1989

Overall heat transfer coefficients, pressure losses, and correlations for serrated finned tubes /

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By

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June 1 1949

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Bethlehem, Pennsylvania

June, 1949

This thesis is presented to the faculty of Lehigh University in partial fulfillment of the requirements for the degree of Master of Science in Chemical Engineering.

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Acknowledgment

This project was sponsored by David E. Kennedy, Inc., manufacturers of "Tilco-Fin" tubing, and was carried out under the supervision of Dr. Darrel E. Mack. The author wishes to thank Dr. Mack for the many invaluable suggestions which he has made. The author also wishes to express his appreciation to Mr. George Lynn, for his assistance in construction of the experimental apparatus, and to Messrs. Fiedler, Mountsier, Vondersmith, Cullen, Parseghian, Beard and Brown for their aid in obtaining data and in calculating results.

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Summary

This report is divided into four sections presenting the following information: (1) overall heat transfer coefficient and pressure drop data for a three-row serrated-type finned tube heat exchange unit (Tilco Unit 587½-1.5-3-45) when used in air refrigeration; (2) overall heat transfer coefficient and pressure loss data for three-row units of 107 7/8 and 107 1/8 serrated tubing and for an Aerofin-type, helically-wound, 5/8 inch-tubing unit, all used for air heating; (3) a correlation for all heating data; and (4) correlations for pressure loss data.

This project represents a continuation of the work of Foust, Hinterleiter, and McKinley, and the correlations presented in this report make use of data obtained by these investigators.

Part I - Air Refrigeration

All experimental data presented are for unit 587½-1.5-3-45. Tests were made in compliance with specifications of the American Society of Refrigeration Engineers for testing tempering coils in the refrigeration range. Since these specifications require that the air coming to the coils be at a temperature of 38 degrees Fahrenheit and have a relative humidity of 85 per cent, it is necessary to recirculate completely the cold air through a return duct, and to humidify the air before it returns to the coils. Runs were made by turning on air blower,

refrigeration, and humidifier sprayer to cool down the apparatus and to bring the air temperature and humidity to the desired conditions; turning the refrigeration off for six minutes to defrost the coils; then turning the refrigeration on again to begin the run. The air velocity was held constant throughout the 50 minute duration of the run. A hook gage manometer was read to determine the pressure drop. The temperature and humidity before and after passing over the coils, Freon temperature and orifice manometer were read to calculate the air flow rate and overall coefficient.

Plots of temperature and relative humidity of the air before and after passing through the coils were made for use in calculation. Points used in calculation were taken from smooth curves drawn through the raw data. This smoothing of the data was necessary because temporary fluctuations of the temperature and humidity resulted in erratic calculated values of the overall coefficient. Plots of overall heat transfer coefficient and pressure drop against time are shown.

It was found that both the overall heat transfer coefficient and pressure drop are functions of time. The pressure drop increases with time, while the overall coefficient increases for a short time while the tubes become filled with Freon, then drops off to some value near six B.T.u./hr./sq. ft/^{o}F. where it remains constant or nearly constant. After about 40 minutes the overall coefficient starts to decrease again. The coefficient of heat transfer is seen to be nearly independent of air rate, of air inlet temperature, and air inlet humidity.

Part II - Air Heating Tests

Heat transfer coefficient data in the air heating range are presented for Tilco-Fin Units 107 7/8-1.875-3-35 and 107 1/8-2.125-3-32, and Aerofin-type Unit 5/8-7-1.5-3-43.

Runs were made by admitting steam to the inside of the tubes and passing air across the outside of the tubes at a measured rate. The quantity of heat exchanged was determined by measuring the change in air temperature. An overall coefficient was calculated from this quantity, the heat transfer area, and the log mean temperature difference. Plots of the overall heat transfer coefficient versus the mass rate of air flow were obtained for each unit.

The overall heat transfer coefficients for one inch tubes were found to be independent of fin length, number of rows, and total number of tubes, and to be about 30 per cent less than the coefficients for five-eighths inch tubes. The overall heat transfer coefficients for five-eighths inch Aerofin-type tubes were found to be about 35 per cent less than for five-eighths inch Tilco-Fin tubes.

Pressure drop data are presented for Tilco-Fin Units 107 7/8-1.875-3-35 and 107 1/8-2.125-3-32 and for the Aerofin-type Unit 5/8-7-1.5-3-43. The pressure drop through duct and unit was measured by means of the hook gage manometer, and the corrected pressure drop was obtained by subtracting from the measured pressure drop, the pressure loss through the empty duct at a corresponding air rate. Tests were made isothermally at 25 degrees Centigrade, and also with the air being heated as it passed through the unit.

The isothermal pressure drop was found to be lower than when the air was being heated. Pressure drops for 107 7/8 and 107 1/8 tubing were nearly the same. The pressure drops for the Aerofin-type Unit 3/8-7-1.5-3-43 are less than those for the similar Tilco-Fin Unit 587½-1.5-3-43 at values of G lower than about 3500 pounds per hour per sq. ft. of net free area, and are greater at values of G above 3500.

Part III - Heat Transfer Correlations

It was found that all heat transfer data may be correlated by the expressions:

$$\left(\frac{h}{CpG}\right)\left(\frac{Cp\,\mu_f}{K}\right)^{2/3} = 0.927\ Re^{-0.554}$$

and

$$\left(\frac{hD}{K}\right)\left(\frac{K}{C\mu}\right)^{0.4} = 0.826\ Re^{0.487}$$

These correlations hold for one, two, and three row units of 587½ tubing in air heating and air cooling, for 587 3/4 tubing on 1.5 inch centers (fins overlapping) in air heating, and for one inch units 107 3/8-2.375-1-9, 107 7/8-1.875-3-35 and 107 1/8-2.125-3-32 in air heating.

Part IV - Pressure Drop Correlations

The Gunter-Shaw pressure drop correlation,

$$\frac{f}{2} = \frac{\Delta P \; g \; D_V}{G^2 \; L} \rho \left(\frac{\mu}{\mu_w}\right)^{0.14} \left(\frac{D_V}{S_T}\right)^{-0.4} \left(\frac{S_L}{S_T}\right)^{-0.6}$$

was found to be unsatisfactory, and the correlation,

$$f = \frac{\Delta P \; g \; D}{N \; V^2}$$

is proposed instead. While not wholly satisfactory, the latter correlation brings together all pressure drop data (except those for Unit 107 3/8-2.375-1-9), with an average deviation of about 10 per cent. The difficulty in correlating pressure drop data is attributed to the great effect which baffling, a factor not included in the correlation, has on the pressure drop.

Introduction

The purposes of this report are multifold. It was desired to determine the overall heat transfer coefficient and pressure losses of 5/8 inch serrated-type finned tube heat exchangers when used in refrigeration applications, previous investigators having determined the performance of 5/8 inch units in the air heating and air conditioning ranges. It was desired to determine the effect of variation of tube size and fin length on the overall heat transfer coefficient and pressure drop; data were desired to compare serrated-type fins with helically-wound fins; and heat transfer and pressure drop correlations for all fin and tube sizes were desired.

In order to determine refrigeration characteristics, tests were carried out on Tilco-Fin Unit 507 5/8-1.5-3-45. The specifications of the American Society of Refrigeration Engineers for testing refrigeration coils were met as closely as possible. It was found that unlike the case for air heating or cooling, equilibrium conditions are never attained, and the overall heat transfer coefficient is a function of time as well as of the mass rate of air flow.

In order to determine the effect of fin size on heat transfer characteristics, tests were performed on Unit 107 7/8-1.875-3-35 and Unit 107 1/8-2/125-3-32. These two units are similar except for the difference in fin length, the fin length of the first being seven-sixteenths inches, and of the second, nine-sixteenth inches.

Test on the one-inch units also serve to provide data for comparison with those previously obtained for five-eighths inch prime tube size. This comparison shows the effect of varying prime tube diameter.

Tests were made on an Aerofin-type helically-wound fin-tube unit in order to compare this unit with a [illegible]-type serrated fin unit having the same size, number and geometry of tubes.

It was desired to correlate all heat transfer data by means of one of the standard forms for heat transfer coefficients correlation employing various dimensionless groups.

Similarly it was desired to try the Gunter-Shaw pressure drop correlation for the data, and if this correlation failed to correlate the data, to discover some other means of correlation.

In order to prevent this report from assuming mammoth proportions, the data presented are restricted to sample data for each type of experimental data and each correlation. Complete data are presented in the author's three previous progress reports (9, 10, 11). All data obtained are represented graphically in this report.

Heat Transfer Unit Nomenclature

In the reports of Foust, Hinterleiter and McKinley, various heat exchange units were arbitrarily assigned unit numbers by each investigator. Confusion resulted from different investigators calling the same unit by different numbers or letters and different units by the same numbers or letters. To eliminate such ambiguity, a system of unit designation was devised and is followed in this report. This system has the desirable qualities of exclusive designation for each unit, complete description of the unit, and universality of application.

According to this system a unit is described by a series of numbers. The particular finned tubes of which the unit is composed is designated according to the system of the Extended Surface Division of David E. Kennedy Inc., manufacturers of the tube (15). The first two numbers indicate the outside diameter of the prime tube. The third number designates the fin pitch per inch, the fourth number is used to indicate the overall outside diameter over the fins. Thus 587½ tube indicates a tube having a 5/8 inch O.D., with fins spaced seven to the inch, and with the overall diameter over the fins <u>one</u> and one-half inches.

In order to designate the arrangement of the tubes in the units, another series of numbers is employed. The first number indicates the distance between tube centers.

The tubes are spaced with their centers on equilateral triangles. The second number designates the number of rows of tubes, and the third number designates the total number of tubes in the unit.

All the units have approximately the same overall face dimensions. The tubes are always two feet long and the number of tubes per row is enough so that the row is approximately two feet. By this system, then, all units are designated unabmiguously, for there will be only one arrangement of the tubes possible.

As an example consider the unit 587 1/2-1.5-3-44. The tubes are as described above, spaced on centers one and one-half inches apart. There are three rows of tubes, forty-four tubes in all. It is obvious that the arrangement of the tubes must be: fifteen in the first row, fourteen in the second, fifteen in the third, since this is the only possible arrangement of forty-four tubes spaced on equilateral triangles in three rows.

Part I - Air Refrigeration Tests

Description of Apparatus

The test unit, constructed by Foust and modified by Hinterleiter and McKinley is described in Part II of this report as it was used in the air heating and air cooling tests made previously. Considerable modification of the unit was required for the air refrigeration tests.

In order that the air coming to the coils be at a temperature of 38 degrees Fahrenheit, in accordance with the specifications of the American Society of Refrigeration Engineers for testing refrigeration coils, it is necessary to recirculate the cooled air. To accomplish this recirculation, a return duct was constructed. This duct is made of galvanized iron, ten inches square, and is located above the original unit. The return duct returns the air from the blower to the cooling coils. Three dampers are provided in the return duct to vary the amount of recirculation, and thus to control the temperature and humidity of the recirculated air. The first damper opens to the atmosphere directly above the blower. The second is located in the duct itself, five and one-half feet from the first damper and one-half foot before the third damper, which, like the first, opens to the atmosphere. With this arrangement, air from the room may be bled into the system, and the degree of recirculation may be varied from 0 to 100 per cent. Throughout this series of experiments complete recirculation was used.

At a point midway between the ends of the return duct, a humidifier sprayer is provided. This humidifier serves to increase the relative humidity and to decrease the temperature of the air as it is returned to the coils. The humidifier consists of an atomizer-type sprayer. Compressed air at approximately five pounds gage pressure is used to spray ice water into the air stream. A large reservoir above the sprayer admits water into the sprayer chamber as the water from it is sprayed into the air stream.

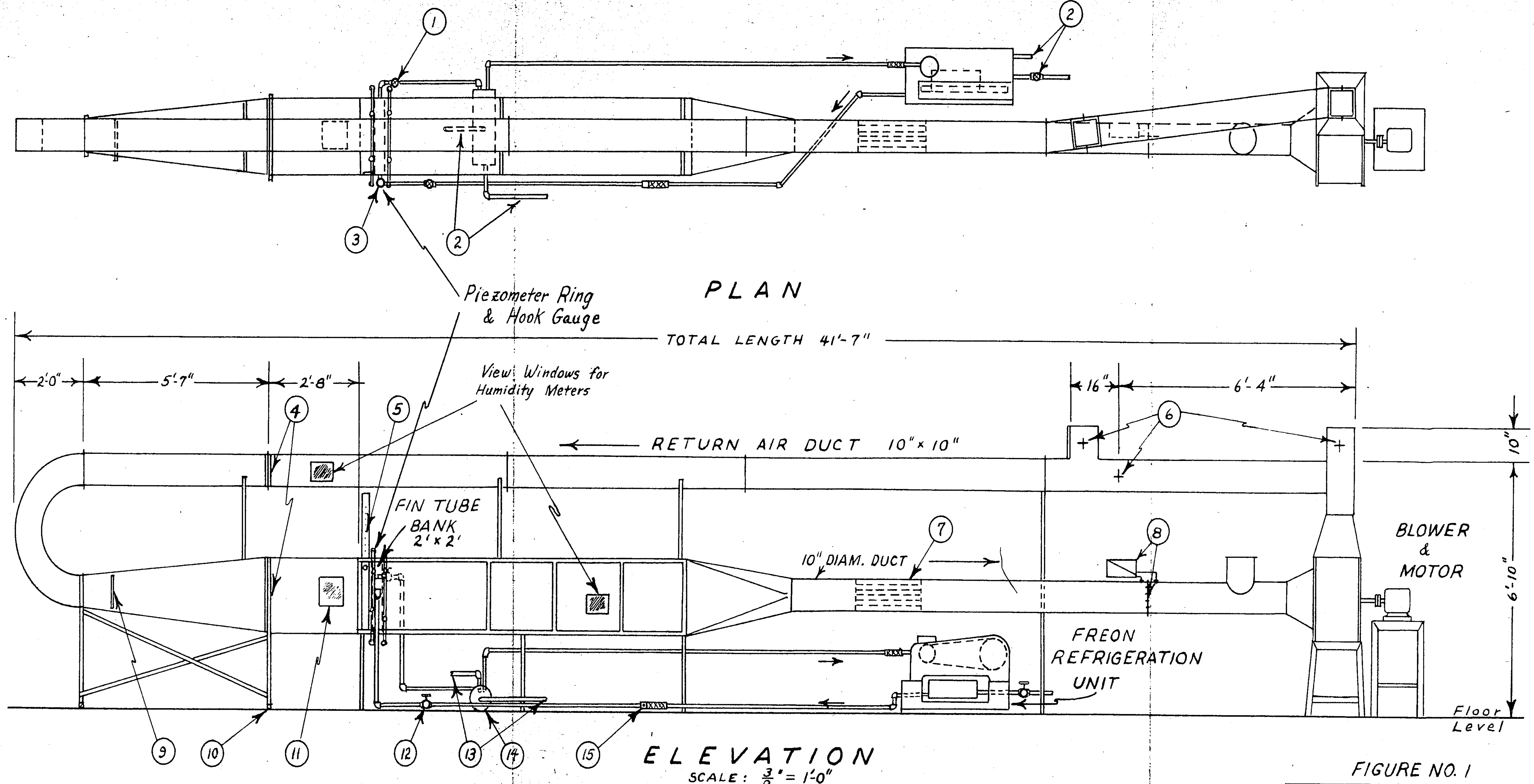
A bend of two feet radius at the end of the return duct, followed by an expansion section, delivers the air to the coils. A system of screens is provided in the expansion section to distribute the air uniformly across the face of the coil. Data for face air velocities are presented in a Chemistry 99 report of Fiedler, Mountsier and Vondersmith (1).

With one side of the unit no longer open to the atmosphere, it was necessary to add a second piezometer ring to the unit in order to measure the pressure drop across the coils. The piezometer rings are connected to the legs of a hook gage manometer. The hook is raised and lowered by means of a threaded wheel, calibrated to read directly in thousandths of inches of water pressure difference.

Two thermometers and two humidity meters are provided to measure the temperature and humidity of the air before and after passing over the coils. A window immediately in front of the coil allows observation of the icing of the coils and reading of the inlet thermometer and humidity meter. A similar window, located after the mixing section, enables reading of the outlet temperature and humidity.

In order to attain the desired low temperature, the apparatus was covered with two layers of asbestos paper. As this did not provide sufficient insulation, a layer of one-quarter inch thick cork was added, and all exposed cold metal surfaces were covered with a black plastic insulating compound to prevent condensation of moisture on these surfaces.

A diagram of the apparatus is to be found in Figure 1.



1	FREON VALVE	4	RUBBER GASKETS	7	STRAIGHTENING TUBES	10	CASTORS	13	COOLING WATER AUX. EVAPORATOR
2	COOLING WATER FOR CONDENSERS	5	SLIDING STRIP FOR AIR VELOCITY MEAS.	8	MANOMETER & ORIFICE	11	4 WINDOWS 9" x 9"	14	AUX. EVAPORATOR
3	EXPANSION VALVE	6	3 DAMPERS 10" x 10"	9	AIR DIFFUSION SCREEN	12	FREON VALVE	15	SIGHT GLASS & STRAINER

FIGURE NO. 1

FIN TUBE AIR COOLER

LEHIGH UNIVERSITY
BETHLEHEM, PENNA.
CHEM. 99

DRAWN BY: F. VONDERSMITH
MARCH 19, 1948

Calibration of Instruments

The thermometers were all calibrated against a Bureau of Standards Thermometer. Calibration data for the thermometers have been presented in a previous report (12).

The humidity meters were calibrated by placing them in a dessicator along with various solutions which maintain known, constant humidities. Saturated aqueous solutions of various substances in contact with an excess of a definite solid phase at given temperatures maintain constant humidity in an enclosed space. Solutions of five different salts were employed giving readings over a range of values of relative humidities from 31 to 92 per cent. The results are given in a previous report (10). Since the humidity meters gave such close agreement with the true humidity, especially through the range of relative humidities involved in this experiment, the meter readings were accepted as read.

The effect of the air stream on the meters was found to be negligible. Meter readings are independent of direction of air flow at the air velocities employed in the runs. It was discovered that sudden changes in humidity or temperature cause the various meters to give widely divergent readings, but in the experiments the humidity and temperature both changed slowly, and little difficulty was experienced from that source.

Test Procedure

It was desired to follow, insofar as possible, the specifications of the American Society of Refrigeration Engineers for testing refrigeration units. These specifications require that the air coming to the coils be at a temperature of 38 degrees Fahrenheit and have a relative humidity of 85 per cent, and that the unit be run with refrigeration on for forty minutes followed by a defrosting period of 20 minutes. It was impossible to maintain the air at 38 degrees Fahrenheit for a 20-minute defrosting period, and it was found that the coils defrosted in less than six minutes with the temperature allowed to rise as the coils were defrosted. Therefore, runs were made by turning the refrigeration on to cool the air and apparatus below 38 degrees Fahrenheit, turning the refrigeration off for six minutes to defrost the coils, and making a run of 50 minutes duration with refrigeration on.

