.6	636	60.6
2	703	63.6	631	60.4	593	58.7	289	45.0	308	45.9	361	48.2	608	59.3	567	57.3	541	56.3
3	679	62.5	615	59.7	585	58.3	371	48.7	338	47.2	487	53.9	626	60.1	551	56.8	548	56.6
4	761	66.2	691	63.1	673	62.3	584	58.2	593	58.7	568	57.5	725	64.6	686	62.9	776	66.9
5	772	66.7	711	64.0	660	61.7	422	51.0	449	52.2	503	54.6	756	66.0	714	64.1	659	61.6
6	768	66.6	695	63.3	637	60.7	594	58.7	493	54.2	618	59.8	845	70.0	721	64.4	674	62.3
8	758	66.1	686	62.8	613	59.6	—	—	—	—	—	—	—	—	—	—	—	—
9	783	67.2	744	65.5	725	64.6	626	60.1	567	57.5	729	64.8	898	72.4	800	68.0	741	65.3
10	794	67.7	741	65.3	—	—	637	60.7	655	61.4	—	—	881	71.6	811	68.4	—	—
11	855	70.2	813	68.6	761	66.2	779	67.0	713	64.1	767	66.5	1005	77.1	906	72.7	836	69.6
12	750	65.7	690	63.0	643	60.9	477	53.5	441	51.8	517	55.3	794	67.7	700	63.5	656	61.5
13	759	66.1	702	63.5	670	62.1	523	55.5	542	56.4	527	55.7	809	68.4	729	64.8	710	63.9
14	810	68.4	757	65.9	638	63.0	786	67.4	744	65.4	645	61.0	911	73.0	830	69.3	783	67.2
15	779	67.0	720	64.4	631	60.4	595	58.7	548	56.6	449	52.2	843	69.9	729	64.8	668	62.0
16	727	64.7	665	61.9	525	55.6	492	54.1	393	49.6	394	49.7	752	65.8	637	60.4	539	56.2
17	834	69.5	782	67.2	714	64.1	936	74.1	772	66.7	750	65.7	1010	77.4	881	71.6	770	66.6
T-1	712	64.0	659	61.6	615	59.7	535	56.0	484	53.8	566	57.4	747	65.6	686	62.9	566	57.5
T 2	882	71.7	848	70.1	818	68.8	814	68.6	810	68.4	683	62.7	981	76.1	957	75.0	778	67.0
T 3	773	66.8	727	64.7	693	63.2	487	53.9	551	56.8	476	53.4	814	68.6	739	65.2	705	63.7
T4	787	67.4	757	66.0	602	59.0	681	62.6	671	62.4	368	48.5	822	69.0	746	65.6	593	58.7

THERMOCOUPLE DATA AND RESULTS

EMF - MICROVOLTS   TEMP °F   30 MINUTE AVERAGE

RUN #	1		2		3		1		2		3	
SERIES	2C		2C		2C		2D		2D		2D	
COUPLE #	EMF	TEMP	EMF	TEMP	EMF	TEMP	EMF	TEMP	EMF	TEMP	EMF	TEMP
1	672	62.3	675	62.3	657	61.5	784	67.2	782	67.2	688	68.1
2	657	61.5	669	62.1	609	59.4	833	69.4	804	65.1	668	62.0
3	599	58.9	634	60.5	601	59.0	862	70.8	806	68.2	629	60.3
4	765	66.4	754	65.9	754	65.9	860	70.7	863	70.8	787	67.4
5	780	67.1	782	67.2	723	64.5	928	73.7	896	72.3	778	67.2
6	763	66.3	754	65.9	708	63.8	1006	77.2	948	72.4	745	65.5
8	—	—	—	—	—	—	—	—	—	—	—	—
9	847	70.1	841	69.8	802	68.1	1059	79.6	959	75.1	857	70.5
10	899	72.4	869	71.1	—	—	985	76.3	937	74.1	—	—
11	995	76.7	943	74.4	906	72.7	1159	84.1	1039	78.6	952	74.3
12	799	68.0	768	66.5	690	63.0	1001	77.0	854	71.7	751	65.8
13	833	69.5	800	68.0	735	65.0	955	74.9	891	72.0	785	67.3
14	924	73.5	870	71.1	795	67.7	996	76.7	947	74.6	825	69.1
15	881	71.6	809	68.4	709	63.9	988	76.4	899	72.4	748	65.6
16	786	67.4	705	63.7	635	60.5	982	76.1	850	70.2	697	62.9
17	1000	76.9	923	73.5	820	68.9	1130	82.8	999	76.9	856	70.5
T1	659	61.6	723	64.5	624	60.0	921	73.4	862	70.8	671	62.2
T2	994	76.6	1010	77.4	964	75.3	1106	81.7	1069	80.0	992	76.6
T3	851	70.3	813	68.5	779	67.0	987	76.3	917	73.2	836	69.6
T4	853	70.4	830	69.3	719	64.3	938	74.2	906	72.7	789	67.5

RUN #	1		2		3		1		2		3	
SERIES	3A		3A		3A		3B		3B		3B	
COUPLE #	EMF	TEMP	EMF	TEMP	EMF	TEMP	EMF	TEMP	EMF	TEMP	EMF	TEMP
1	411	50.5	398	49.9	421	51.0	677	62.4	631	60.4	673	62.3
2	288	45.0	249	43.2	321	46.5	555	56.9	479	53.5	574	57.8
3	550	56.7	319	46.4	480	53.6	588	58.4	544	56.4	579	58.0
4	693	63.2	636	60.6	627	60.2	850	70.2	776	66.9	844	70.0
5	431	51.4	378	49.0	449	52.2	715	64.2	633	60.5	724	64.6
6	840	69.8	414	50.6	620	59.9	811	68.5	678	62.5	706	63.8
8	—	—	—	—	—	—	—	—	—	—	—	—
9	801	68.0	533	56.0	750	65.7	902	72.5	786	67.4	861	70.7
10	720	64.4	660	61.7	—	—	—	—	—	—	—	—
11	916	73.2	733	64.9	864	70.8	1057	79.5	943	74.4	974	75.8
12	491	54.1	345	47.5	479	53.5	753	65.9	500	54.5	692	63.1
13	591	58.6	511	55.0	533	55.9	846	70.1	713	64.0	773	66.8
14	811	68.5	807	68.3	637	60.6	1061	79.7	937	73.9	868	71.0
15	564	57.3	540	56.3	400	50.0	873	71.2	712	64.0	714	64.1
16	392	49.6	339	47.2	303	45.6	731	64.9	560	57.2	539	56.2
17	992	76.6	809	68.4	794	67.7	1134	83.0	959	75.1	872	71.2
T1	791	67.6	509	54.9	605	59.2	755	66.0	669	61.8	625	60.1
T2	902	72.5	880	71.6	774	66.8	1069	80.0	1050	79.2	1038	78.6
T3	475	53.3	464	52.9	482	53.7	821	69.0	708	63.8	809	68.4
T4	850	70.8	736	65.1	644	60.9	928	73.7	815	68.6	834	69.5