To make a run, a zero reading of the hook gage manometer was obtained with the blower off, and the air blower and refrigeration compressor were turned on. The air rate was set at the desired value by varying the setting of the sliding damper above the blower. The reservoir of the humidifier sprayer was filled with ice water, and the air compressor was turned on to spray water into the air stream. When the temperature of the air coming to the coils dropped to 38 degrees Fahrenheit and the relative humidity increased to 85 per cent, the refrigeration was turned off while the air blower was kept on to defrost the coils. The humidifier was turned off, since defrosting the coils increased the relative humidity of the recirculated air to some value greater than 85 per cent.

At the end of six minutes time the coils were no longer covered with frost and had stopped dripping moisture; the refrigeration was turned on and the run begun. The humidifier was turned on again to maintain the relative humidity at 85 per cent. Data were taken at two-minute intervals for the first 18 minutes, at three-minute intervals for the next 12 minutes, and at four-minute intervals until the end of the run.

The following quantities were recorded: Air temperature before passing over the cooling coils (Temp. In); Relative humidity before passing over the cooling coils (Rel. Hum. In); Air temperature after passing over the cooling coils (Temp. Out); Relative humidity after passing over the cooling coils (Rel. Hum. Out); Freon temperature (T_F); Hook gage pressure drop; and Orifice manometer differential head (h_w).

Sample Calculations

Air Flow Rate:

For the orifice employed (2):

$$W = 20200\, k \sqrt{p\ \ h_w}$$

where W = pounds of air per hour

k = a dimensionless orifice coefficient dependent on the Reynold's Number, Re.

p = air density, pounds per cubic foot.

h_w= pressure drop across orifice, inches of water.

From a plot of the square root of air density against temperature the square root of air density is obtained at the temperature of the air passing through the orifice, and from a plot of k against h_w, k is obtained at the same temperature at the value of h_w in question (2).

For Run R-3

$$W = 20200\, k \sqrt{\rho\, h_w}$$

$$W = 20200\ (0.729)\ (0.2318)\ \sqrt{2.18}$$

$$W = 6120 \text{ pounds per hour}$$

The mass rate of air flow, G, is defined as the weight rate of flow per unit of free area.

$$G = \frac{W}{A}$$

where G = mass rate of air flow, pounds of air per hour per square foot of net free area.

A = net free area for air passage through the heat exchange unit

For Run R-3:

$$G = \frac{W}{A}$$

$$G = \frac{6120}{1.952} = 3135 \text{ pounds per hour per sq. ft. net free area}$$

<u>Overall Heat Transfer Coefficient:</u>

In calculating the overall heat transfer coefficient by using the data as read, it was found that a plot of the overall coefficients against time, while showing a definite trend of the coefficient to drop off slightly with time, was quite erratic and bumpy. This lack of smoothness in the curve is caused by slight errors in reading thermometers or humidity meters or by momentary fluctuations in temperature or humidity, possibly the result of temporary variation in humidifier sprayer pressure. As a means of improving the results, the raw data were corrected for these temporary variations.

The data were corrected by plotting, against time, the inlet and outlet air temperatures and the inlet and outlet relative humidities. A smooth curve was drawn through the points. Points where the temperature momentarily increased greatly and points where the humidity momentarily decreased greatly were disregarded since they were caused by occasional short-lived malfunctioning of the humidifier sprayer. Plots of temperature and humidities against time for Run R-3 are shown in Figures 2 and 3.

From plots of temperature and humidities against time, new data were taken, and these data were employed in calculating the overall heat transfer coefficient.

All Centigrade temperature readings were converted to Fahrenheit.

In order to calculate the amount of moisture condensed out of the air, it is necessary to determine the absolute humidity of the air before and after passing over the coils. Since psychrometric charts do not yield values sufficiently accurate for use here, it is necessary to determine the moisture content of the air in another manner.

At temperatures below 40 degrees Fahrenheit when the relative humidity is 70 per cent or higher, the per cent relative humidity differs from the per cent absolute humidity by less than three tenths of one per cent, and therefore:

$$H_{abs.} = \mu H_{Sat.} \cong \phi H_{Sat.}$$

where $H_{abs.}$ = the absolute humidity of the air, grains of H_2O per pound of dry air.

$H_{Sat.}$ = the absolute humidity the air would have at the given temperature if it were saturated.

μ = percentage saturation or percentage absolute humidity.

ϕ = per cent relative humidity.

For Run R-3 at t = 14 minutes:

$$H_{Abs.} = \phi H_{Sat.}$$

$H_{Abs.}$ = (0.79) (33.73) = 26.65 grains H_2O per pound of dry air.

The absolute humidity of the air at saturation at a particular temperature may be obtained from generally available tables of saturated moisture contents of air at various temperatures. The table in Jennings and Lewis, "Air Conditioning and Refrigeration", was used in these calculations (7). It is necessary to interpolate from the table since humidities are given only at every degree of temperature. The differences are essentially constant, and linear interpolation was employed to determine the saturation humidities for fractions of degrees.

Quantity of heat transferred:

Air:

The amount of heat involved in cooling the air is given by:

$$Q_{air} = W\ C_p\ \Delta T$$

where Q_{air} = the amount of heat transferred by cooling the air, B.t.u. per hour.

W = pounds of air flowing through the coils per hour.

C_p = mean specific heat of dry air over the temperature interval, B.t.u. per pound per degree Fahrenheit.

ΔT = difference in air temperature, inlet and outlet, degrees Fahrenheit.

For Run R-3 at t = 14 minutes:

$$Q_{air} = W\ C_p\ \Delta T$$

Q_{air} = (6120) (0.2375) (6.45) = 9360 B.t.u. per hour.

Water:

It is assumed that all the water condensed out remains on the fins, is frozen and cooled to the temperature of the Freon inside the tubes.

The heat transferred from the Freon to the water is the sum of the heats involved in: 1. cooling water vapor from the inlet temperature to the temperature at which it condenses or to the temperature at which it leaves the coils; 2. Condensing water vapor; 3. Cooling liquid water from the temperature at which it was condensed to the temperature at which it freezes; 4. Freezing liquid water; 5. Cooling solid ice from the temperature at which it was frozen to the Freon temperature. Quantities 2. and 4. are by far the most important and are of the order of thousands of B.t.u. per hour. The sum of the quantities 1., 3., and 5. are of the order of less than two hundred B.t.u. per hour. To simplify the calculations, these latter quantities were calculated once for each run, rounded off to the nearest hundred B.t.u. and added to quantities 2. and 4. as a constant in determining the total amount of heat transferred from water to Freon at any particular time. Since the amount of heat exchanged is dependent only on the initial and final states and is independent of the path, it may be assumed without error that all the water that is condensed condenses at 32 degrees Fahrenheit, the temperature at which it freezes. Thus quantity 3. is eliminated in the calculations. Quantity 1. becomes:

$$Q_{H_2O}\ \text{vapor} = \frac{H_{\text{Abs. in.}}}{7000}\ W\ (C_p)\ (T\text{in} - 32)$$

where Q_{H_2O} vapor = the heat involved in cooling water vapor from the inlet temperature to the temperature at which it condenses out or to the temperature at which it leaves the coil, B.t.u. per hour.

$\frac{H_{Abs.\ in}}{7000}\ W$ = quantity of moisture in the inlet air, pounds per hour.

C_p = mean specific heat of water vapor over the temperature interval, B.t.u. per pound per degree Fahrenheit.

$T_{in} - 32$ = the difference between the temperature of the water vapor in and the temperature at which it condenses, for that which condenses; and is approximately equal to the temperature difference between water vapor in and water vapor out, for that which does not condense.

For Run R-3 at t = 14 minutes:

$$Q_{H_2O}\ \text{vapor} = \frac{26.65\ (6120)}{7000}\ (0.45)\ (38.10 - 32)$$

$$Q_{H_2O}\ \text{vapor} = 64\ \text{B.t.u. per hour}$$

Quantity 5. is given by:

$$Q_{H_2O}\ \text{solid} = \frac{(\Delta H)\ W}{7000}\ (C_p)\ (32 - T_F)$$

where Q_{H_2O} solid = the amount of heat involved in cooling solid ice from the temperature at which it freezes to the temperature of the liquid Freon inside the coils.

$\Delta H = H_{Abs.\ in.} - H_{Abs.\ out}$ = quantity of air condensed, grains per hour.

C_p = mean specific heat of solid ice over the temperature interval.

T_F = Freon temperature.

For Run R-3 at t = 14 minutes:

$$Q_{H_2O}\ \text{solid} = \frac{4.49}{7000}\ (6120)\ (0.48)\ (32 - 13.3)$$

$$Q_{H_2O}\ \text{solid} = 35\ \text{B.t.u. per hour.}$$

Q_{H_2O} vapor + Q_{H_2O} solid = 64 + 35 = 100 B.t.u. per hour.

Quantities 2. and 4. are given by:

$$Q_{H_2O} \text{ cond.} = \frac{(\Delta H)\ (W)}{7000}\ (L_v)$$

$$Q_{H_2O} \text{ freez.} = \frac{(\Delta H)\ (W)}{7000}\ (L_f)$$

where L_v and L_f are the latent heats of vaporization and fusion respectively, both at 32 degrees Fahrenheit, B.t.u. per pound.

The total heat exchanged, Freon to water, then becomes:

$$Q_{H_2O} = Q_{H_2O} \text{ cond.} + Q_{H_2O} \text{ freez.} + 100$$

$$Q_{H_2O} = \frac{(\Delta H)\ W}{7000}\ (L_v) + \frac{(\Delta H)\ W}{7000}\ (L_f) + 100$$

$$Q_{H_2O} = \frac{(\Delta H)\ (W)}{7000}\ (L_v + L_f) + 100$$

For Run R-3 at t = 14 minutes:

$$Q_{H_2O} = \frac{(4.49)\ (6120)}{7000}\ (1072 + 143.2) + 100$$

$$Q_{H_2O} = 4880 \text{ B.t.u. per hour.}$$

The total amount of heat exchanged is the sum of the heats involved in the water and in the air:

$$Q_{total} = Q_{air} + Q_{H_2O}$$

For Run R-3 at t = 14 minutes:

$$Q_{total} = 9360 + 4880 = 14240 \text{ B.t.u. per hour}$$

Log mean temperature difference:

The log mean temperature difference, Freon to air, is given by:

$$\Delta T_{mean} = \frac{(T_{in} - T_F)\ (T_{out} - T_F)}{\ln \dfrac{T_{in} - T_F}{T_{out} - T_F}}$$

For Run R-3 at $t = 14$ minutes:

$$\Delta T_{mean} = \frac{(38.12 - 13.3) - (31.69 - 13.3)}{\ln \frac{38.12-13.3}{31.69-13.3}}$$

The overall heat transfer coefficient is then calculated from:

$$U = \frac{Q_{total}}{A\ (\Delta T_{mean})}$$

where U = the overall heat transfer coefficient, B.t.u. per hour per square foot per degree Fahrenheit.

A = heat transfer area, square feet.

For Run R-3 at $t = 14$ minutes:

$$U = \frac{14240}{(113.6)\ (21.40)}$$

$U = 5.87$ B.t.u. per hour per square foot per degree Fahrenheit.

Pressure Drop:

The pressure loss data are read from the hook gage manometer as the pressure drop across the coils in thousandths of inches of water and are plotted against time without further calculation.

Table 7

Original Data

Run R-3

Time	In: Rel. Hum. %	In: Temp. °C		Out: Rel. Hum. %	Out: Temp. °C	Freon Temp. °C	Hook Gage in H_2O	h_w in H_2O
2:54	Off							
3:00	On							
3:02	83	8.0		92	3.8	+1.5	64×10^{-3}	2.18
3:04	75	6.4		89	2.8	-9.8	78 "	
3:06	74	5.6		85	1.0	-10.0	82 "	
3:08	75	4.5	(sprayer	84	0.4	-10.2	86 "	
3:10	70	4.9	off)	83	0.1	-10.3	93 "	
3:12	78	3.7		85	0.0	-10.3	112 "	
3:14	80	3.4		85.5	-0.1	-10.4	116 "	
3:17	81	3.2		86	-0.2	"	146 "	
3:20	84	3.1		87	-0.2	"	164 "	
3:23	84	3.2		88	-0.1	"	197 "	
3:26	83	3.5		88	0.0	"	228 "	
3:30	85	3.4		88	+0.2	"	269 "	
3:34	82	4.0		88	+0.5	"	328 "	
3:38	84	3.9		88	+1.0	"	386 "	
3:43	84	4.2		88	+1.4	"	455 "	
3:46	84	4.5		88	+1.7	"	512 "	
3:50	84	4.7		88	+2.0	"	585 "	

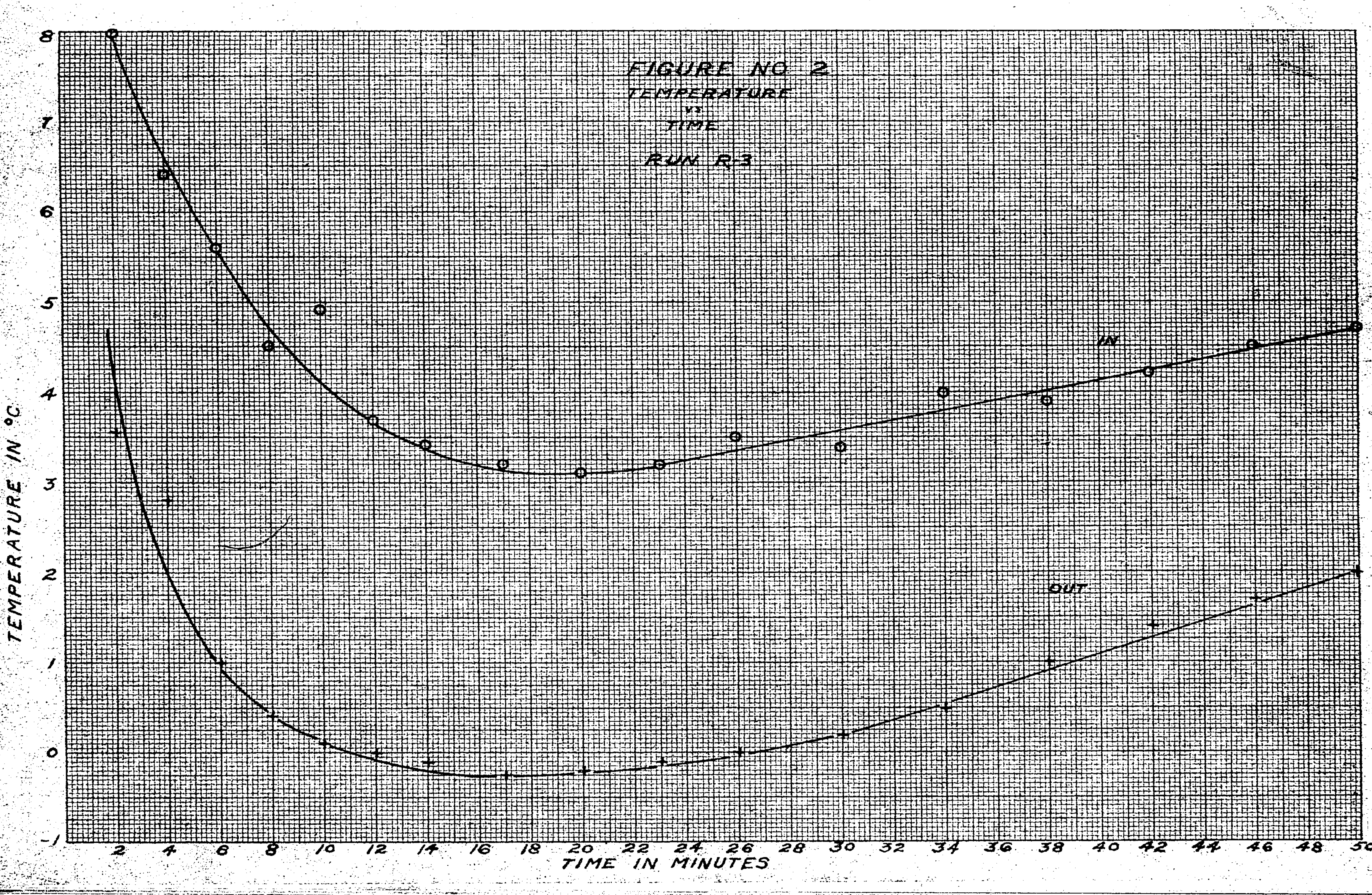


FIGURE NO 2
TEMPERATURE
VS
TIME
RUN R-3
IN
OUT
TEMPERATURE IN °C
TIME IN MINUTES
8
7
6
5
4
3
2
1
0
-1
2
4
6
8
10
12
14
16
18
20
22
24
26
28
30
32
34
36
38
40
42
44
46
48
50

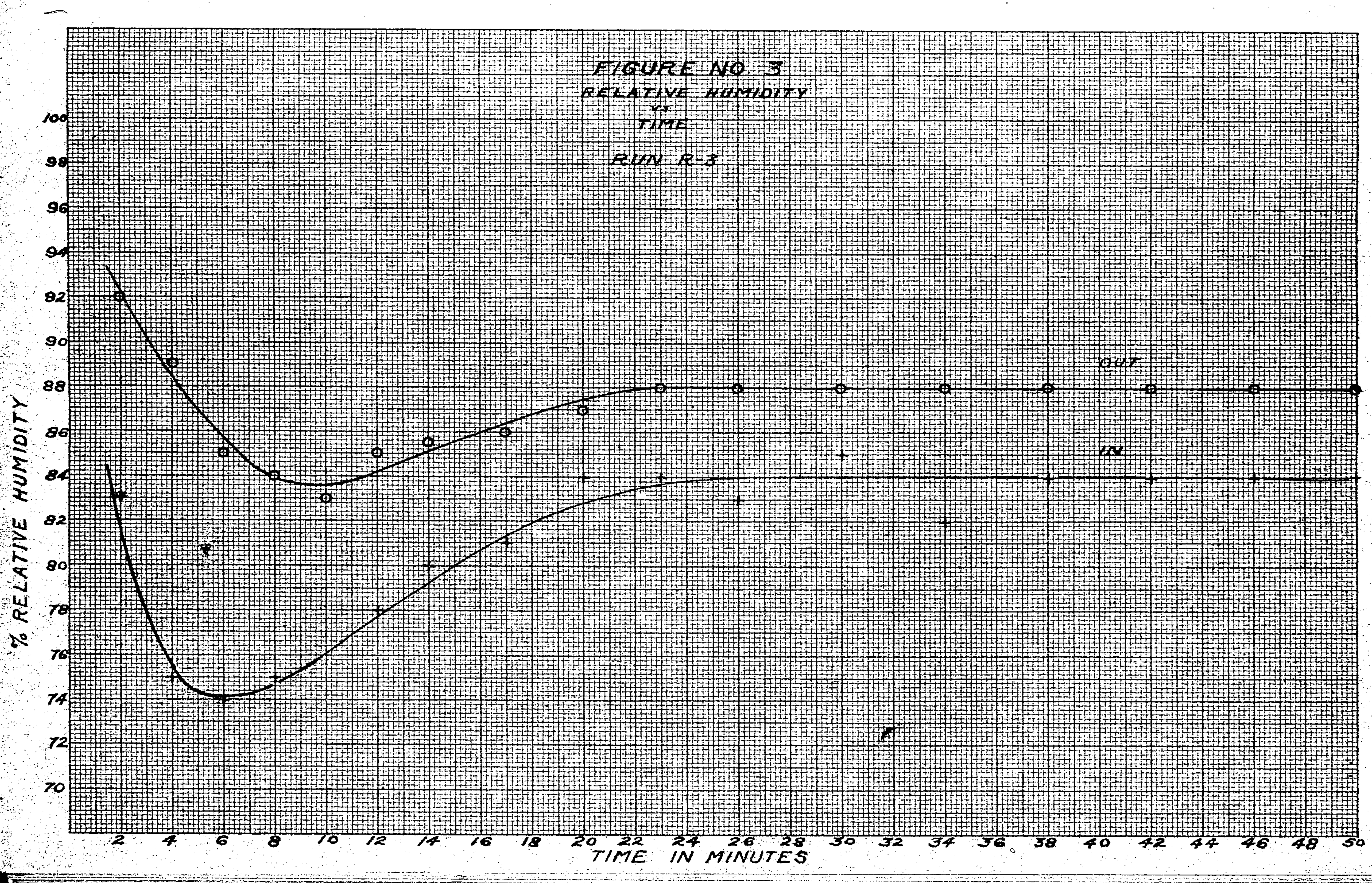


FIGURE NO. 3
RELATIVE HUMIDITY
vs
TIME
RUN R-3
OUT
IN
% RELATIVE HUMIDITY
TIME IN MINUTES
100
98
96
94
92
90
88
86
84
82
80
78
76
74
72
70
2
4
6
8
10
12
14
16
18
20
22
24
26
28
30
32
34
36
38
40
42
44
46
48
50

Table II

Corrected Data Run- R-3

	IN				OUT						
Time in Min.	Temp. °F.	Rel. Hum. %	Sat. Hum. Grains H_2O Lb. dry air	Abs. Hum. Grains H_2O Lb. dry air	Temp. °F.	Rel. Hum. %	Sat. Hum. Grains H_2O Lb. dry air	Abs. Hum. Grains H_2O Lb. dry air	Freon Temp. °F.	h_w ins. H_2O	Hook Gage Ins. H_2O
2	46.40	81.5	46.59	38.00	39.38	92.20	35.51	32.72	34.70	2.18	$64x10^{-3}$
4	43.97	75.5	42.43	31.90	35.60	88.40	30.57	27.00	14.36	"	$78x10^{-3}$
6	42.08	74.2	39.44	29.25	33.80	85.60	28.43	24.35	14.00	"	$84x10^{-3}$
8	40.55	74.7	37.15	27.75	32.81	83.90	27.31	22.91	13.54	"	$90x10^{-3}$
10	39.37	76.1	35.40	27.70	32.25	83.60	26.68	22.30	13.46	"	$98x10^{-3}$
12	38.66	77.6	34.51	26.75	31.91	84.10	26.29	22.10	13.46	"	$114x10^{-3}$
14	38.12	79.0	33.78	26.65	31.69	85.20	26.03	22.16	13.30	"	$119x10^{-3}$
18	37.61	81.9	33.12	27.10	31.59	86.80	25.91	22.48	"	"	$150x10^{-3}$
22	37.67	83.4	33.19	27.62	31.73	87.90	26.08	22.92	"	"	$188x10^{-3}$
26	38.07	84.0	33.73	28.32	32.00	88.00	26.40	23.22	"	"	$228x10^{-3}$
30	38.48	84.0	34.27	28,80	32.45	88.00	26.90	23.65	"	"	$275x10^{-3}$
34	38.85	84.0	34.77	29.20	33.01	88.00	27.53	24.20	"	"	$324x10^{-3}$
38	39.29	84.0	35.37	29.65	33.71	88.00	28.33	24.92	"	"	$388x10^{-3}$
42	39.65	84.0	35.87	30.15	34.34	88.00	29.06	25.55	"	"	$452x10^{-3}$
46	40.10	84.0	36.50	30.62	35.01	88.00	29.84	26.26	"	"	$518x10^{-3}$
50	40.46	84.0	37.02	31.05	35.69	88.00	30.69	27.00	"	"	$584x10^{-3}$

Table III

Oct. 18, 1948 Calculated Data Run - R-3

Ave. Temp. In = 39.2°F.
Ave. Rel. Hum. In = 82%
G = 3135 lb/hr/sq.ft.

Time in min.	T °F.	H Grains H_2O / Lb. dry air	Q_{air} Btu/Hr	Q_{H_2O} Btu/Hr	Q_{total} Btu/Hr	T_{mean} °F	A sq.ft.	G Lb/Hr (sq.ft. net free area)	U Btu/Hr (sq.ft) (°F)
2	7.02	5.28	10200	5660	15860	29.70	113.6	3135	4.75
4	8.37	4.90	12170	5270	17440	26.02	"	"	5.95
6	8.28	4.90	12080	5270	17350	24.60	"	"	6.29
8	7.74	4.84	11240	5210	16450	22.10	"	"	6.63
10	7.12	5.40	10340	5800	16140	22.62	"	"	6.33
12	6.75	4.65	9800	5010	14810	22.20	"	"	5.93
14	6.43	4.49	9360	4880	14240	21.20	"	"	5.87
18	6.02	4.62	8750	4980	13730	21.10	"	"	5.82
22	5.94	4.70	8620	5060	13680	21.15	"	"	5.77
26	6.07	5.10	8820	5490	14310	21.62	"	"	5.90
30	6.03	5.15	8750	5540	14290	22.00	"	"	5.78
34	5.84	5.00	8420	5380	13850	22.45	"	"	5.50
38	5.58	4.73	8100	5100	13200	23.10	"	"	5.10
42	5.31	4.60	7710	4960	12670	23.55	"	"	4.80
46	5.09	4.36	7390	4710	12100	24.10	"	"	4.48
50	4.77	4.05	6940	4380	11320	24.70	"	"	4.11

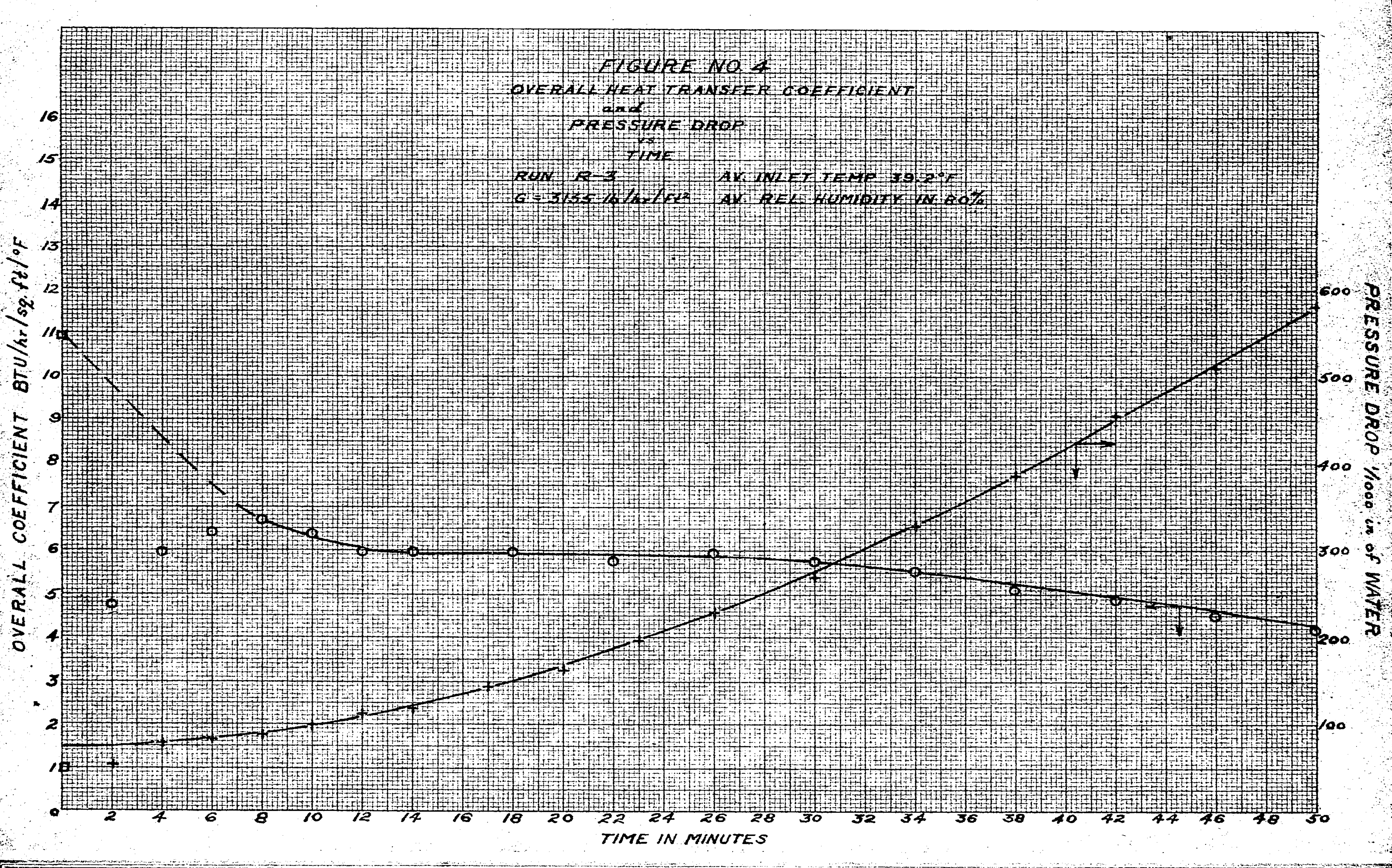


FIGURE NO. 4
OVERALL HEAT TRANSFER COEFFICIENT
and
PRESSURE DROP
vs
TIME
RUN R-3
AV. INLET TEMP 39.2°F
G = 3135 lb/hr/ft²
AV. REL. HUMIDITY IN 80%
OVERALL COEFFICIENT BTU/hr/sq. ft/°F
PRESSURE DROP 1/1000 in of WATER
TIME IN MINUTES

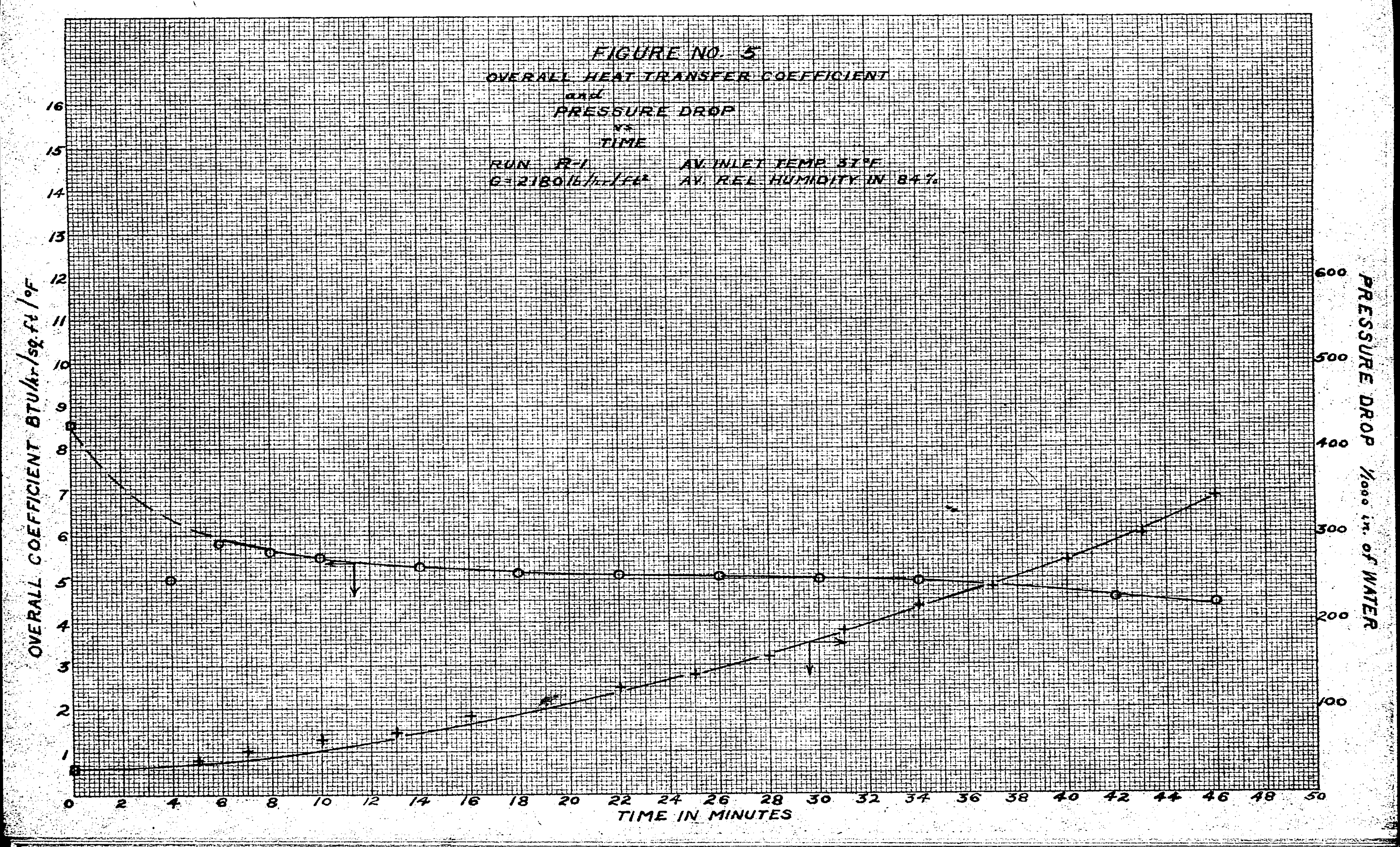


FIGURE NO. 5
OVERALL HEAT TRANSFER COEFFICIENT
and
PRESSURE DROP
vs
TIME
RUN R-1
AV. INLET TEMP. 37°F
G=2180 lb/hr/ft²
AV. REL. HUMIDITY IN 84%
OVERALL COEFFICIENT BTU/hr/sq.ft/°F
PRESSURE DROP 1/1000 in. of WATER
TIME IN MINUTES
0
2
4
6
8
10
12
14
16
18
20
22
24
26
28
30
32
34
36
38
40
42
44
46
48
50
1
2
3
4
5
6
7
8
9
10
11
12
13
14
15
16
100
200
300
400
500
600

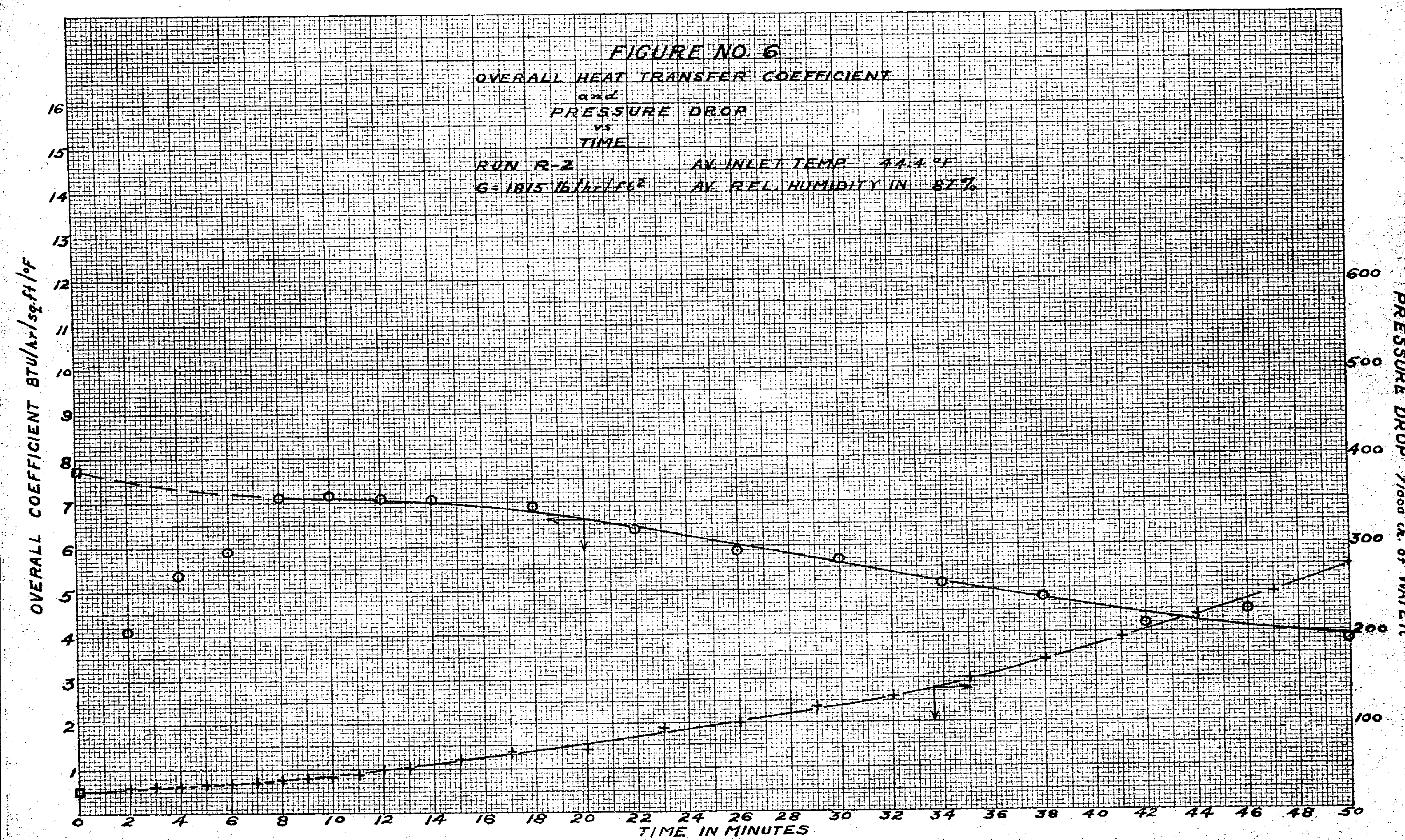


FIGURE NO. 6
OVERALL HEAT TRANSFER COEFFICIENT
and
PRESSURE DROP
vs
TIME
RUN R-2
AV. INLET TEMP. 44.4°F
G=1815 lb/hr/ft²
AV. REL. HUMIDITY IN 87%
OVERALL COEFFICIENT BTU/hr/sq.ft/°F
PRESSURE DROP 1/1000 in. of WATER
TIME IN MINUTES