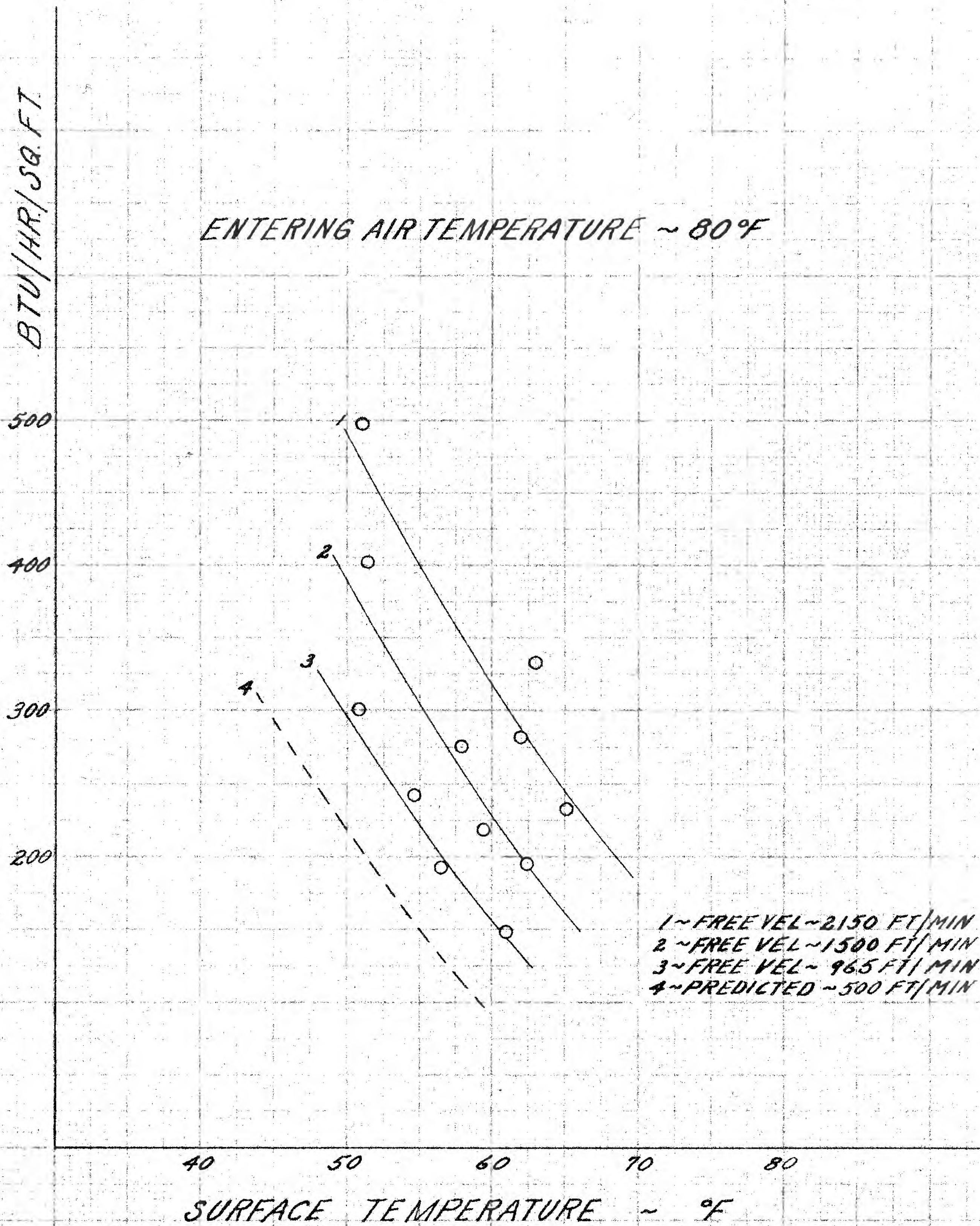
RUN #	1		2		3		1		2		3	
SERIES	3C		3C		3C		3D		3D		3D	
COUPLE #	EMF	TEMP	EMF	TEMP	EMF	TEMP	EMF	TEMP	EMF	TEMP	EMF	TEMP
1	743	65.4	807	68.3	732	64.9	991	76.5	965	75.4	826	69.2
2	670	62.1	657	61.5	628	60.2	982	76.1	947	74.6	775	66.8
3	741	65.3	654	61.4	610	59.4	923	73.5	894	72.2	721	64.4
4	908	72.8	926	73.6	879	71.5	1099	81.4	1017	80.4	937	74.1
5	797	67.8	803	68.1	805	68.2	1117	82.2	1096	81.3	940	74.2
6	882	71.6	791	67.6	797	67.8	1085	80.8	1033	78.4	895	72.2
8	—	—	—	—	—	—	—	—	—	—	—	—
9	930	73.8	872	71.2	937	74.1	1170	84.6	1134	83.0	1036	78.6
10	—	—	—	—	—	—	—	—	—	—	—	—
11	1107	81.8	1021	77.9	1045	79.0	1317	91.3	1219	86.8	1148	83.6
12	819	68.8	762	66.3	788	67.4	1124	82.5	1059	79.6	901	72.5
13	905	72.7	884	71.7	838	69.7	1181	85.1	1102	81.6	930	73.8
14	1067	79.8	1014	77.5	900	72.5	1201	86.0	1126	82.7	964	75.3
15	946	74.5	881	71.6	777	67.0	1181	85.1	1069	80.1	892	72.1
16	827	69.2	722	64.5	683	62.7	1105	81.7	1001	77.0	808	68.3
17	1154	83.9	1009	77.3	939	74.2	1256	88.5	1152	83.8	1020	77.8
T1	761	66.2	731	64.9	703	63.6	937	74.1	948	74.6	793	67.7
T2	1149	83.7	1106	81.7	1102	81.6	1317	91.3	1277	89.5	1190	85.5
T3	883	71.7	892	72.1	886	71.8	1209	86.4	1128	82.7	984	76.2
T4	902	72.5	896	72.3	810	68.4	1044	78.9	1007	77.2	901	72.5

TABULATED DATA AND RESULTS BASED ON THERMOCOUPLES

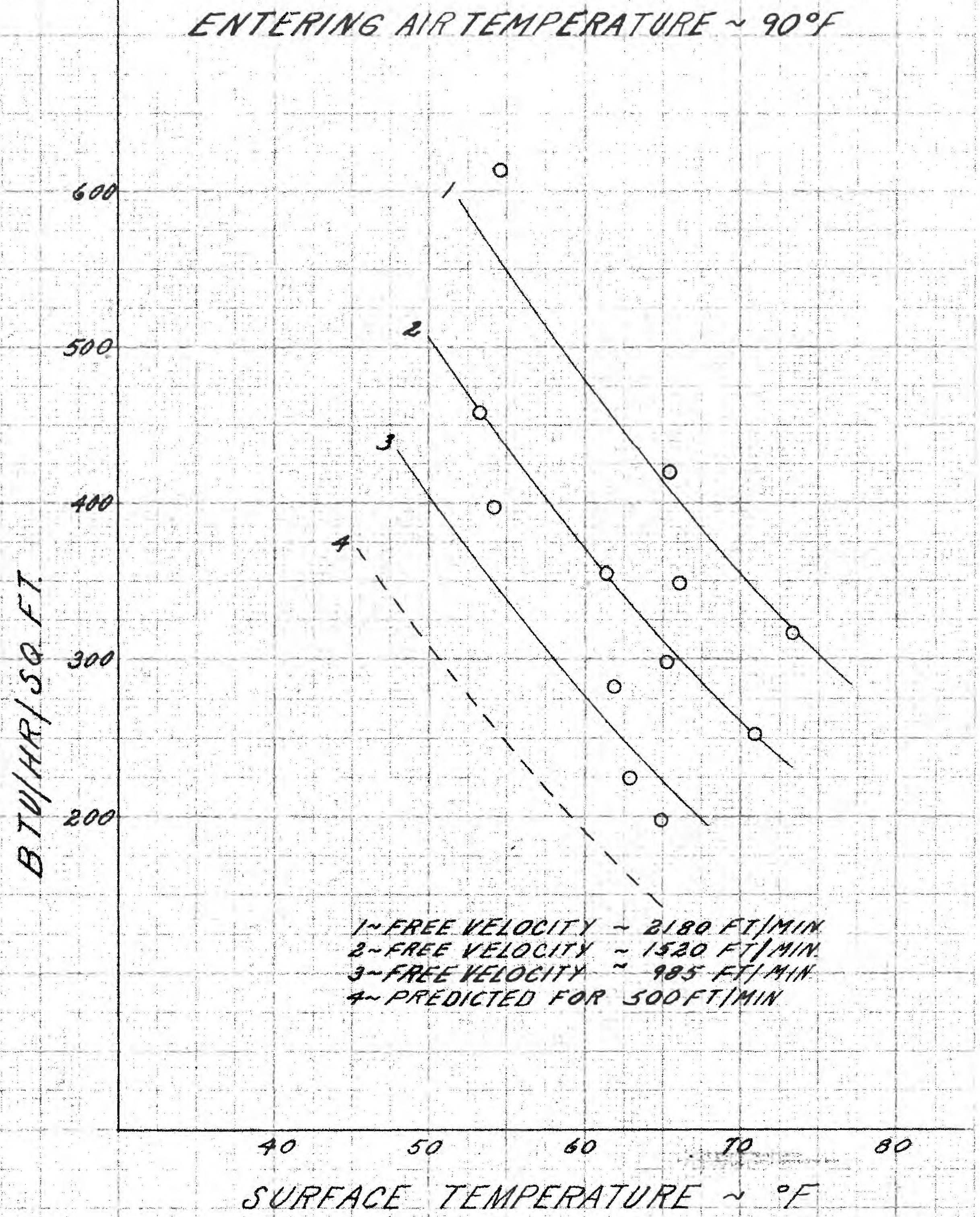
SERIES		1A			1B			1C			1D		
RUN	UNITS	1	2	3	1	2	3	1	2	3	1	2	3
VELOCITY ENTERING COIL	FT/MIN	2150	1500	965	2150	1500	965	2150	1500	965	2150	1500	965
AMMONIA SATURATION TEMP	°F	11.5	11.5	11.44	21.5	21.5	21.5	30.0	30.0	30.0	41.0	41.0	41.0
FIN SURFACE TEMP	°F	51	51.4	50.7	63.0	57.9	54.7	62.1	59.3	56.4	65.2	62.4	60.8
ENTERING AIR TEMP DRY BULB	°F	79.0	79.8	79.1	81.0	80.0	79.5	80.2	80.2	79.6	80.4	79.7	79.9
LEAVING AIR TEMP DRY BULB	°F	68.0	66.0	68.4	74.0	71.0	70.4	73.8	73.0	71.1	74.7	73.4	72.6
LOG MEAN TEMP. DIFF	°F	22.0	20.7	22.45	14.35	17.15	19.90	14.60	16.95	18.30	12.11	13.95	15.11
COEFFICIENT OF HEAT TRANSFER	BTU/□' HR °F	22.6	19.4	13.45	23.10	16.05	12.20	19.30	12.90	10.45	19.10	14.0	9.75

SERIES		2A			2B			2C			2D		
RUN	UNITS	1	2	3	1	2	3	1	2	3	1	2	3
VELOCITY ENTERING COIL	FT/MIN	2180	1520	985	2180	1520	985	2180	1520	985	2180	1520	985
AMMONIA SATURATION TEMP	°F	11.5	11.5	11.44	21.5	21.5	21.5	30.0	30.0	30.0	41.0	41.0	41.0
FIN SURFACE TEMP	°F	54.3	53.3	54.1	65.3	61.8	61.8	66.0	65.2	62.8	73.3	70.7	64.8
ENTERING AIR TEMP DRY BULB	°F	90.2	90.2	90.0	90.2	90.3	90.8	89.0	90.2	90.0	89.0	90.0	89.0
LEAVING AIR TEMP DRY BULB	°F	75.2	75.6	79.9	80.5	79.5	79.3	80.0	80.2	79.5	81.0	81.1	79.1
LOG. MEAN TEMP DIFF	°F	27.65	28.80	30.70	19.60	22.70	22.75	18.20	19.50	21.60	11.20	14.35	18.85
COEFFICIENT OF HEAT TRANSFER	BTU/□' HR °F	22.2	15.9	13.0	24.6	15.65	12.45	19.10	15.30	10.40	28.10	17.60	10.55