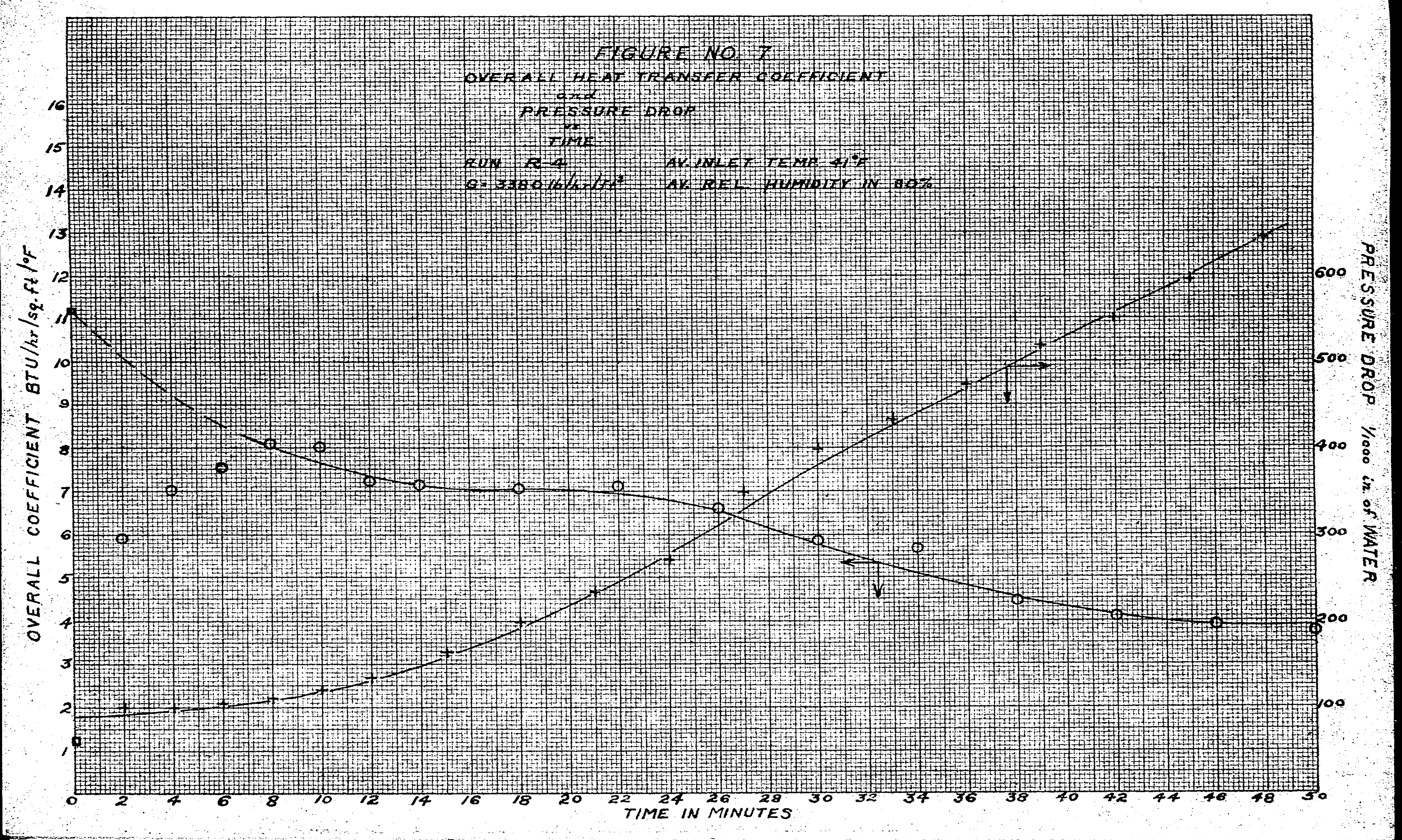


FIGURE NO. 7
OVERALL HEAT TRANSFER COEFFICIENT
and
PRESSURE DROP
vs
TIME
RUN R-4
AV. INLET TEMP. 41°F
G = 3380 lb/hr/ft²
AV. REL. HUMIDITY IN 80%
OVERALL COEFFICIENT BTU/hr/sq.ft/°F
PRESSURE DROP 1/1000 in. of WATER
TIME IN MINUTES

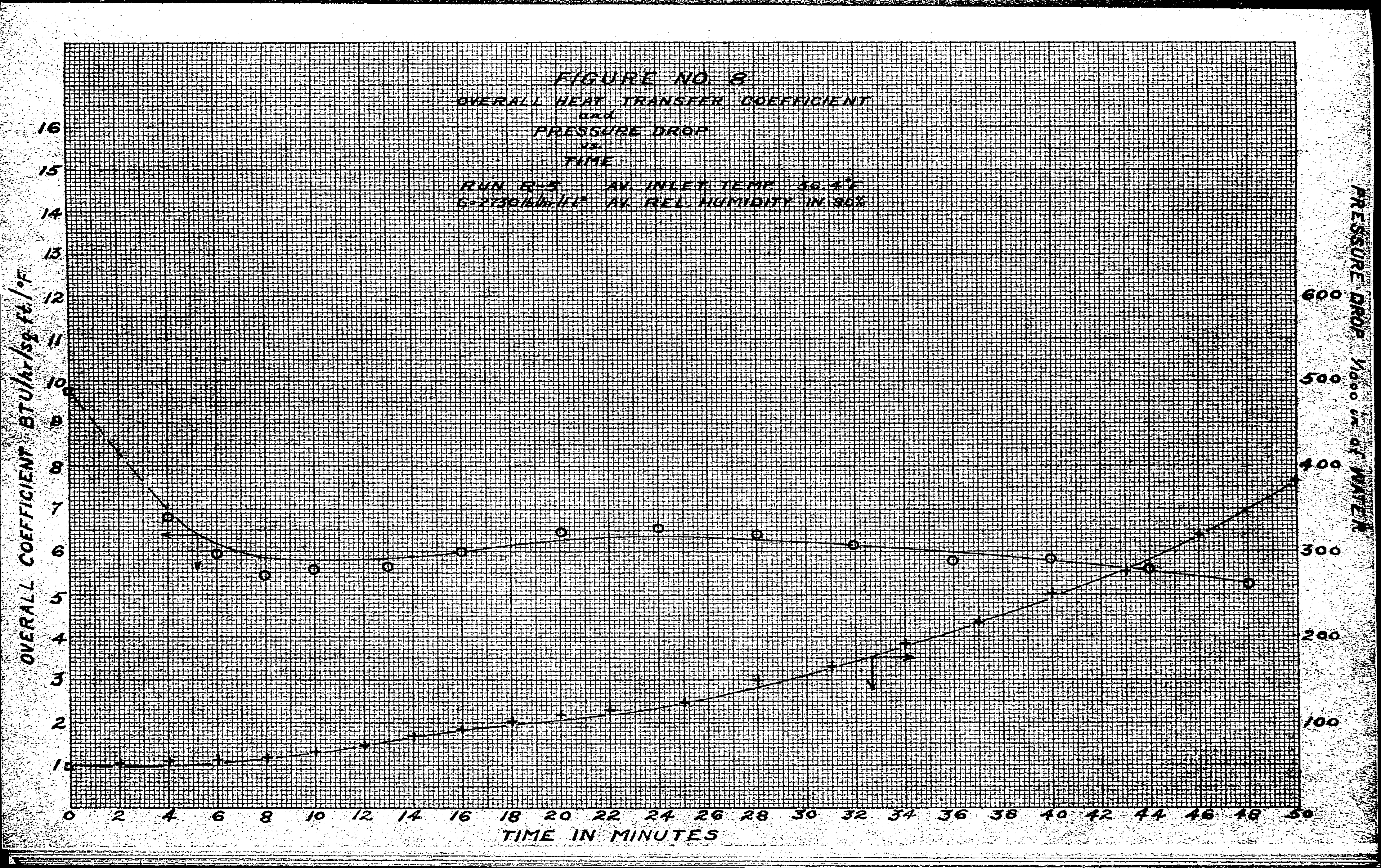


FIGURE NO. 8
OVERALL HEAT TRANSFER COEFFICIENT
and
PRESSURE DROP
vs.
TIME

RUN R-5 AV. INLET TEMP. 56.4°F
G=2730 lb/hr/ft² AV. REL. HUMIDITY IN 96%

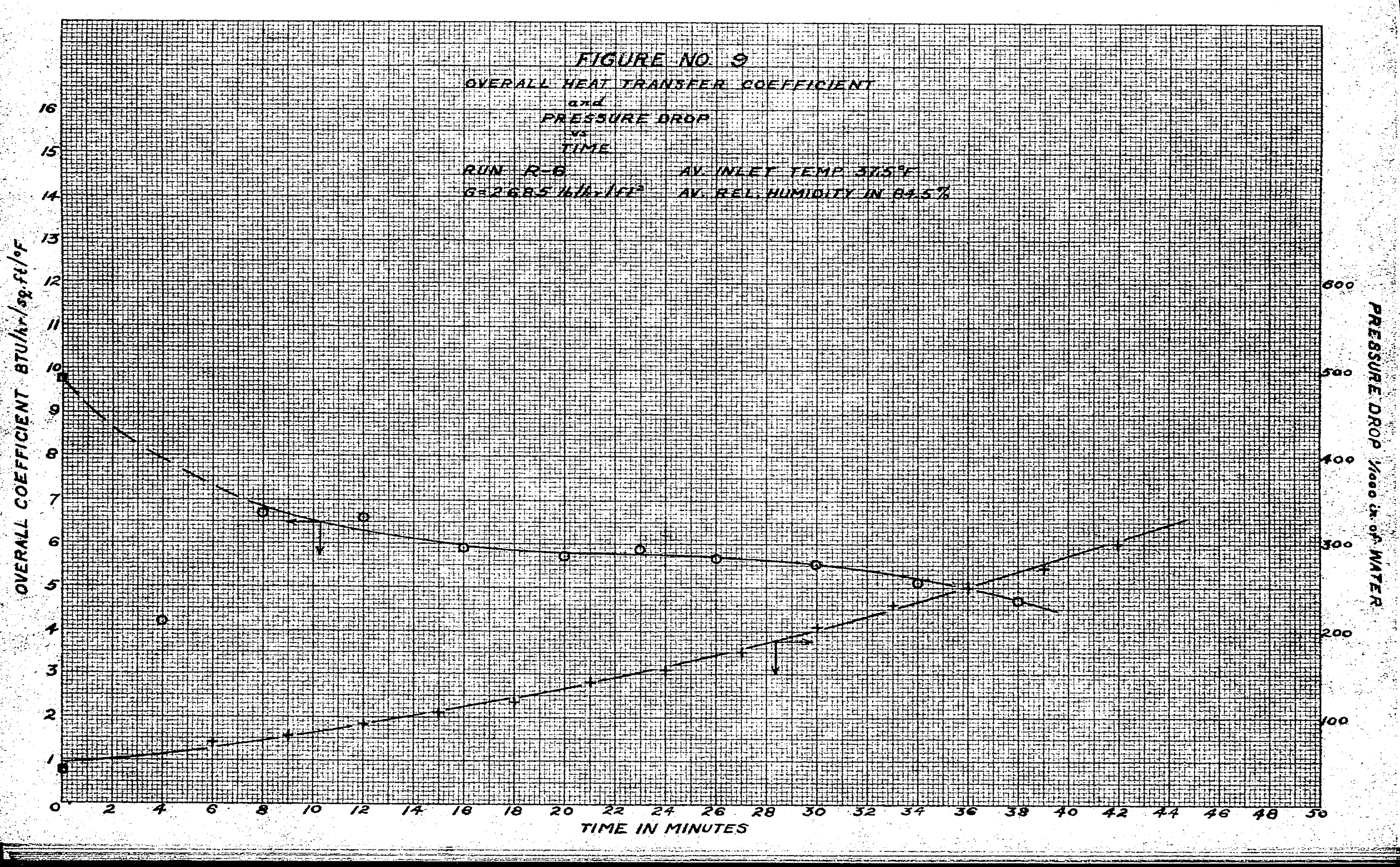


FIGURE NO. 9
OVERALL HEAT TRANSFER COEFFICIENT
and
PRESSURE DROP
vs
TIME
RUN R-6
G=2685 lb/hr/ft²
AV. INLET TEMP 37.5°F
AV. REL. HUMIDITY IN 84.5%
OVERALL COEFFICIENT BTU/hr/sq.ft/°F
PRESSURE DROP 1/1000 in. of WATER
TIME IN MINUTES
0
1
2
3
4
5
6
7
8
9
10
11
12
13
14
15
16
100
200
300
400
500
600
0
2
4
6
8
10
12
14
16
18
20
22
24
26
28
30
32
34
36
38
40
42
44
46
48
50

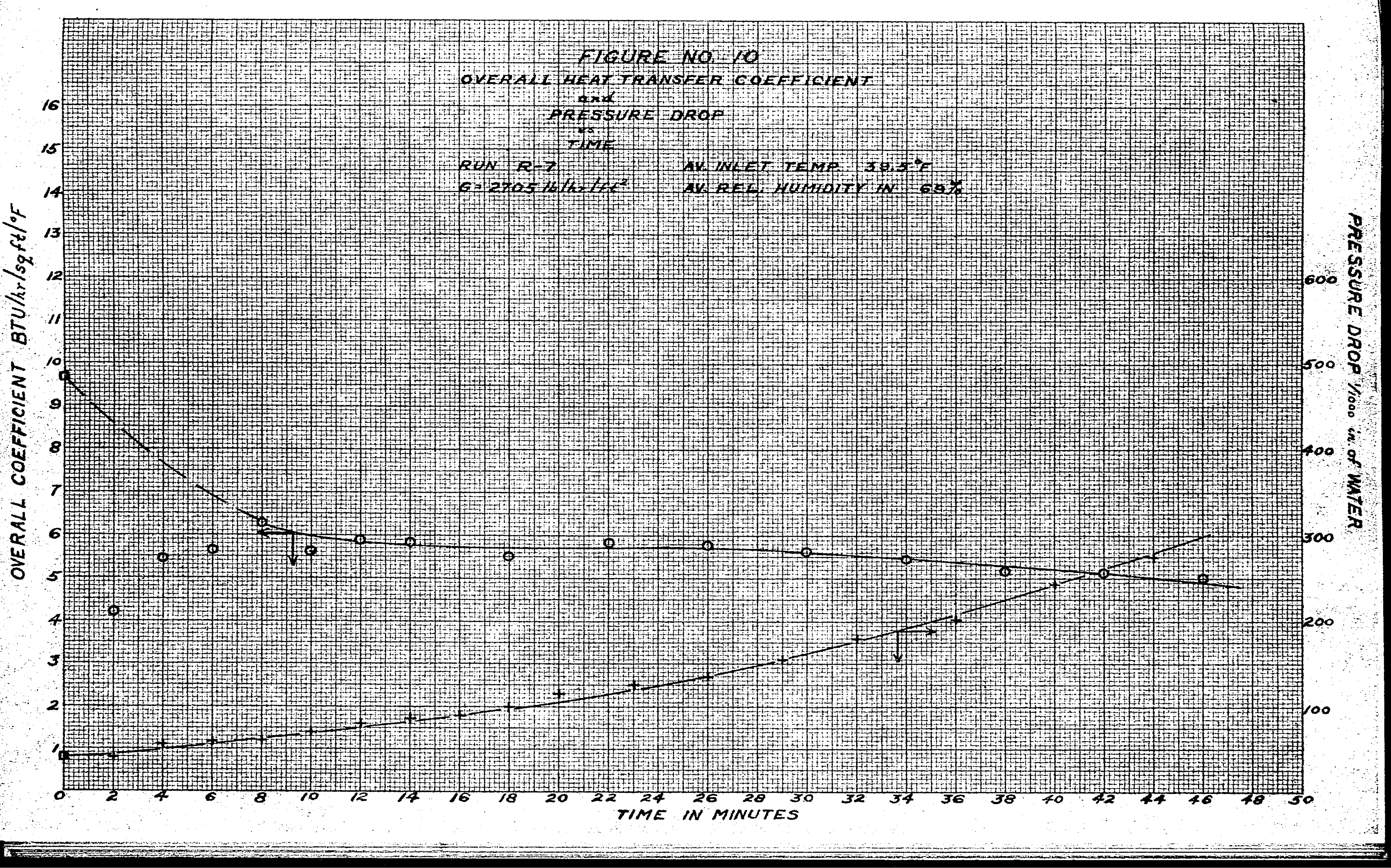


FIGURE NO. 10
OVERALL HEAT TRANSFER COEFFICIENT
and
PRESSURE DROP
vs
TIME
RUN R-7
G = 2705 lb/hr/ft²
AV. INLET TEMP. 39.5°F
AV. REL. HUMIDITY IN 69%
OVERALL COEFFICIENT BTU/hr/sq ft/°F
PRESSURE DROP 1/000 in. of WATER
TIME IN MINUTES