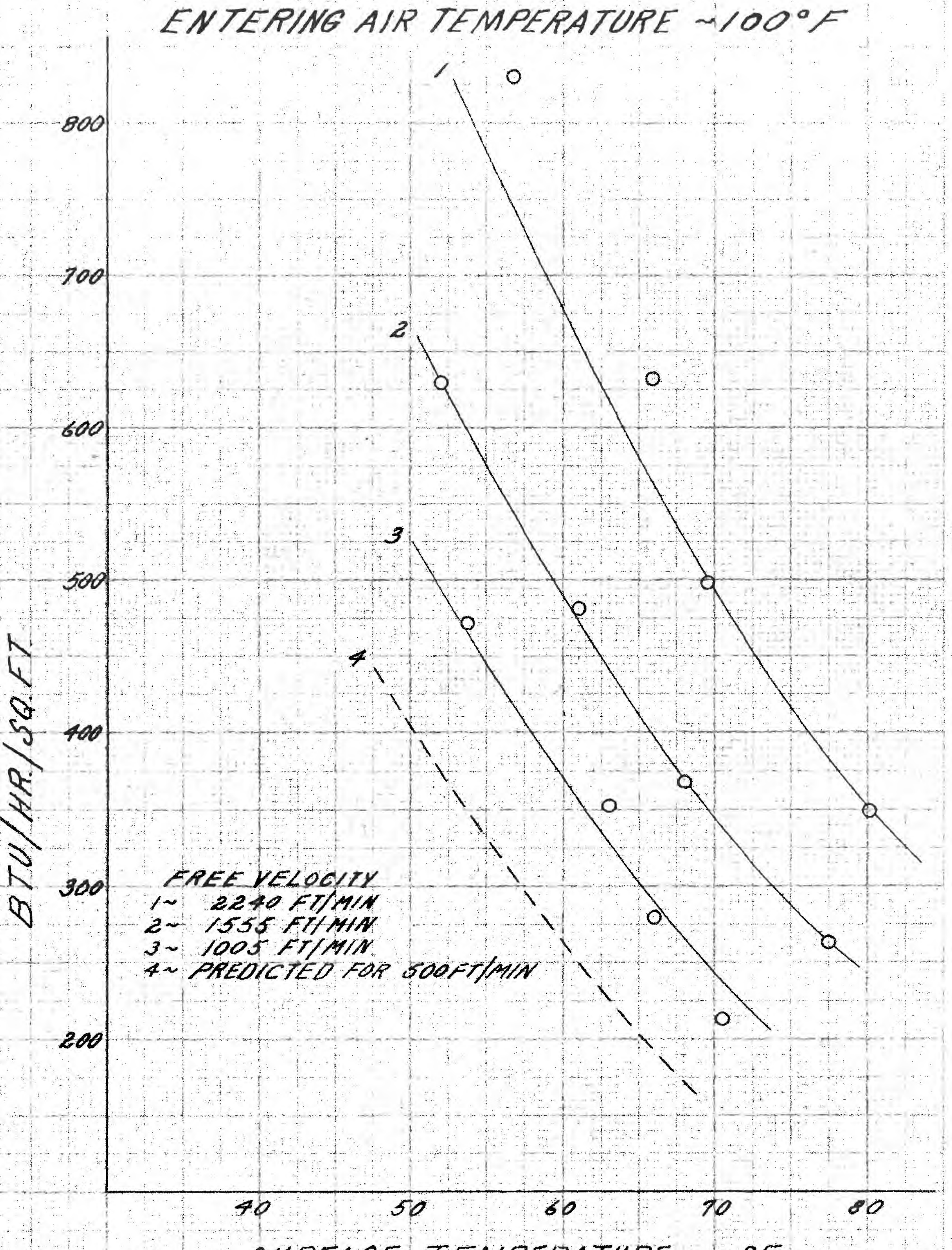
SERIES		3A			3B			3C			3D		
RUN	UNITS	1	2	3	1	2	3	1	2	3	1	2	3
VELOCITY ENTERING COIL	FT/MIN	2240	1555	1005	2240	1555	1005	2240	1555	1005	2240	1555	1005
AMMONIA SATURATION TEMP	°F	11.5	11.5	11.44	21.5	21.5	21.5	30.0	30.0	30.0	41.0	41.0	41.0
FIN SURFACE TEMP	°F	56.9	52.0	53.8	65.7	61.0	63.0	69.5	67.9	65.9	80.2	77.5	70.5
ENTERING AIR TEMP DRY BULB	°F	102.1	101.0	99.8	100.9	101.0	100.5	100.5	100.0	99.7	99.8	100.6	99.8
LEAVING AIR TEMP. DRY BULB	°F	83.1	80.9	81.5	86.4	84.7	82.0	89.5	86.9	86.6	91.0	91.7	88.3
LOG MEAN TEMP. DIFF	°F	34.70	39.90	36.20	27.40	31.10	27.30	25.10	24.95	26.80	14.70	18.25	22.95
COEFFICIENT OF HEAT TRANSFER	BTU/□' HR °F	23.90	16.60	13.0	23.0	15.50	12.90	17.90	14.70	10.45	238	14.40	9.35



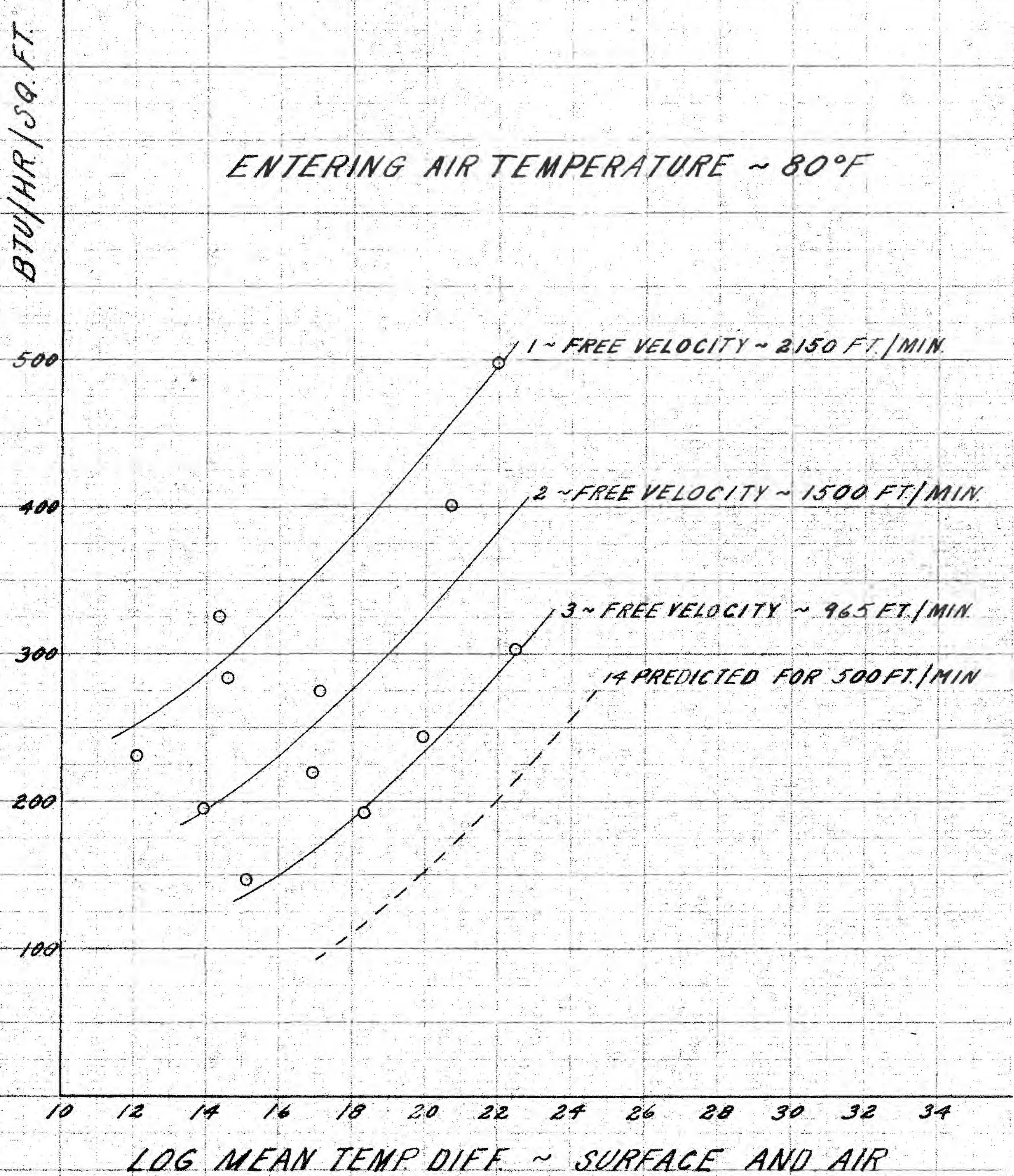
PERFORMANCE CURVES ON AMMONIA COOLING COIL
FIN SPACING ~ 7/16 INCH



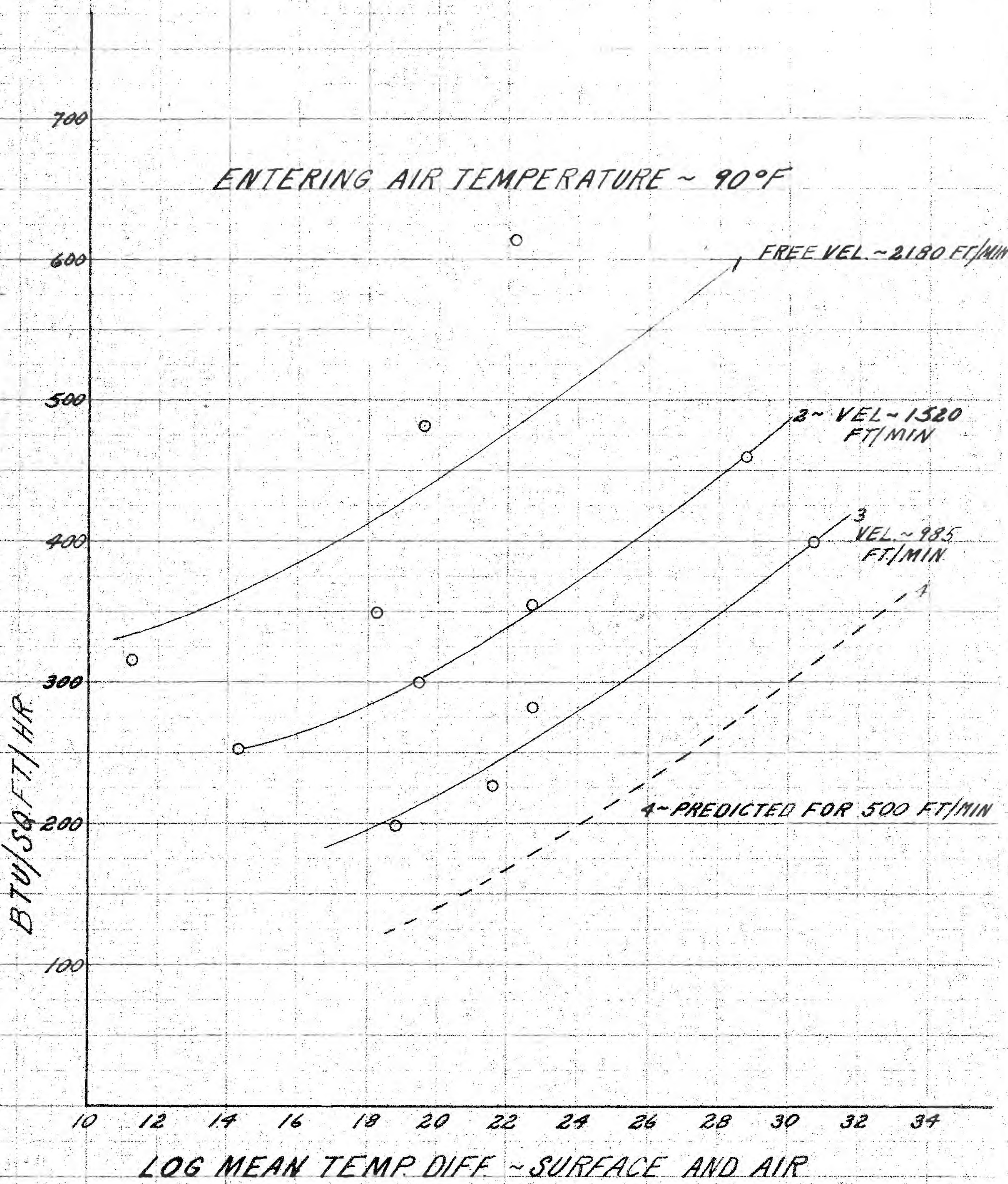
PERFORMANCE CURVES ON AMMONIA COOLING COIL
FIN SPACING ~ 7/16 INCH



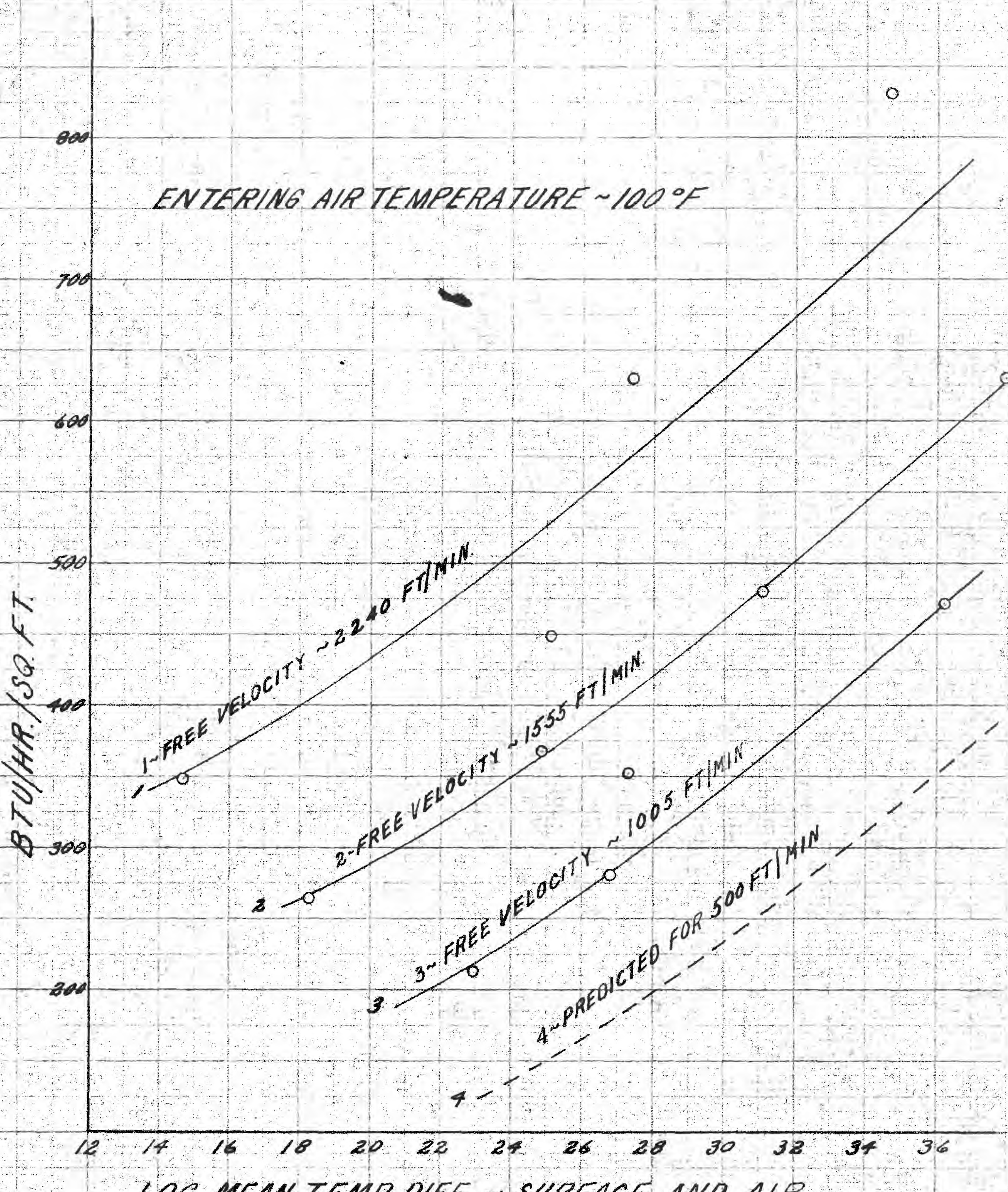
PERFORMANCE CURVES ON AMMONIA COOLING COIL
FIN SPACING ~ 7/16 INCH



PERFORMANCE CURVES ON AMMONIA COOLING COIL
FIN SPACING ~ 7/16 INCH



PERFORMANCE CURVES ON AMMONIA COOLING COIL
FIN SPACING ~ 7/16 INCH



PERFORMANCE CURVES ON AMMONIA COOLING COIL
FIN SPACING ~ 7/16 INCH

APPENDIX

Manometer.

The type micro manometer used needed no calibration or corrections. The density of the alcohol or manometer fluid at the temperature used had to be known.

It consisted of a horizontal bottle half-filled with alcohol connected below the fluid level by a flexible hose to a glass tube with a cross hair engraved on it. The inclined tube was raised or lowered by turning the dial on top of the main screw. The sensitiveness of the instrument was changed by varying the inclination of the tube. (See Page 13). The zero was set after leveling the base and turning the dial to zero. The inclined tube was then raised or lowered relative to the carriage until the bottom of the meniscus was on the cross hair.

In measuring a differential pressure, the lower pressure was connected to the top of the inclined tube and the higher pressure to the reservoir. If pressure above atmospheric was to be measured, the connection would be made to the reservoir, the top of the cross hair left open to the air. For a pressure below that of the atmosphere, this connection was shifted to the inclined tube.

To measure either pressure the dial was turned to raise the inclined tube until the meniscus of the liquid level was opposite the cross hair. Since the main screw had ten threads to the inch, one revolution gave 0.1 inch which was indicated on the main column. The rim of the dial had one hundred equal divisions, so that 0.01 revolution meant .001 vertical travel of the carriage.

The instrument reads directly to 0.001 inch of alcohol and could be estimated to 0.0005 inches. Its accuracy was determinedby the precision with which the main screw was cut. These threads were cut by an expert machinist on a special toolmaker's lathe. Care was taken while turning to prevent heating above approximate room temperature at which it would be used. Temperature changes had no appreciable effect on its accuracy. 20°F. change in temperature affected the length of the brass screw only 0.02%.