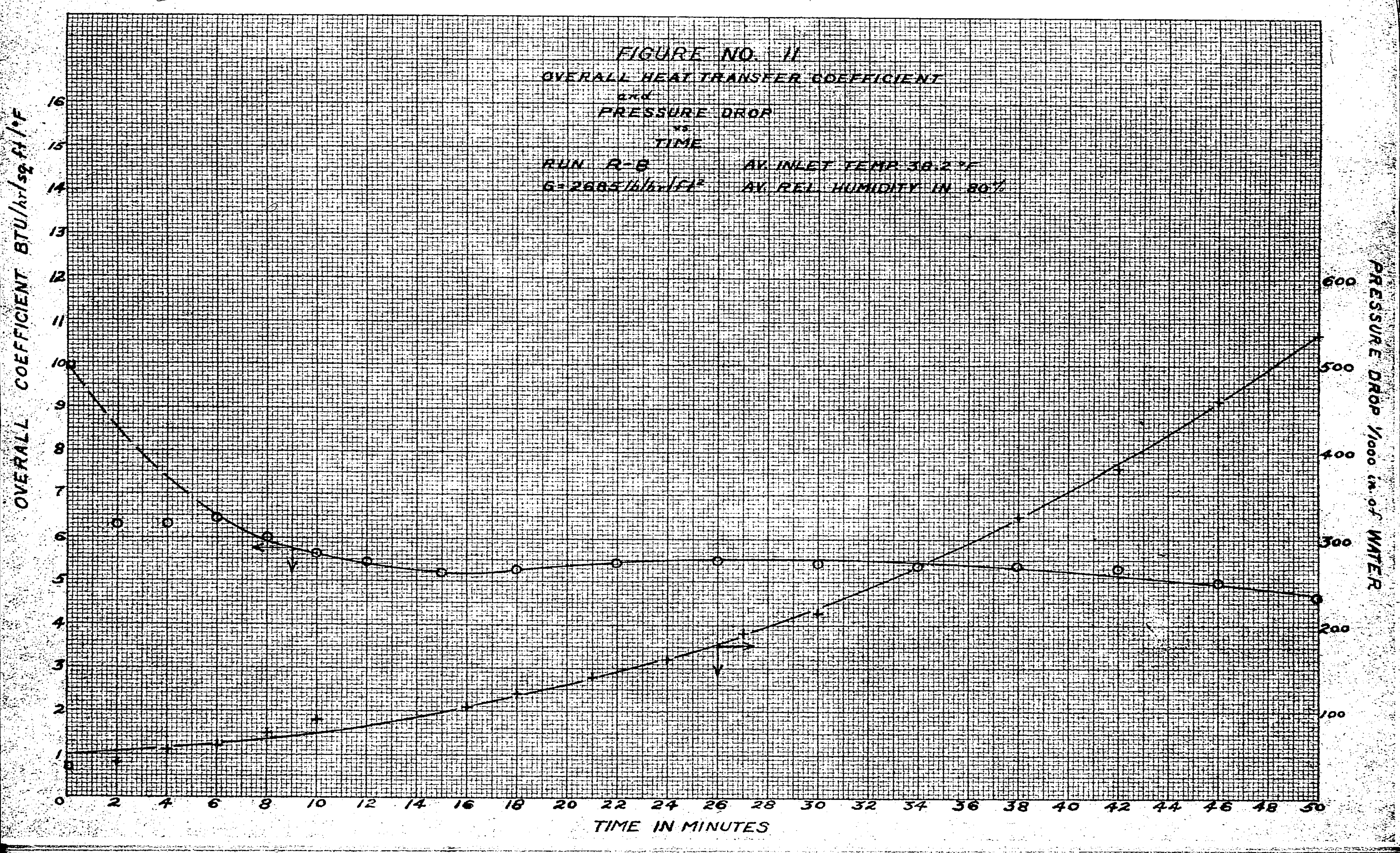


FIGURE NO. 11
OVERALL HEAT TRANSFER COEFFICIENT
and
PRESSURE DROP
vs
TIME
RUN R-8
AV. INLET TEMP. 38.2 °F
G=2685 lb/hr/ft²
AV. REL. HUMIDITY IN 80%
OVERALL COEFFICIENT BTU/hr/sq.ft/°F
PRESSURE DROP 1/1000 in. of WATER
TIME IN MINUTES
0
2
4
6
8
10
12
14
16
18
20
22
24
26
28
30
32
34
36
38
40
42
44
46
48
50
1
2
3
4
5
6
7
8
9
10
11
12
13
14
15
16
100
200
300
400
500
600

Discussion of Results

A comparison of the eight curves of overall heat transfer coefficient against time reveals that for all values of the mass rate of air flow used, the curves are of the same general shape. The effective overall coefficient is low at first because the tubes are not filled with liquid Freon, but as soon as the tubes become filled with liquid Freon the coefficient reaches a higher value, but immediately begins to drop off as the fins frost up. The coefficient drops off to some value between 5 and 7 B.t.u./hr./sq.ft./^{o}F. and remains essentially constant. At approximately forty minutes, the coefficient begins to drop off again.

All the curves extrapolate well to the values at zero time. These values were obtained from air conditioning data for the same unit at the same mass rate of air flow. Before there is any frosting of the tubes the overall coefficient should be the same as in the air conditioning range at the same air rate.

Comparison of figures 8, 9, 10, and 11 shows that the overall coefficient of heat transfer at a particular mass rate of flow is substantially independent of inlet temperature and humidity within the ranges of inlet temperatures from 36.4 to 39.5^{o}F. and inlet relative humidities of 68 to 85%.

A comparison of all eight curves of overall coefficients against time indicates that the overall coefficient in air refrigeration is essentially independent of mass rate of air flow within the ranges of 1815 to 3380 pounds per hour per square foot.

The pressure drop is found to be a function of the mass rate of air flow, as expected. At higher flow rates the initial pressure drop is greater and the rate of increase in pressure drop is greater. With the limits of inlet temperatures 36.4 and 39.5°F. and relative humidities between 68 and 85%, no great change in the pressure drop at a particular air rate was noted.

From Figures 8, 9, 10, and 11 it is noted that at a constant air rate, the variation of pressure drop is slight when either the inlet relative humidity or the temperature is changed, within the limits of humidities of 68 to 85% and limits of temperatures of 36.4 and 39.5 degrees Fahrenheit.

Part II - Air Heating Tests

Description of Apparatus

The test unit was designed and constructed by P. G. Foust and modified later by R. W. Hinterleiter and L. J. McKinley. The unit provides for heat exchange between finned tube test units and air passing through the units. Means are provided for varying the air flow rate and for measuring the amount of air flow, the pressure drop across the unit, and the amount of heat transferred.

The overall length of the unit is approximately 38 feet. Air from the room is drawn successively through a heat transfer test unit, a mixing chamber, straightening vanes, and an orifice by a blower which discharged the air back to the room. A diagram of the equipment is to be found in Figure 12. A detailed description of the various component parts follows.

<u>Tunnel</u>:

The tunnel, which is 23.5 inches wide, 24.0 inches high and 13.5 feet long, is constructed of asbestos millboard. The millboard is mounted on a wooden base supported on a steel framework. A wooden approach funnel is attached to the entrance of the tunnel and provides an even flow of air to the heat transfer unit which is located in the tunnel near the air entrance. At the entrance to the tunnel a thermometer, equipped with a metal shield to minimize radiation effects, is provided to measure inlet air temperature. The tunnel also serves to house the mixing chamber which is described in detail below.

<u>Heat Transfer Units:</u>

The heat transfer units are prefabricated interchangeable units of copper Tilco-Fin tubes manufactured by the Extended Surface Division, David E. Kennedy Inc. of Brooklyn, New York. The tubes are mounted in rows. One, two, and three row units are used. A staggered tube arrangement is employed with tube centers spaced on equilateral triangles. In such an arrangement adjacent rows terminate half a tube width apart, and to prevent "edge effects" which lower the overall heat transfer coefficient, baffles at the end of short tube rows are provided as recommended by Jameson (6).

For tests in which the heat transfer units are employed for heating air, the tubes are mounted in a vertical position with steam inside the tubes. For tests in which the units are employed for cooling air, the tubes are mounted in a horizontal position with Freon-12 inside the tubes. Air is blown across the outside of the tubes perpendicular to the tube length.

In those tests in which the heat transfer units are employed for air heating, steam from the lines is run through a strainer, is reduced to five pounds gage pressure by means of a reducing valve, and is passed to a two-inch manifold to which a steam thermometer and pressure gage are attached. From there the steam passes to the top manifolds of the tube banks, then through the tubes to a two inch exhaust manifold leading to a steam trap equipped with a by-pass valve.

In the course of conducting the air heating tests, "cold spots" were discovered in the heat transfer units, and it became necessary to modify the steam system. The present arrangement, therefore, differs slightly from that described above as it was used by the previous investigators. It was found that at high air flow rates, certain of the tubes in the first row were not hot over their entire length, even though live steam was being continuously exhausted. This condition was attributed to insufficient steam admission and/or poor steam distribution in the unit. In order to admit a greater amount of steam to the tubes, a by-pass valve was connected in parallel with the reducing valve. A section of pipe immediately after this valve was jacketed so that cooling water could be passed over the outside of the pipe to remove superheat from the steam. This arrangement provides steam in quantity sufficient to eliminate all cold spots. See Figure 14 for a diagram of the modified steam system.

In those tests in which the heat transfer units are used for cooling air, Freon-12 is supplied to the tubes as a refrigerant. Liquid Freon from the refrigeration unit is admitted to the tubes through a thermal-regulated expansion valve. The amount of liquid Freon passing through the expansion valve is regulated automatically by a control bulb strapped onto the discharge manifold of the unit. The purpose of this bulb is to keep the refrigerant temperature in the tubes constant by allowing a greater or or lesser flow of Freon through the expansion valve. There are five leads from the expansion valve to the tubes to provide for an equal distribution of Freon in the cooling unit. A thermometer well and Freon pressure gage in the outlet manifold of the unit are provided for measuring respectively, the liquid Freon temperature and the suction pressure. The partially vaporized Freon passes from the outlet manifold to an auxiliary evaporator where the Freon is completely vaporized, and then to the compressor of the refrigeration unit.

<u>Piezometer Ring and Hook Gage Manometer:</u>

A piezometer ring with eight pressure taps, two on each tunnel wall, is located about nine inches from the tunnel entrance.

Each tap consists of a four inch square, sheet steel plate with beveled edges. A one-eighth inch hole in the center of each plate is connected by a one-quarter inch pipe to a rubber hose manifold encircling the tunnel section.

It is possible to vary the plate tap position anywhere from flush against the wall of the tunnel to a distance two inches out from the tunnel wall. The pressure drop is independent of plate tap position, (9). Taps are secured flush with the wall.

The manifold, in turn, is joined to one leg of a hook-gage manometer, the other leg of which is open to the atmosphere. The manometer legs are made of two-inch inside diameter glass tubing. The hook is raised and lowered by means of a wheel threaded on the upper part of the hook shaft and calibrated to read in thousandths of an inch.

Mixing Chamber:

In order to thoroughly mix the air coming from the heat transfer units prior to measuring its temperature, a "plate-and-doughnut" type baffle mixer is employed. A seventeen inch square plate located in the middle of the tunnel is followed by a seventeen inch square "doughnut" hole, followed by another plate and another "doughnut." A thermometer located half an inch downstream from the last baffle gives constant temperature readings along a vertical line.

To minimize heat exchange between the air in the mixing chamber and the surroundings, the tunnel is insulated here with an extra layer of asbestos millboard with an air space of three quarters of an inch between the tunnel wall and the outside millboard.

Straightening Vanes:

A sheet-metal reducer connects the tunnel section to the ten-inch outlet pipe line. Two feet downstream from the reducer, a bundle of seven three-inch sheet-metal drain pipes, two feet in length, are mounted inside the ten inch pipe to straighten the air flow in the pipe line section and to reduce the length of straight pipe needed to give maximum accuracy to orifice measurements.

Orifice and Manometer:

A thin plate orifice, 7.5 inches in diameter in a ten inch pipe, installed according to the specifications of the A. S. M. E. Fluid Meters Report, with flange taps, is used to determine the air flow rate (2). These taps are connected to an inclined manometer, the smallest scale division of which is equal to 0.02 inches of water.

Blower:

The blower is a Buffalo blower, type LL-3, class 1, driven by a two-phase, 220 volt, 1720 r.p.m. motor. The blower and motor are mounted on a steel framework with wooden blocks acting as shock absorbers between motor and blower and the steel supports.

Refrigeration Unit:

The refrigeration unit, manufactured by the Typhoon Air Conditioning Company, employs Freon-12 and is rated at five ton capacity. The unit consists of a compressor and a condenser. The compressor is driven by a five horse-power, 220 volt, 60 cycle, two phase General Electric induction motor connected to the compressor by four V-belts. The condenser is water cooled, all copper, and of horizontal shell-and-tube construction with refrigerant inside the tubes. The supply of cooling water to the condenser is controlled by an automatic valve.

The high pressure liquid line is 1/2 inch copper tubing and is connected to the condenser through a flexible vibration hose. A strainer and sight glass are located in the line, and a liquid Freon valve is provided to stop the flow of Freon while replacing heat transfer units. Located after the liquid Freon valve is a thermal-regulated expansion valve which admits the liquid Freon to the tube of the heat exchange unit.

The tubes are connected to a manifold which is, in turn, connected to the 1-1/4 inch copper suction line. A Freon pressure gage measures the suction pressure and a thermometer and well are provided for measuring the liquid Freon temperature. The refrigerant next passes through a Freon valve used when replacing units, then to an auxiliary evaporator. The function of this auxiliary evaporator, which is, in effect, a condenser working in reverse, is to insure that only completely vaporized refrigerant returns to the compressor. The auxiliary evaporator is an all-copper, shell-and-tube type heat exchanger with Freon in the tubes, water in the shell. From the auxiliary evaporator, the suction line returns the refrigerant to the compressor through a flexible vibration hose connection.

The refrigeration apparatus is sketched in Figure 13.