The great advantage of the instrument over the usual inclined manometer was that the liquid level in the reservoir was unchanged regardless of the reading so long as the meniscus was at the cross hair, and since the same volume of liquid was in the connecting tube regardless of the reading.

There was no error in uneven capillary effect in a tube of varying bore, since for all readings the liquid was at the same position in the tube.

The density of the ethyl alcohol used was found by comparing the weight of a given volume with that of the same valume of water, both at room temperature. It was found to be .80 at 83°F.

On trying to use distilled water in the manometer, it was found that the meniscus broke when the tube was inclined at slight angles. Alcohol had an additional advantage in that it gave about 25% greater reading for the same pressure.

Theremocouples.

Thermocouples used were made of copper and constantan furnished by Leeds & Northrup. The hot junction was fused and spot soldered at each position on the fins and tubes as the coil was being assembled. The fins were pressed over the tubes.

A common cold junction was used. This consisted of a sheet metal interior around which was placed a two-inch cork-board insulation. The outer finish was of $\frac{1}{4}$" plywood. All joints and cracks were sealed with an asphalt compound.

In the common cold junction were placed pieces of glass tubing sealed at one end and filled with mercury, into which were placed the individual leads. The junction was packed with a sufficient amount of shaved ice and water to hold $32^{o}F.$ constant for better than a day.

A Leeds & Northrup, K-2 potentiometer was used in conjunction with the thermocouples. The range used was from .0000001 to .015 volt.

A sketch showing the position of the thermocouples is shown on pages 16 and 17.

The wiring diagram was shown on page 20.

Values for the calibration curve on standard copper constantan thermocouples were taken from Leeds & Northrup conversion tables with the cold junction at $32^{o}F$[12].

These results were checked and verified by tables for copper-constantan determined by the Geophysical Laboratory.[12]

Sample couples were also independently checked and found to be correct.

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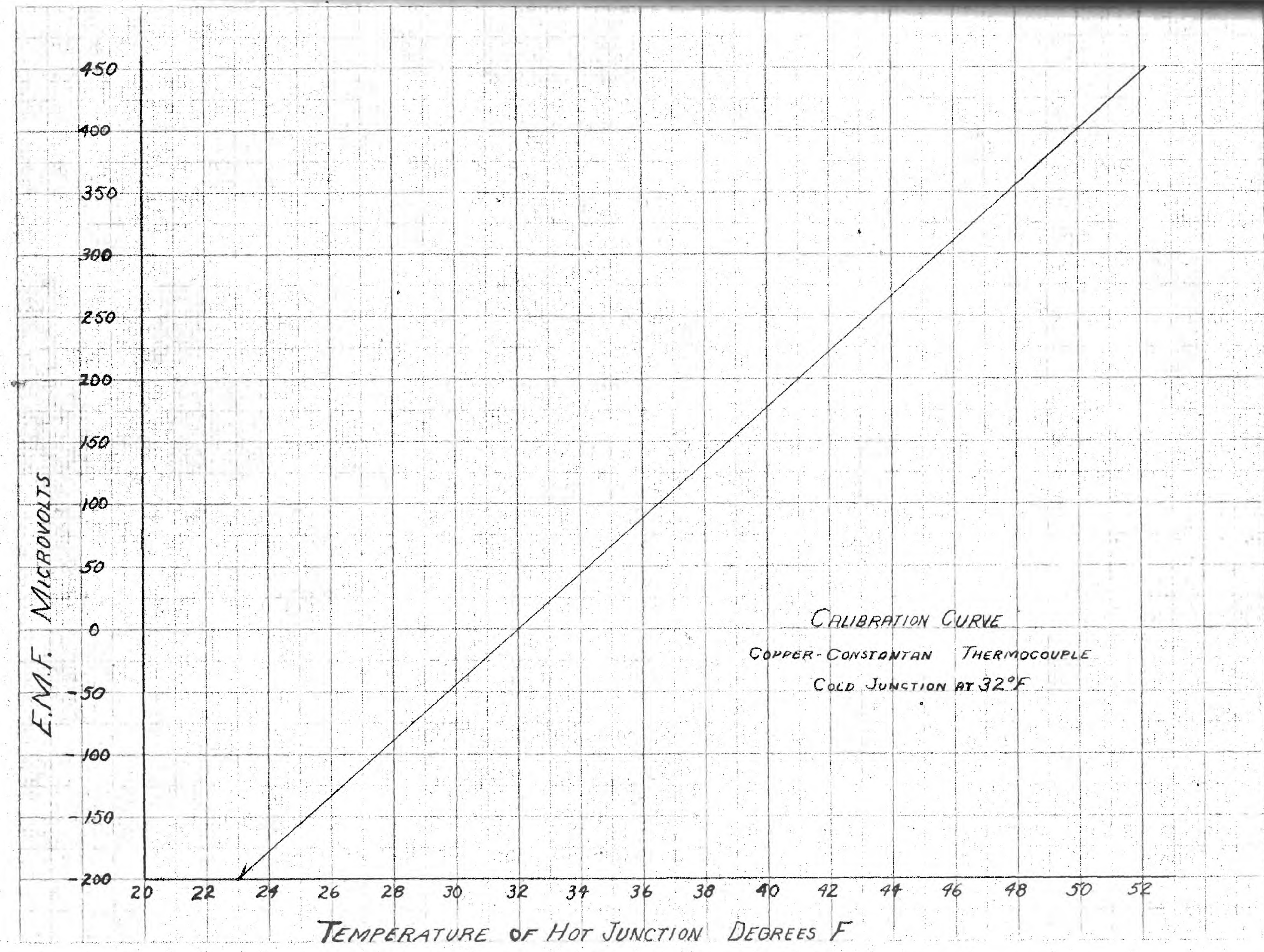
12. Leeds & Northrup Standard Conversion Tables.
12. Adams, Bull, A.I.M.M.M., 159 Pg. 2111, 1919.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-150
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E.M.F. MICROVOLTS
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42
44
46
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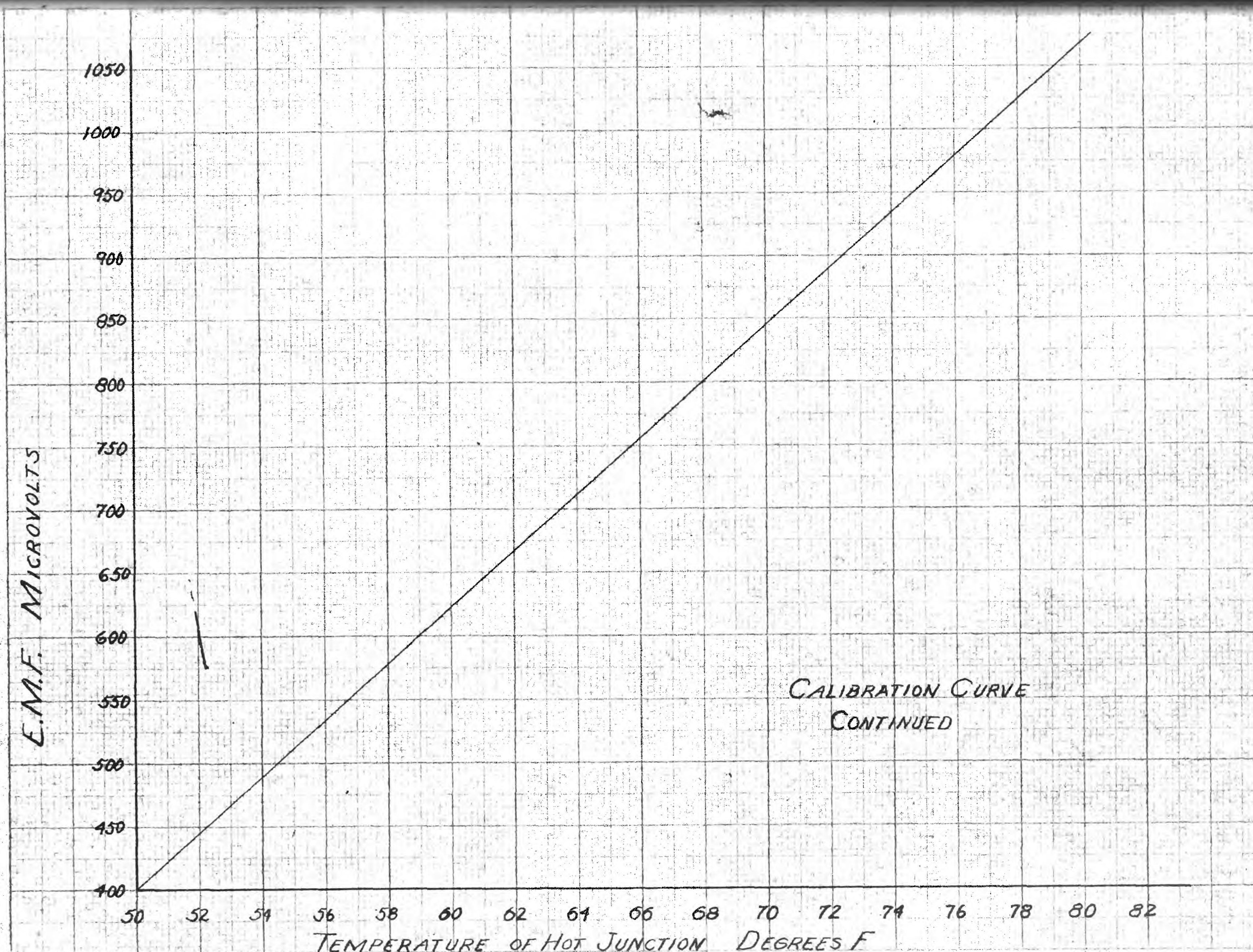
52
TEMPERATURE OF HOT JUNCTION DEGREES F
CALIBRATION CURVE
COPPER-CONSTANTAN THERMOCOUPLE
COLD JUNCTION AT 32°F