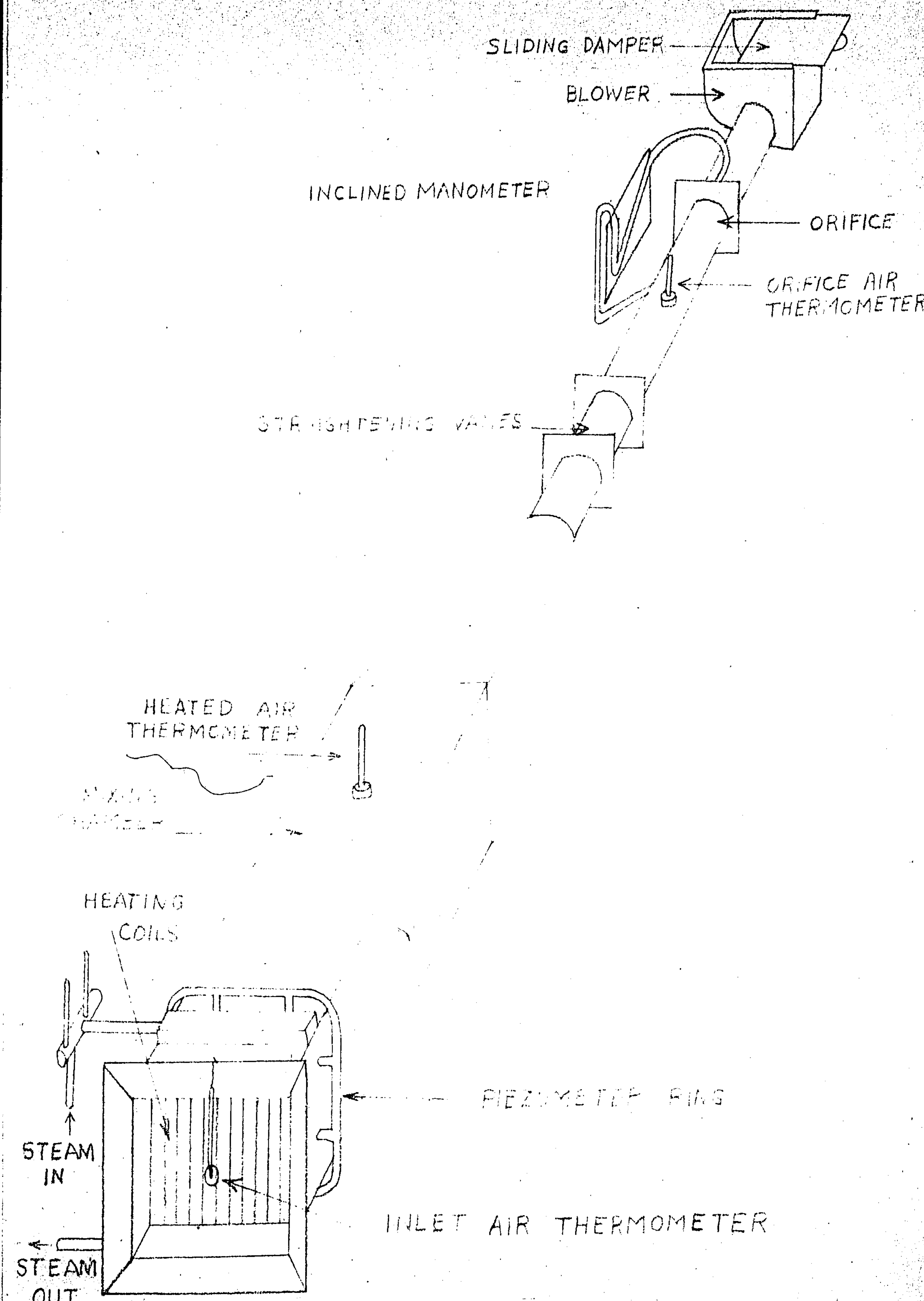


FIGURE 12
SKETCH OF TEST APPARATUS

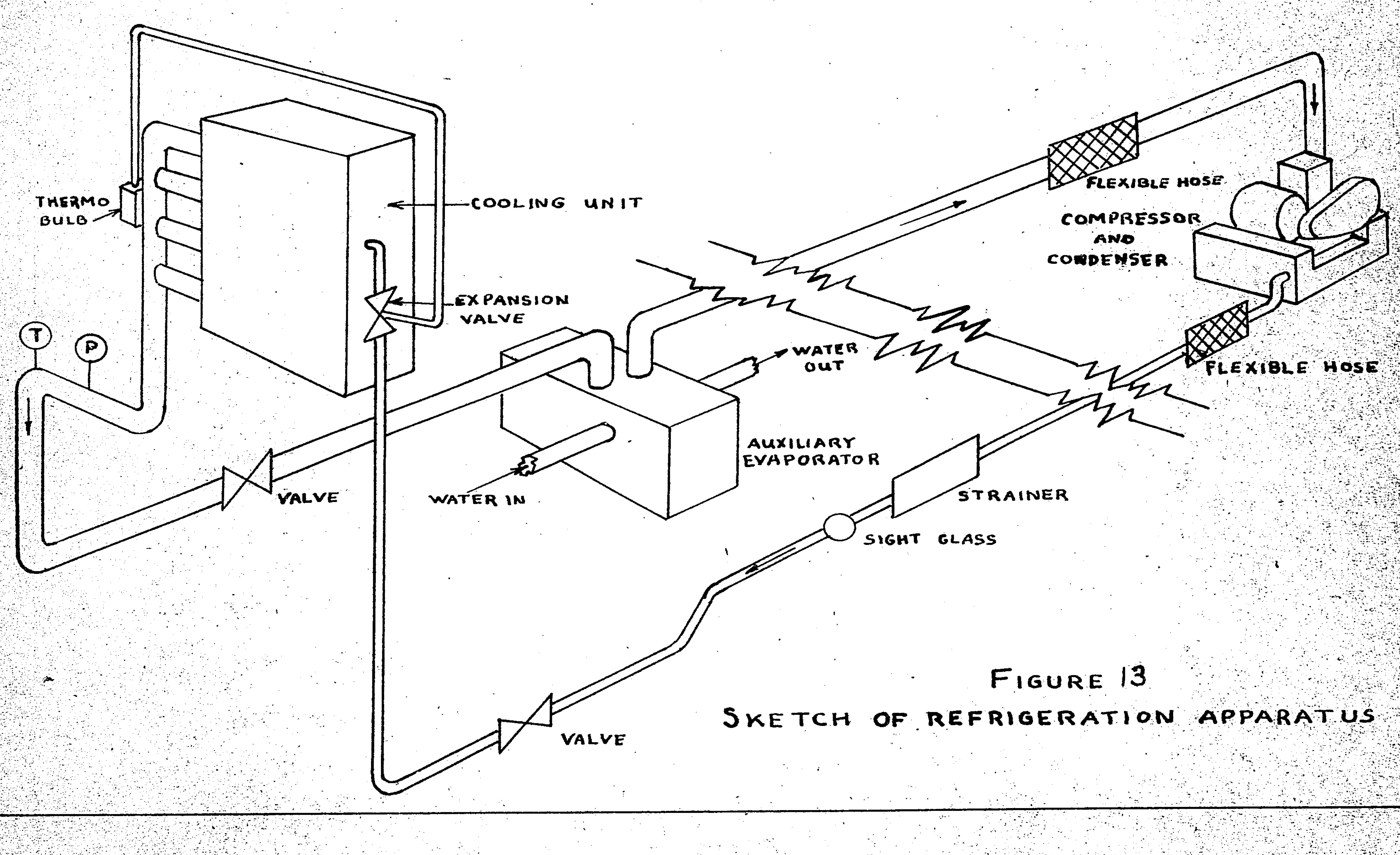


FIGURE 13
SKETCH OF REFRIGERATION APPARATUS

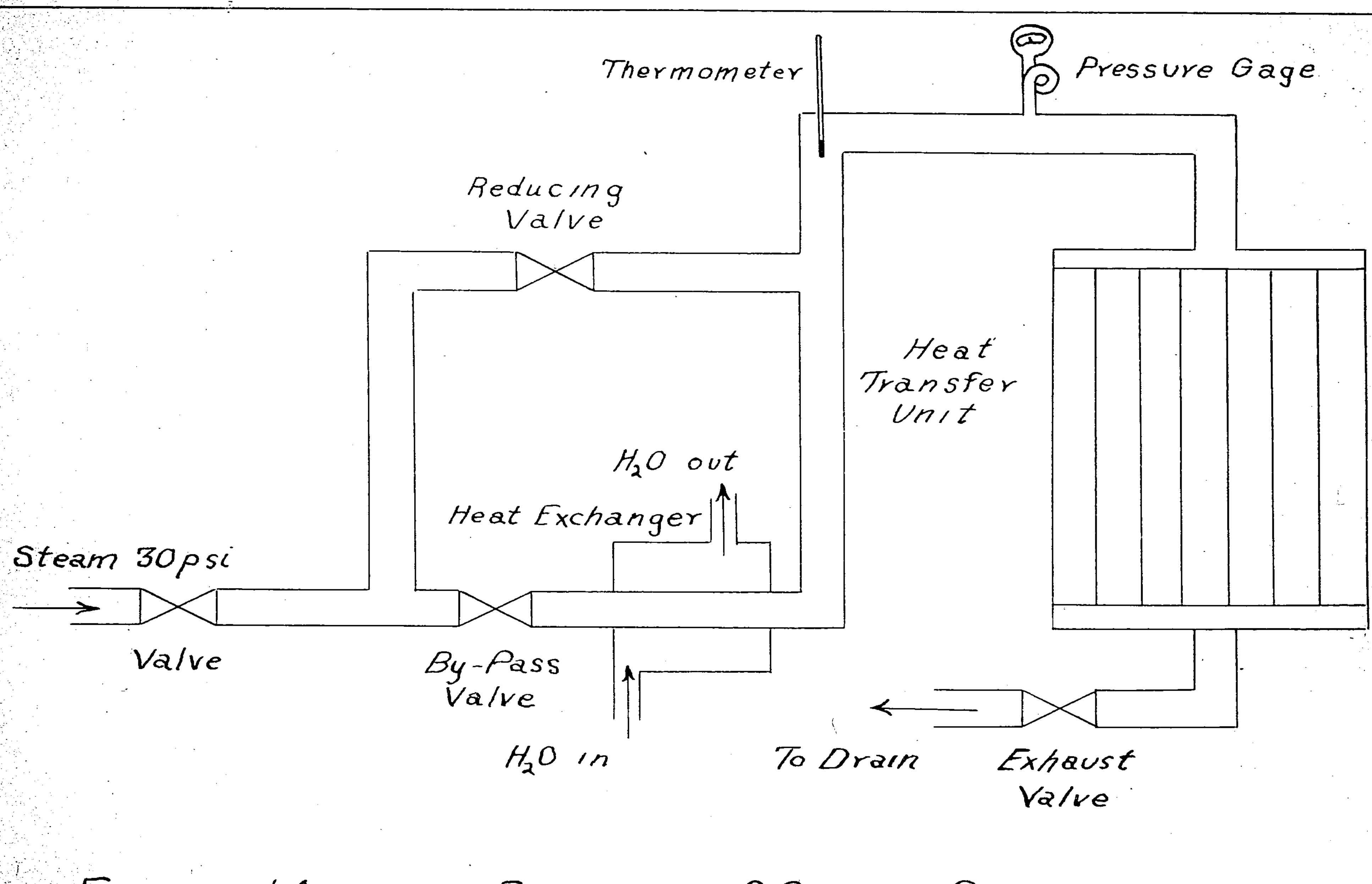


FIGURE 14 DIAGRAM of STEAM SYSTEM

Calibration of Instruments

The steam pressure gage was calibrated with a dead weight gage tester and was found to be accurate within 0.05 pounds at zero, five, and ten pounds dead weight, and all readings were therefore accepted as correct as read.

The inlet and outlet air thermometers were calibrated against a Bureau of Standards Thermometer. The steam thermometer was checked by allowing saturated steam at a definite pressure to fill the unit and comparing the temperature read on the thermometer with the temperature obtained from steam tables for saturated steam at the particular pressure. The thermometer was found to be accurate within 0.2 degrees at steam pressure up to ten pounds gage. An error of 0.2 degrees in steam temperature causes an error in the overall heat transfer coefficient of less than 0.3 percent, and steam thermometer readings were accepted as read.

The orifice was constructed according to the A. S. M. E. specifications and the orifice coefficients listed in the Fluid Meters report were used and accepted as correct (2, 3). As a rough check of the accuracy of the orifice coefficients, the orifice was used in conjunction with a Pitot tube, and using the orifice coefficients, a Pitot tube coefficient was determined. This coefficient was found to be 0.94, a reasonable value.

Calibration data are not included in this report. These data are to be found in the original reports of Foust, Hinterleiter, McKinley, and McDonnell. (3, 5, 12, 9).

Experimental Procedure

In tests to determine the overall heat transfer coefficient, the blower pulls air through the test apparatus and the desired rate of air flow is achieved by adjusting the sliding damper over the flower outlet. Steam is admitted through the reducing valve and its by-pass valve, and the cooling water is turned on to eliminate steam superheat. The exhaust valve is partially opened so that live steam is continuously exhausted. After allowing sufficient time to elapse for the unit to reach a steady state, readings are made every two or three minutes over a period of approximately fifteen minutes. The following quantities are obtained: steam temperature; steam pressure; inlet air temperature; heated air temperature; and orifice pressure drop.

In determining pressure drop, measurements are made: 1. isothermally at room temperature; and 2. with air being heated. In either case the following readings are made: hook gage manometer pressure drop; orifice pressure drop; and orifice air temperature. Pressure drop measurements were also made with no unit in the apparatus to get a correction value.

Tests were also made with "cold spots" present and without baffling, to determine the effect of these two factors on pressure drop and overall heat transfer coefficient. These data are not presented in this report. A qualitative description of the results of these runs is presented in the discussion of results.

Table IV

Sample Experimental Data -- Run M-1

Time	Air Temperature In °C	Air Temperature Out °C	Steam Pressure p.s.i.g.	Temp. °F	K_w in H_2O	Hook Gage in H_2O
9:47	26.6	76.2	5.0	228	3.12	102×10^{-3}
9:49	26.5	76.5		228		
9:51	27.0	76.7		227.8		
9:53	27.0	76.9		228		
9:55	26.9	77.0		228		
9:57	27.0	77.0		228		
9:59	27.0	77.0		228		
10:01	27.0	77.0		228		
Av.	27.0	77.0	5.0	228	3.12	102×10^{-3}
	- 0.10	- 0.3				
Corrected	26.9	76.7		228	3.12	102×10^{-3}

Sample Calculations

Air Flow Rate

The air flow rate is calculated as in the refrigeration tests. See Sample Calculations Part I.

Overall Heat Transfer Coefficient:

For Run No. M-1, Unit 107 7/8-1.875-3-35.

$A = 139.8$ sq. ft.

$G = 4210$ lb/hr./sq.ft. free area

$W = 6558$ lb./hr.

$T_s = 228\ ^oF.$

$T_i = 80.4\ ^oF.$, $T_o = 169.95\ ^oF.$

$$Q = W\, C_p\, (T_o - T_i)$$

$$Q = (4210)(0.2375)(169.95 - 80.4) = 139{,}600 \text{ B.t.u./hr.}$$

$$U = \frac{Q}{A\ \Delta t_{mean}}$$

$$U = \frac{139{,}600}{(139.8)\ \dfrac{(228-80.4) - (228-169.95)}{2.303 \log \dfrac{228 - 80.4}{228 - 169.95}}}$$

$U = 10.39$ B.t.u./hr./sq. ft./^oF

Pressure Drop:

The pressure drop through the unit as measured by the hook gage manometer is read directly in inch of water. To correct the pressure drop reading, it is necessary to subtract the pressure drop through the empty duct at the same air rate, W.

For Unit 107 7/8 - 1.875-3-35 at G = 2010:

$P = 28.0 \times 10^{-3}$ in of H_2O

$W = G \times A_f = 2010 \times 1.560 = 3135$ lbs./hour

From Figure 18:

At $W = 3135$, empty duct drop $= 5.4 \times 10^{-3}$ in of H_2O

Corrected Pressure Drop, $\Delta H = (28.0-5.4) \times 10^{-3}$

$\Delta H = 22.6 \times 10^{-3}$ in of H_2O

Table V

Heat Transfer Data

Heater Unit 107-7/8-1.875-3-35

Run No.	Mass Rate of Air Flow, G pounds / (hr.)(sq. ft.)	Overall Heat Transfer Coefficient, U B.t.u. / (hr)(sq.ft.)(°F.)
M-1	4210	10.39
M-2	4100	10.35
M-3	3980	10.17
M-4	3845	10.01
M-5	3720	9.79
M-6	3560	9.55
M-7	3435	9.205
M-8	3300	9.02
M-9	3155	8.69
M-10	3012	8.555
M-11	2860	8.30
M-12	2685	7.93
M-13	2508	7.71
M-14	2310	7.29
M-15	2075	6.73

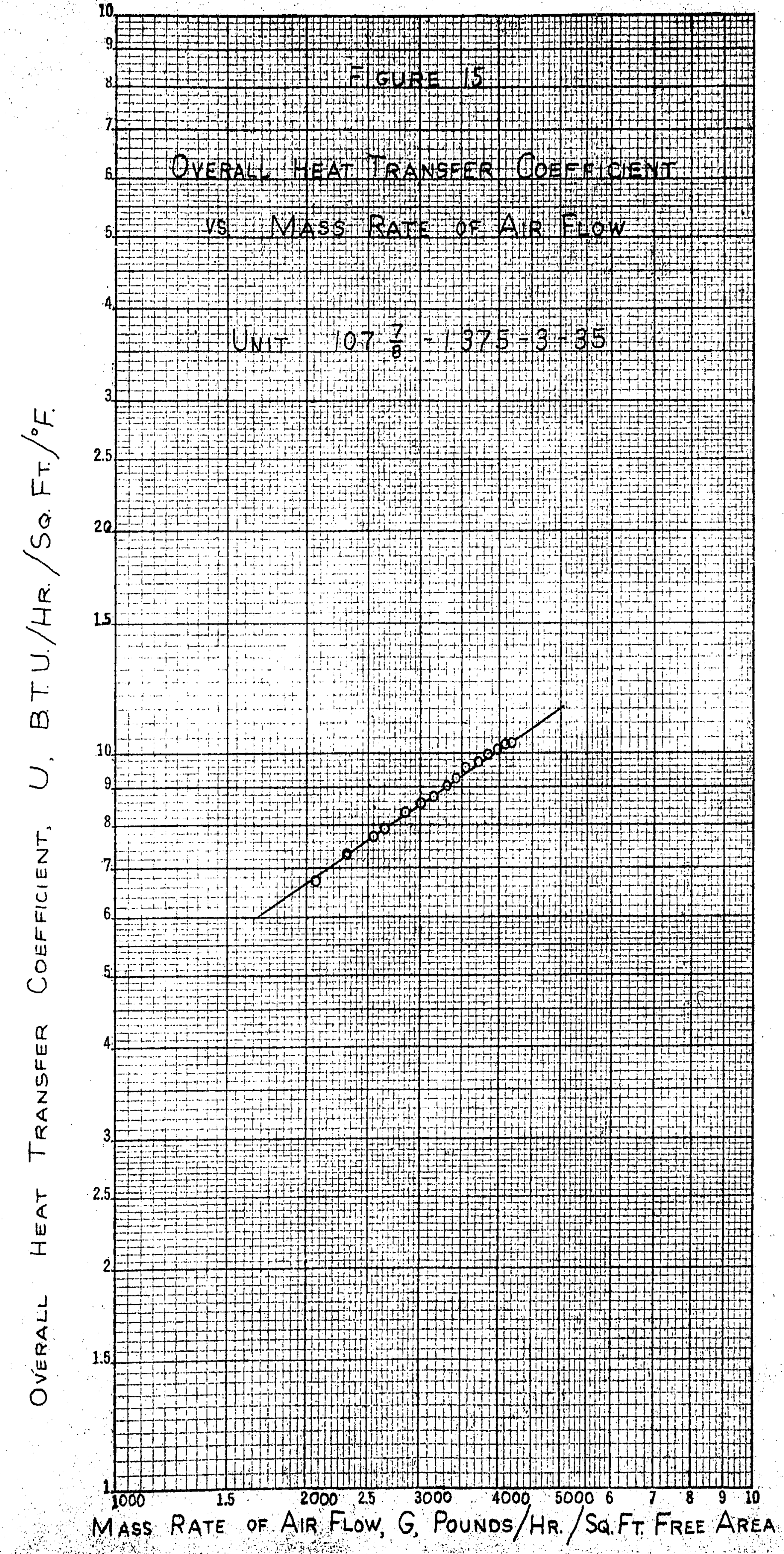


Figure 15
Overall Heat Transfer Coefficient
vs Mass Rate of Air Flow
Unit 107 7/8 - 1375-3-35
Overall Heat Transfer Coefficient, U, B.T.U./Hr./Sq. Ft./°F.
Mass Rate of Air Flow, G, Pounds/Hr./Sq.Ft. Free Area

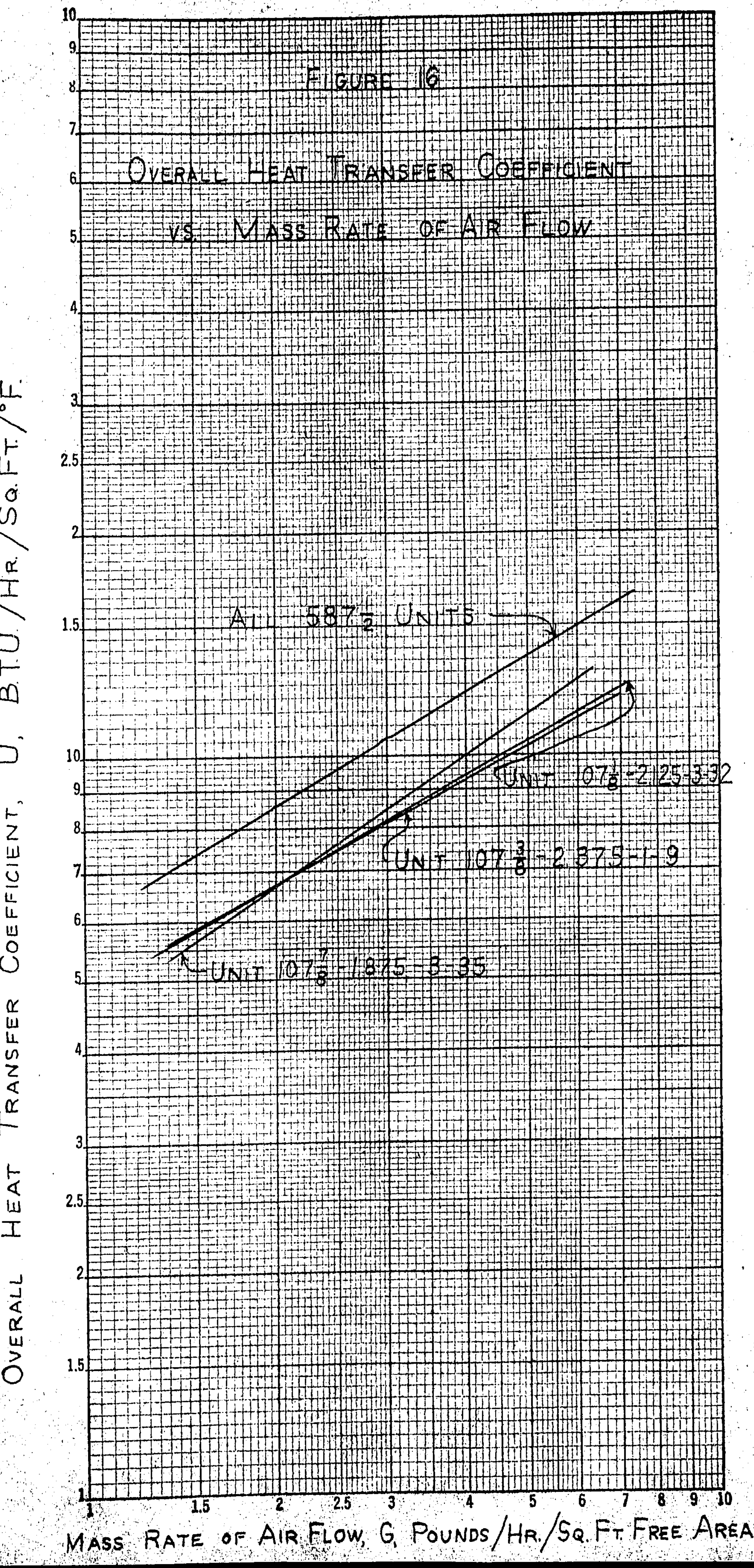


Figure 16
Overall Heat Transfer Coefficient
vs. Mass Rate of Air Flow
All 587½ Units
Unit 07⅛-2.25-3-32
Unit 107⅜-2.375-1-9
Unit 107⅞-1.875-3-35
Overall Heat Transfer Coefficient, U, B.T.U./Hr./Sq.Ft./°F.
Mass Rate of Air Flow, G, Pounds/Hr./Sq.Ft. Free Area
10
9
8
7
6
5
4
3
2.5
2
1.5
1
1.5
2
2.5
3
4
5
6
7
8
9
10