E.M.F. Microvolts
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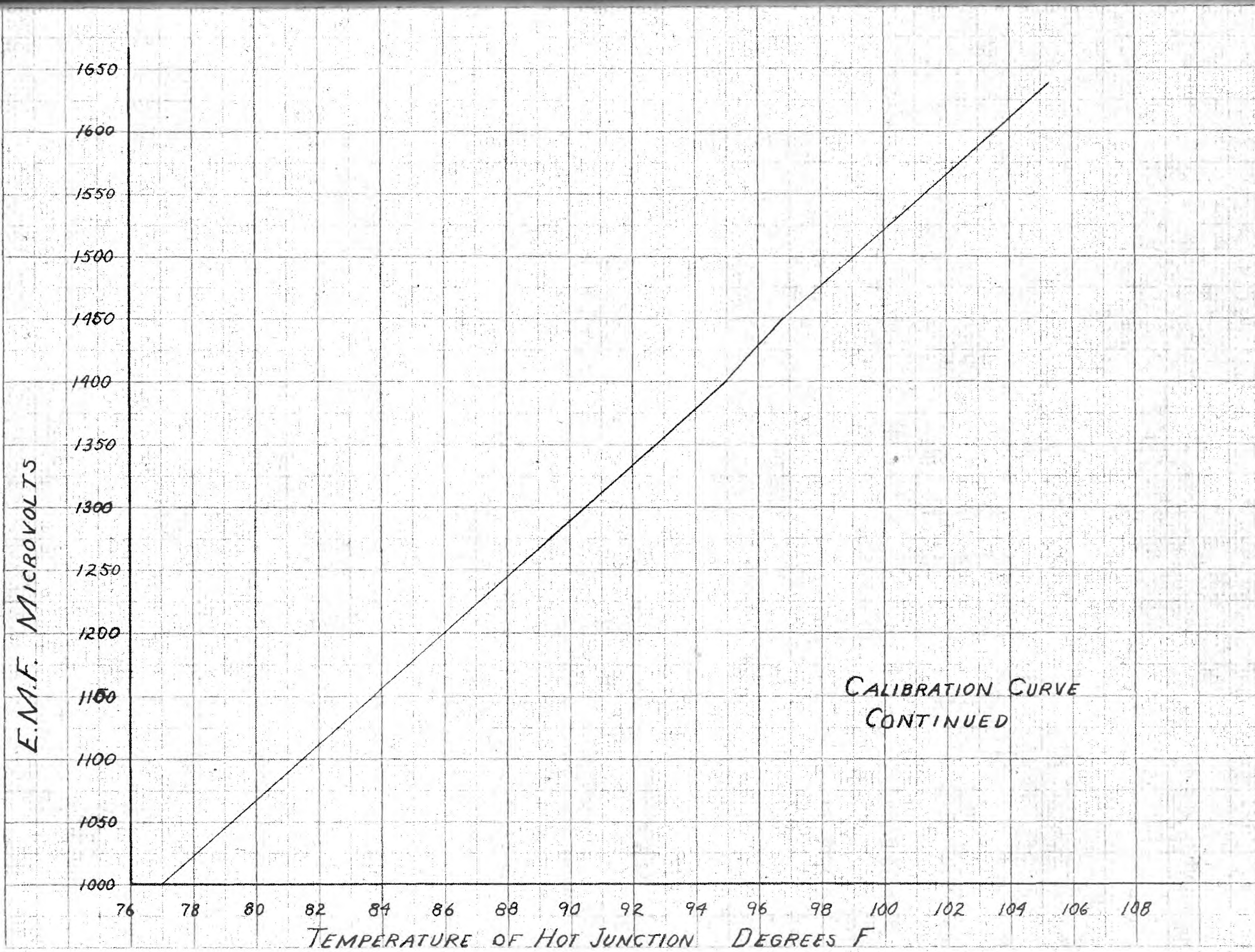
Temperature of Hot Junction Degrees F
Calibration Curve
Continued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E.M.F. MICROVOLTS
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108
TEMPERATURE OF HOT JUNCTION DEGREES F
CALIBRATION CURVE
CONTINUED

COIL DATA

Size of coil $12\frac{1}{4}$" x $12\frac{1}{4}$" x 7"

Fins

Spacing = 7/16"

27 Fins 7" x $12\frac{1}{4}$"

1 Fin $9\frac{3}{4}$" x $12\frac{1}{4}$"

Thickness of fins = .028"

28 Holes .5" Diameter.

Tubes

4 rows of 7 each $12\frac{1}{4}$" long

.627 outside diameter

.525 inside diameter

Free Frontal Area

Total	12.25 x 12.25	=	150.0625 sq. in.
Tubes	7 x 12.25 x .627	=	53.7653 sq. in.
Fins	28 x 12.25 x .028"	=	9.6040 sq. in.

Free Area = Total - Tube Area - Fin Area

Free Area = 86.69 sq. in. = .602 sq. ft.

Cooling Surface Area.

Total Fins	28 x 12.25 x .028	=	9.6040
	27 x 7 x 12.25	=	2315.2500
Holes	28 x 28 x .5 = 392.0		
Net Fin Area	= 2056.12 x 2	=	4112.24
Tubes	28 x 12.25 x .627	=	675.64

Total Area = Net Fin Area + Tube Area

= 4787.88 sq. in. = 33.25 sq. ft.

SIMPLIFICATION OF FORMULAS

AIR CALCULATION BY INTAKE ORIFICE

$$\#air/min. = 1096.2\ C.A.\sqrt{hd}$$

C = Coefficient of discharge for orifice

A = Area of orifice in square feet.

h = Head in inches of water.

d = Density of air at orifice.

Coefficient of discharge for both orifices used is .6 as given by Whitfield in his thesis on Intake Orifice Method of Measuring Air.

<u>15.25" Orifice.</u>

$$\#air/min. = 1096.2\ CA\sqrt{hd}$$

$$C = .6$$

$$A = 1.27 \text{ sq. ft.}$$

$$\#air/min. = 835\sqrt{hd}$$

<u>10.20" Orifice.</u>

$$\#air/min. = 1096.2\ CA\sqrt{hd}$$

$$C = .6$$

$$A = .567 \text{ sq. ft.}$$

$$\#air/min. = 1096.2 \times .6 \times .567\sqrt{hd}$$

$$\#air/min. = 373\sqrt{hd}$$

REASON FOR CONSTANT AIR QUANTITIES WITH SLIGHTLY VARYING DENSITIES.

Maximum air temperature at orifice is:

T - Dry = $90^{o}F$

T - Wet = $70.5^{o}F$

Volume from chart = 14.1 at 14.7 #/□" press.

$$14.1 \times \frac{14.7}{14.3} = 14.5$$

$$\text{Density} = \frac{1}{14.5} = .069$$

Substituting in orifice formula as simplified:

$$\#\text{air/min.} = 835\sqrt{hd}$$

$$= 835\sqrt{.168 \times .069}$$

$$= 90 \ \#/\text{min.}$$

Thus we have a maximum error of 2 #/min. in 92 #/min. This gives an error of 2.28% and was the balance of our data does not warrant a greater degree of accuracy, our assumption is justified

AMMONIA HEAT BALANCE CHECK.

As shown by calculations in the results for the quantity of air by the intake orifice, a value of 92# of air per minute for Run 1, Series 1A is found.

Below are calculations showing the ammonia heat balance check:

NH_3 BALANCE CHECK

$$\#NH_3\ (\triangle H) = \#air\ (\triangle H)$$

Heat of air at T_W of 51.7^oF = 21.3 BTU/#

Heat of air at T_W of 46.4^oF = 18.3 BTU/#

H of air = 3.0 BTU/#

Heat of NH_3 liquid at 65^o = 114.8 BTU/#

Heat of superheat of NH_3 at 14^o and 39.35 #/□"

abs. = 617.1

$$\#NH_3\ (\triangle H) = \#air\ (\triangle H)$$

$$\#air/min. = \frac{\#NH_3/30\ (\triangle H)}{air\ (\triangle H)\ x\ 30}$$

$$= \frac{16.5\ (617.1 - 114.8)}{3\ x\ 30}$$

$$= \frac{16.5\ x\ 502.3}{90}$$

$$\#air/min. = 91.9$$

Thus our orifice value of air flow is correct.

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11. E. Ower - Measurement of Air Flow, Chapman and Hall, Ltd., London, 1933.

12. Leeds and Northrup Standard Conversion Tables. Adams, Bull, A.I.M.M.E., 159 P. 2111, 1919.

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