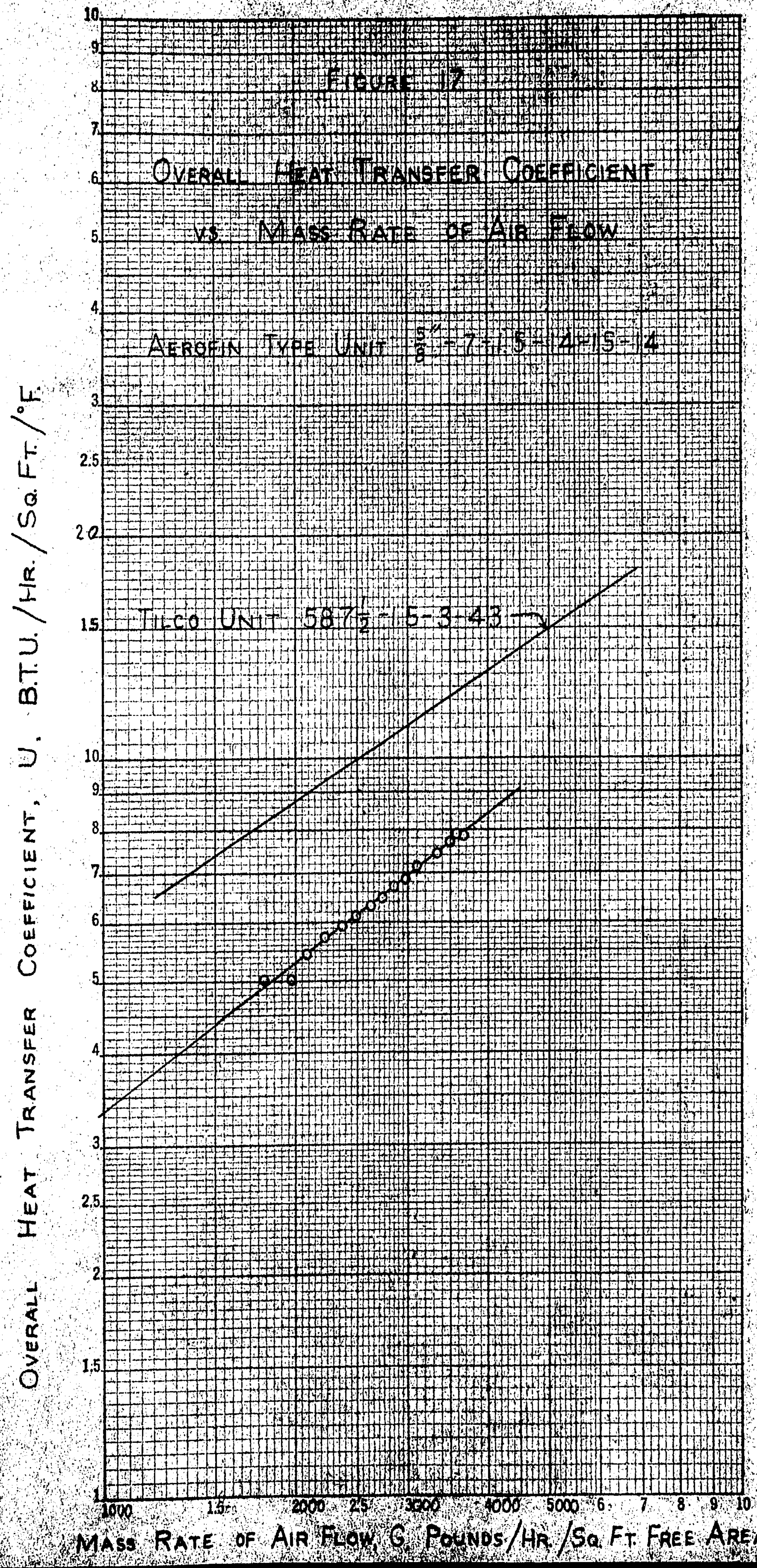


FIGURE 17
OVERALL HEAT TRANSFER COEFFICIENT
VS MASS RATE OF AIR FLOW
AEROFIN TYPE UNIT 5/8"-7-1.5-14-15-14
TILCO UNIT 587 1/2-1.5-3-43
OVERALL HEAT TRANSFER COEFFICIENT, U, B.T.U./HR./SQ.FT./°F
MASS RATE OF AIR FLOW, G, POUNDS/HR./SQ.FT. FREE AREA

Table VI

Pressure Drop Through Empty Duct (No Unit)

Isothermal Tests at 25°C.

Inches of Water Pressure Drop Through Empty Duct	W, Pounds of Air per Hour
5	3105
6	3435
6.5	3719
7	3990
8	4260
9	4520
9.5	4725
11	4940
11.5	5150
12	5360
12.5	5550
12.5	5700
15	5960
16	6230
16.5	6490
18	6740
19.5	6950
20.5	7250
21.5	7620
19.5	7250
17.5	6860
15.0	6560
14.0	6330
13.5	6070
12.5	5810
11.5	5620
11	5440
11	5260
10.5	5150
10	4920
9.5	4630
9	4210
8.5	3940
7.5	3580
6.5	3200
6.0	2820
6	2810
7	2580
8	4500
11.5	5550
13	6240
15	6850
16	7210

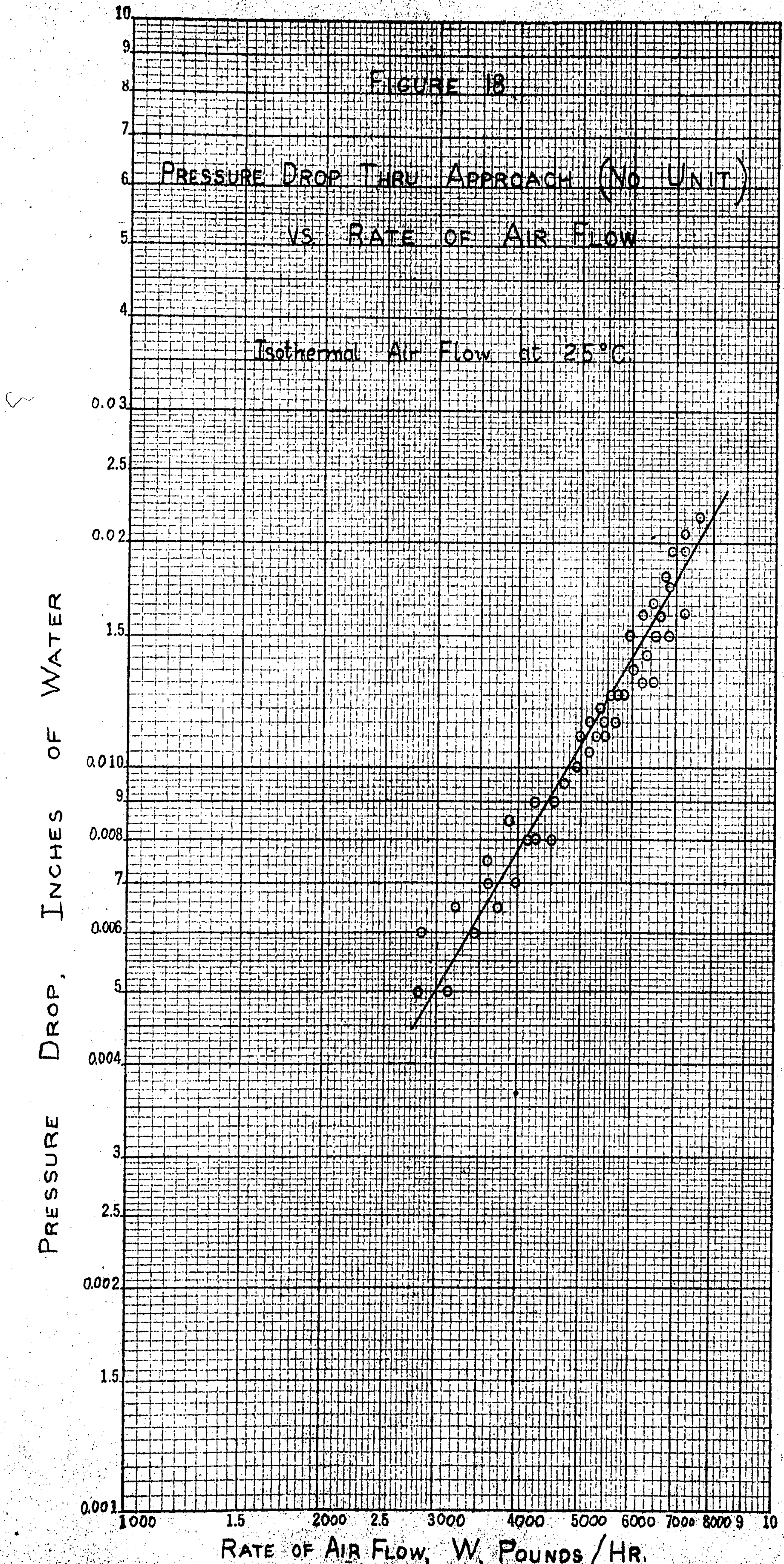


FIGURE 18
PRESSURE DROP THRU APPROACH (NO UNIT)
VS. RATE OF AIR FLOW
Isothermal Air Flow at 25°C.
PRESSURE DROP, INCHES OF WATER
RATE OF AIR FLOW, W, POUNDS/HR.

Table VII

Pressure Drop Data

Unit 107-7/8-1.875-3-35

Isothermal Air Flow at 25°C.

Mass Air Rate, G pounds / (hr.)(sq.ft.)	Pounds Air Per Hr., W	Pressure Drop Through Unit, inches of Water, Measured	Correction Value inches of Water	Corrected Pressure Drop Δ P, in. of water
2010	3136	28.0×10^{-3}	5.4×10^{-3}	22.6×10^{-3}
2165	3375	30.5	6.0	24.5
2386	3725	35	7.0	28
2582	4030	39	7.7	31.3
2780	4330	46	8.6	37.4
2980	4645	55	9.4	45.6
3165	4940	61	10.5	50.5
3342	5210	65	11.4	53.6
3525	5500	72	12.5	59.5
3708	5785	78.5	13.3	65.2
3840	6060	87	14.2	72.8
4080	6360	94	15.3	78.7
4210	6610	100.5	16.4	84.1
4430	6906	106	17.5	88.5
4610	7200	118	18.5	99.5
4790	7470	124	19.7	104.3

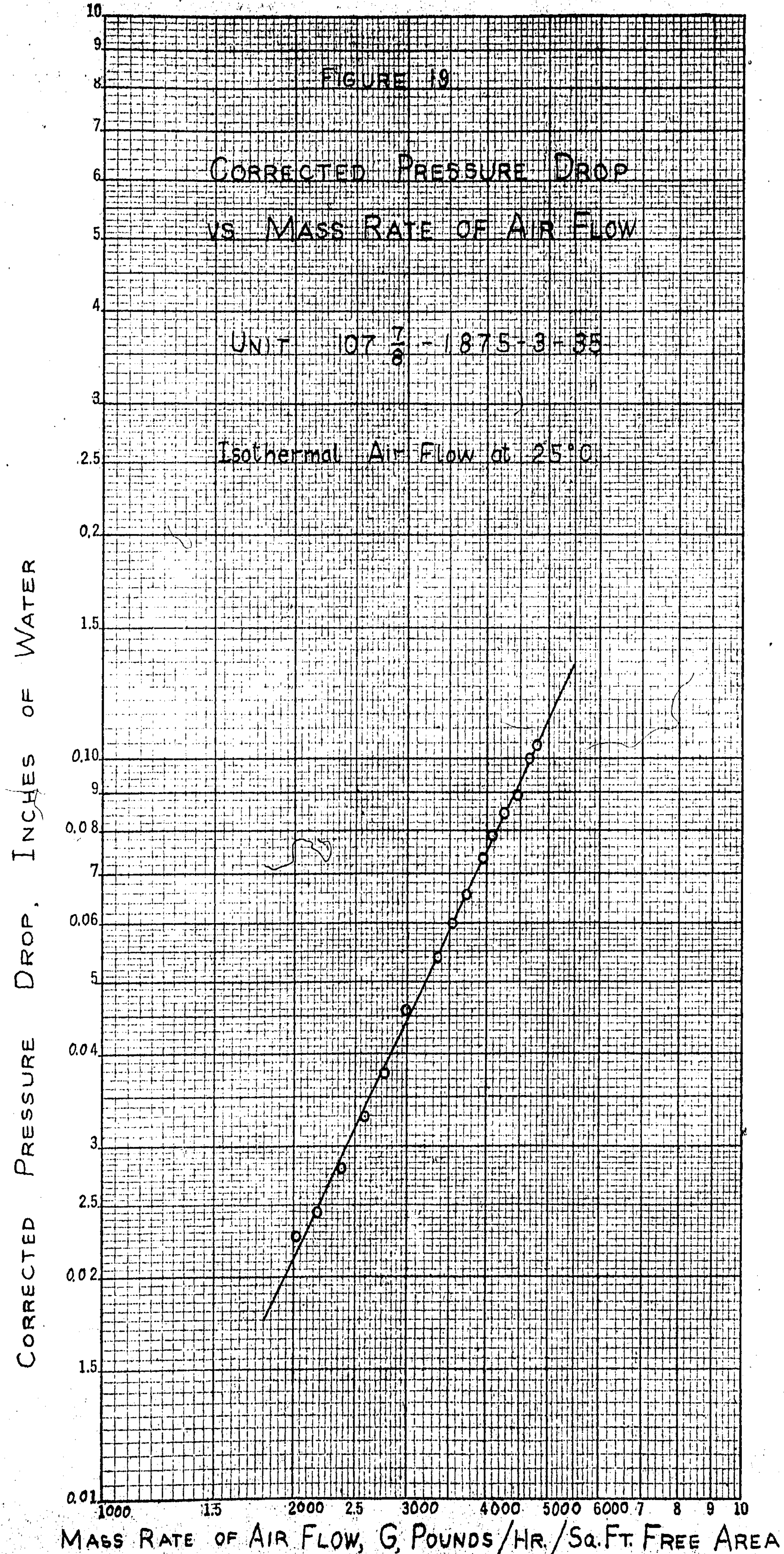


FIGURE 19
CORRECTED PRESSURE DROP
VS MASS RATE OF AIR FLOW
UNIT 107 7/8 - 1875-3-35
Isothermal Air Flow at 25°C
CORRECTED PRESSURE DROP, INCHES OF WATER
MASS RATE OF AIR FLOW, G, POUNDS/HR./SQ.FT. FREE AREA
10
9
8
7
6
5
4
3
.2.5
0.2
1.5
0.10
9
0.08
7
0.06
5
0.04
3
2.5
0.02
1.5
0.01
1000
1.5
2000
2.5
3000
4000
5000
6000
7
8
9
10

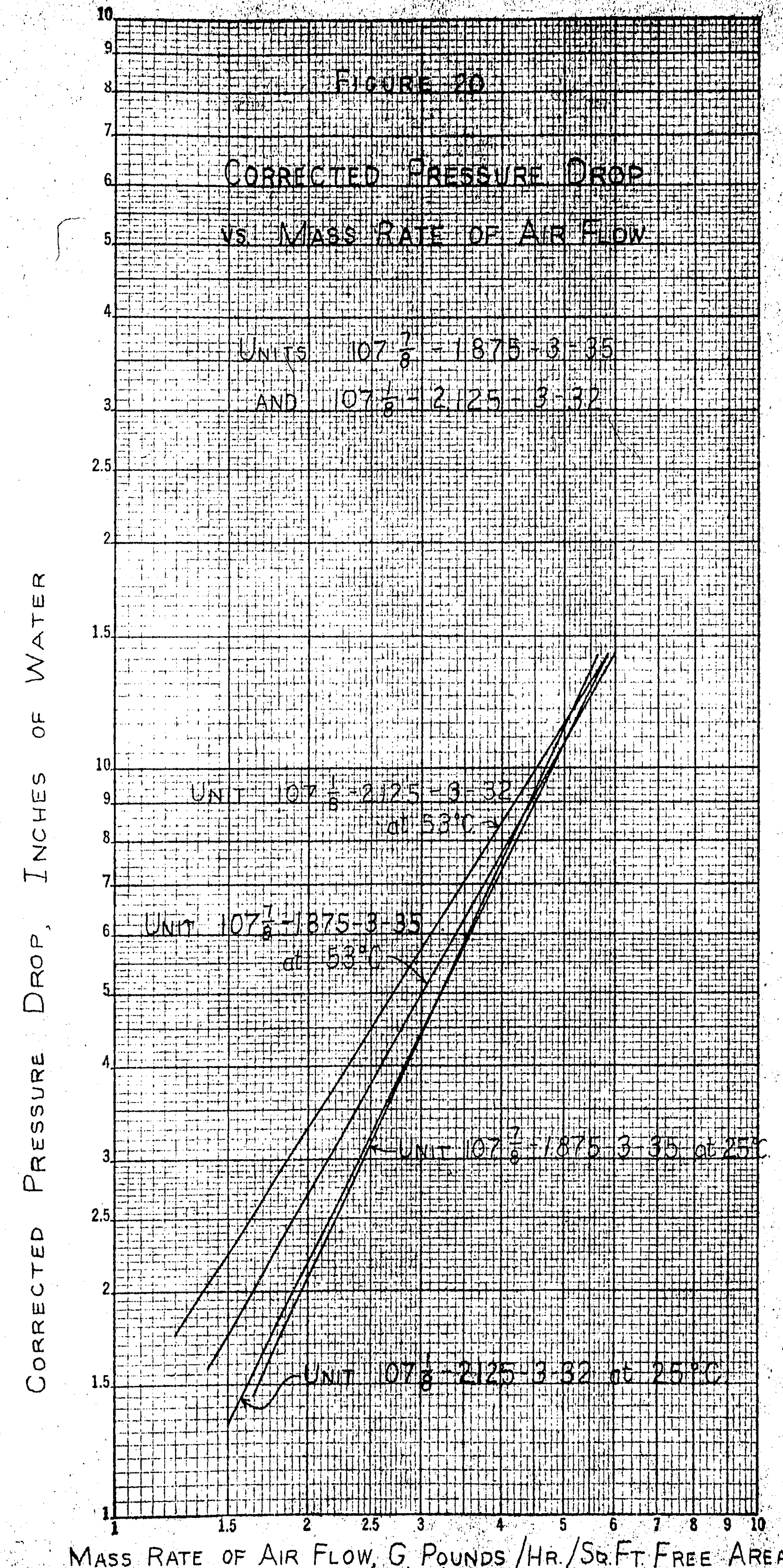


FIGURE 20
CORRECTED PRESSURE DROP
VS. MASS RATE OF AIR FLOW
UNITS 107 7/8 - 1875 - 3 - 35
AND 107 1/8 - 2125 - 3 - 32
UNIT 107 1/8 - 2125 - 3 - 32 at 53°C
UNIT 107 7/8 - 1875 - 3 - 35 at 53°C
UNIT 107 7/8 - 1875 - 3 - 35 at 25°C
UNIT 107 1/8 - 2125 - 3 - 32 at 25°C
CORRECTED PRESSURE DROP, INCHES OF WATER
MASS RATE OF AIR FLOW, G, POUNDS/HR./SQ.FT. FREE AREA

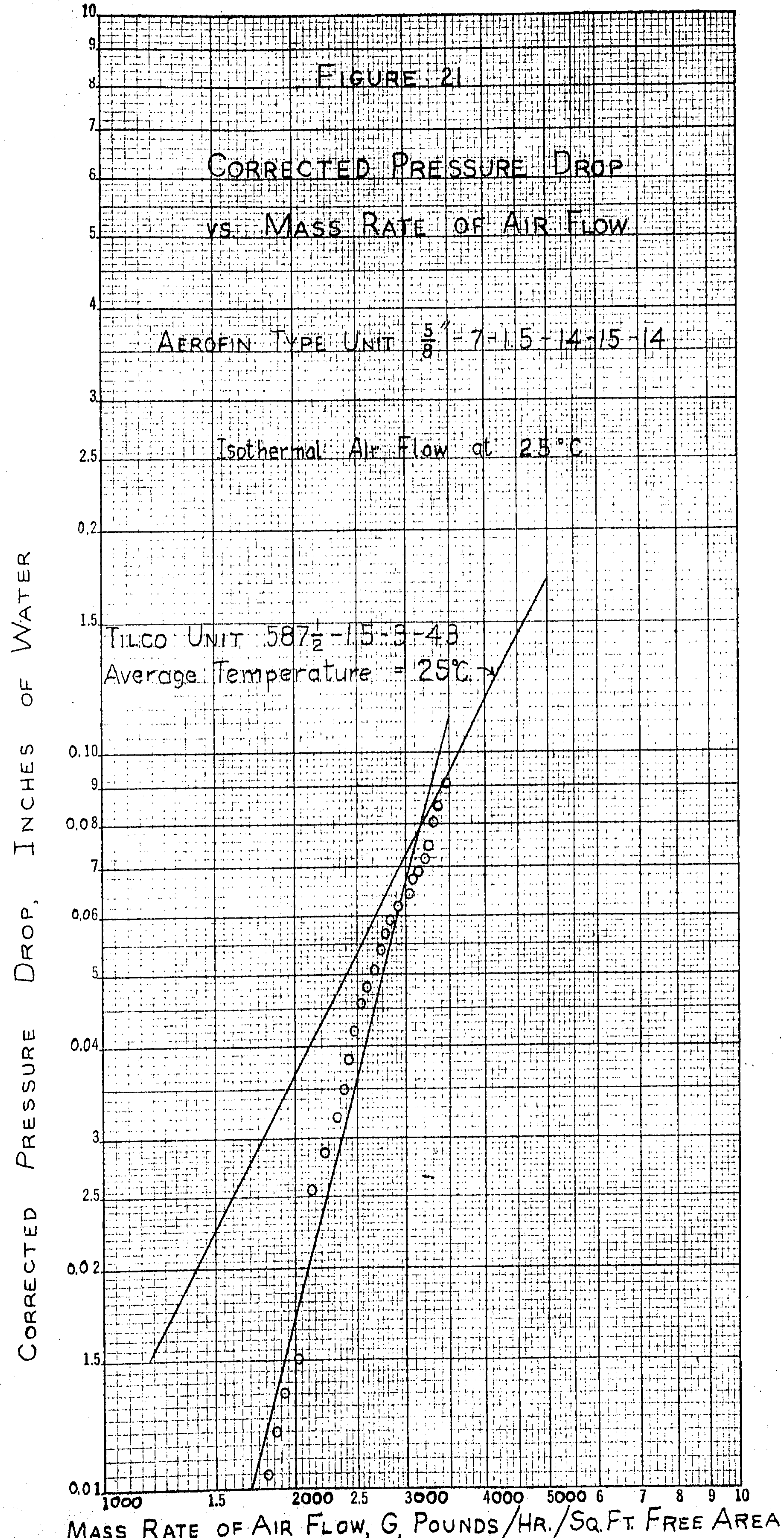


FIGURE 21
CORRECTED PRESSURE DROP
VS. MASS RATE OF AIR FLOW
AEROFIN TYPE UNIT 5/8" - 7 - 1.5 - 14 - 15 - 14
Isothermal Air Flow at 25°C.
TILCO UNIT 587½ - 1.5 - 3 - 43
Average Temperature = 25°C.
CORRECTED PRESSURE DROP, INCHES OF WATER
MASS RATE OF AIR FLOW, G, POUNDS/HR./SQ.FT. FREE AREA

Discussion of Results

Heat Transfer Coefficients

For all units the overall heat transfer coefficients are plotted against mass rate of air flow on logarithmic graph paper. In each case the best straight line fitting the data was drawn.

In Figure 16 the heat transfer coefficient versus the mass rate of air flow for units 107-7/8-1.5-3-35, 107-1/8-5-32, and 107-5/8-2.375-1-[illegible] (corrected from a previous report) are plotted. It is to be seen that the overall heat transfer coefficient is nearly independent of fin length in air heating applications. From Figure 16 it is also to be seen that the overall heat transfer coefficient for one inch tubes is about 20 percent less than for five-eighths inch tubes.

In Figure 17 the data for the Aerofin-Type Unit 5/8-7-1.5-3-43 are plotted, and the line $U = 0.204\ G^{0.435}$, representing the performance of the similar Wilco-Fin Unit is drawn for comparison. The overall coefficient of the Wilco-Fin Unit is seen to be about [illegible]7.5% higher at the lowest test values of G, and about [illegible]6.7% higher at the highest values of G.

In those tests of the Aerofin-Type unit where cold spots were present, the overall coefficient was found to be 84% of the value with baffling and no cold spots. Without baffling and with cold spots caused the coefficient to be reduced to 80% of the baffled, no cold spot value.

Pressure Drops

For each unit, pressure drop measurements were made isothermally at approximately 25 degrees Centigrade, and with the air being heated as it passes through the heat transfer unit. In all cases the corrected pressure drops are plotted versus the mass rate of air flow.

From Figure 20 in which are shown the data for Units 107-7/8-1.875-3-35 and 107-1/8-2.375-3-32, it may be seen that the isothermal pressure drop of Wilco-Fin tubing differs considerably from the pressure drop with the air being heated. In general the isothermal pressure drop is lower than the pressure drop with the air being heated; this is to be expected since the pressure drop is a function of the square of the velocity. The velocity increases with the decrease in density caused by the increase in temperature.

The variation in pressure drop with change in fin length is small as seen in Figure 20. The isothermal pressure drop at 25 degrees Centigrade for the 107-1/8 tubes (9/16 inch fin) is less than that for the 107-7/8 tubes (7/16 inch fin), but with the air being heated, the pressure drop for the 107-1/8 tubes is greater than for the 107-7/8 tubes.

In Figure 21, the data for the isothermal pressure drop at 25 degrees Centigrade of the Aerofin-Tube Unit 5/8-7-1.5-3-43 are presented along with the isothermal pressure drop at 25 degrees Centigrade data for Wilco-Fin

Unit 587-1/2-1.5-3-43. It may be seen that at lower values (1000-3500) of G, the Tilco-Fin Unit has the higher pressure drop, while at values of G above 3500, the Aerofin-Type Unit has the higher pressure drop.

Part III - Heat Transfer Correlation

Discussion of Theory

It was desired to correlate all the heat transfer data by means of one of the standard equations:

$$\left(\frac{h}{c_p G}\right)\left(\frac{c_p \mu_f}{k}\right)^{2/3} = \phi\, Re\,; \quad \left(\frac{hD}{k}\right)\left(\frac{k}{c\mu}\right)^{0.4} = \phi\, Re\,; \text{ and } hD = \phi G.$$

The first two of these forms are equivalent, and if one form correlates the data, the other must also. When it was found that the first correlation proposed does correlate the data, it was known that the second does also. Since all one inch units yield substantially the same line on a plot of [illegible] against [illegible], and all five-eighths units yield another line, for the second correlation, two points from each average line of [illegible] vs. [illegible] were selected to determine the correlation line. This method was also used in the third correlation attempt.

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Heat Transfer Correlation Calculations

I. $$\left(\frac{h}{C_pG}\right)\left(\frac{C_p\mu_f}{K}\right)^{2/3} = \phi\ (Re)$$

Definition of Symbols:

h = heat transfer coefficient = U, overall heat transfer coefficient, B.t.u./hr./sq.ft./°F.

C_p = specific heat of air at the average temperature, B.t.u./lb./°F.

G = mass rate of air flow, lb./hr./sq.ft./net free area.

$\frac{C_p\mu_f}{K}$ = Prandtl number = 0.74 for air, where f = film temperature, and k = thermal conductivity of air, B.t.u./hr./sq.ft./°F./ft. (2)

$Re = DG/\mu$ = Reynold's number where D is the [illegible] tube size, ft., and μ is the viscosity of air at the average temperature, lb./sec.ft.

For Run No. 22-1 - Unit 587-1/2-1.5-1-15:

$$\left(\frac{h}{C_pG}\right)\left(\frac{C_p\mu_f}{K}\right)^{2/3} = \frac{7.51}{(.2375) \times 1426}\ (0.818) = 1.820\times10^{-2}$$

$$Re = \frac{DG}{\mu} = \frac{\frac{0.625}{12} \times 1426}{.01895 \times 2.42} = 1620$$

II. $$\left(\frac{hD}{k}\right)\left(\frac{k}{C_p\mu}\right)^{0.4} = \phi\ (Re)$$

Definition of Symbols:

As Above:

For 5/8" tubing: U = 7.5 at [illegible] = 1500

$$\left(\frac{hD}{k}\right)\left(\frac{k}{C_p\mu}\right)^{0.4} = \frac{7.5 \times \frac{0.625}{12}}{.0160} \quad \frac{1}{.899} = 27.16 \qquad (8)$$

$$Re = \frac{DG}{\mu} = \frac{\frac{0.625}{12} \times 1500}{.0506} = 1543$$

III. $hD = \phi\, G$

For 5/8" tubing: $U = 7.5$ at $G = 1500$

$$hD = (7.5)\ \frac{0.625}{12} = 0.3908$$

$G = 1500$

Table VIII

Heat Transfer Correlation

Heater Unit 107-7/8-1.875-3-35

Re	$\left(\frac{h}{C_pG}\right)\left(\frac{C_p\,\mu_f}{k}\right)^{2/3}$
7410	8.50×10^{-3}
7210	8.64
7000	8.81
6770	9.05
6545	9.07
6260	9.24
6040	9.24
5805	9.40
5550	9.50
5300	9.78
5035	9.99
4725	10.18
4410	10.58
4060	10.89
3650	11.20

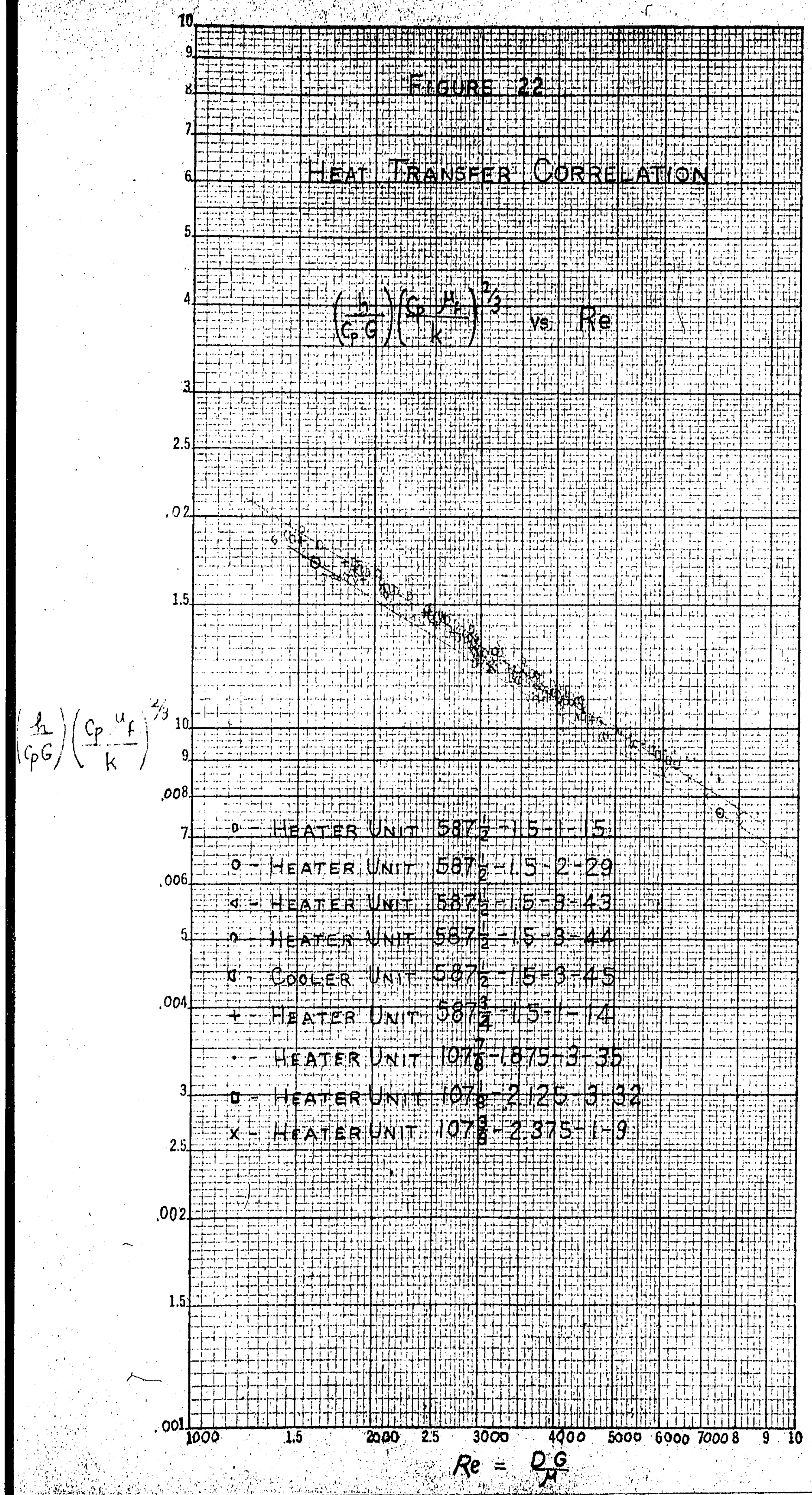
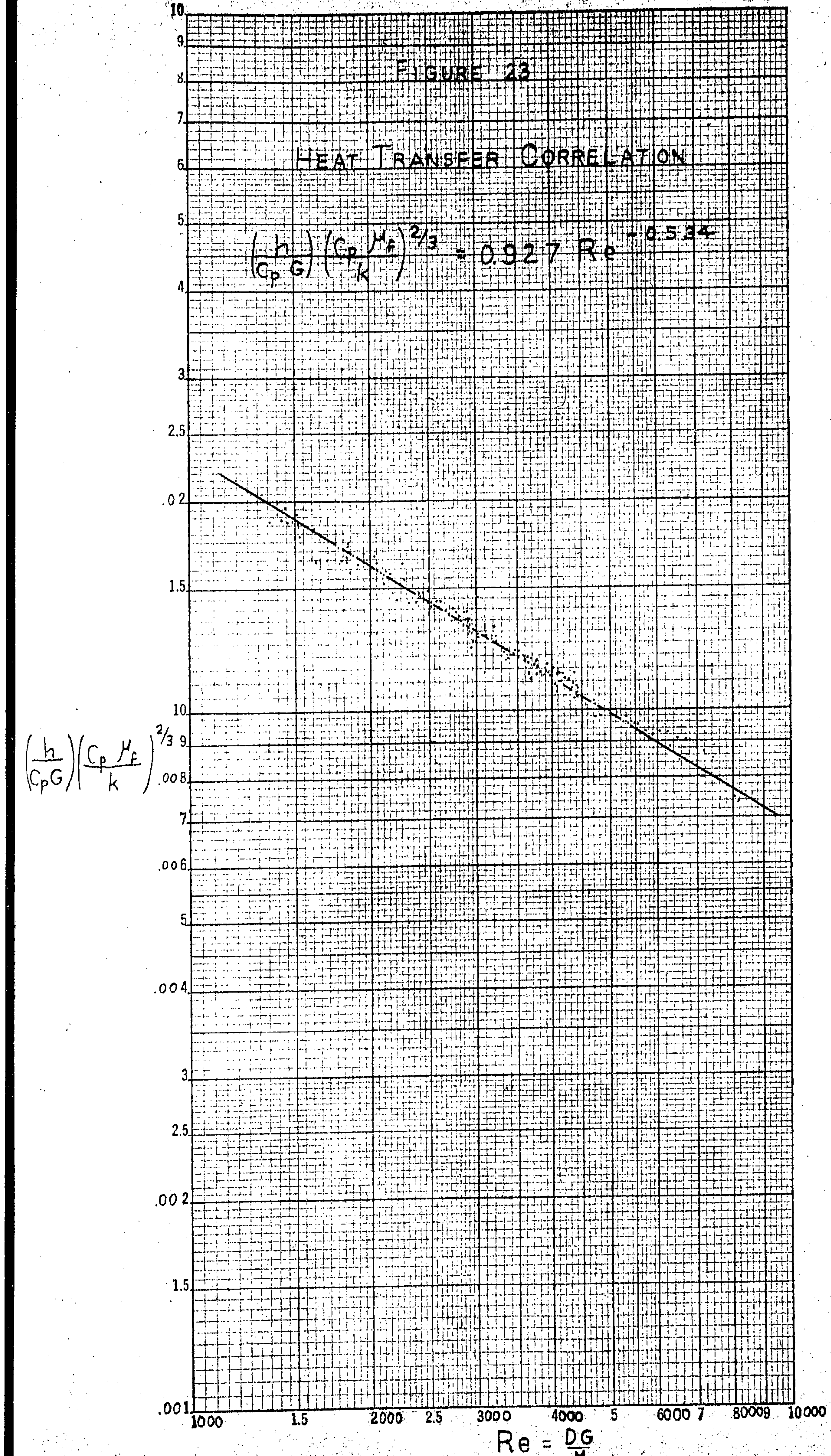


FIGURE 22
HEAT TRANSFER CORRELATION
$\left(\frac{h}{C_p G}\right)\left(\frac{C_p \mu_f}{k}\right)^{2/3}$ vs. Re
$\left(\frac{h}{C_p G}\right)\left(\frac{C_p \mu_f}{k}\right)^{2/3}$
- HEATER UNIT 587½-1.5-1-15
o - HEATER UNIT 587½-1.5-2-29
- HEATER UNIT 587½-1.5-3-43
- HEATER UNIT 587½-1.5-3-44
- COOLER UNIT 587½-1.5-3-45
+ - HEATER UNIT 587¾-1.5-1-14
· - HEATER UNIT 107⅞-1.875-3-35
- HEATER UNIT 107⅞-2.125-3-32
x - HEATER UNIT 107⅞-2.375-1-9
$Re = \frac{DG}{\mu}$

FIGURE 23

HEAT TRANSFER CORRELATION



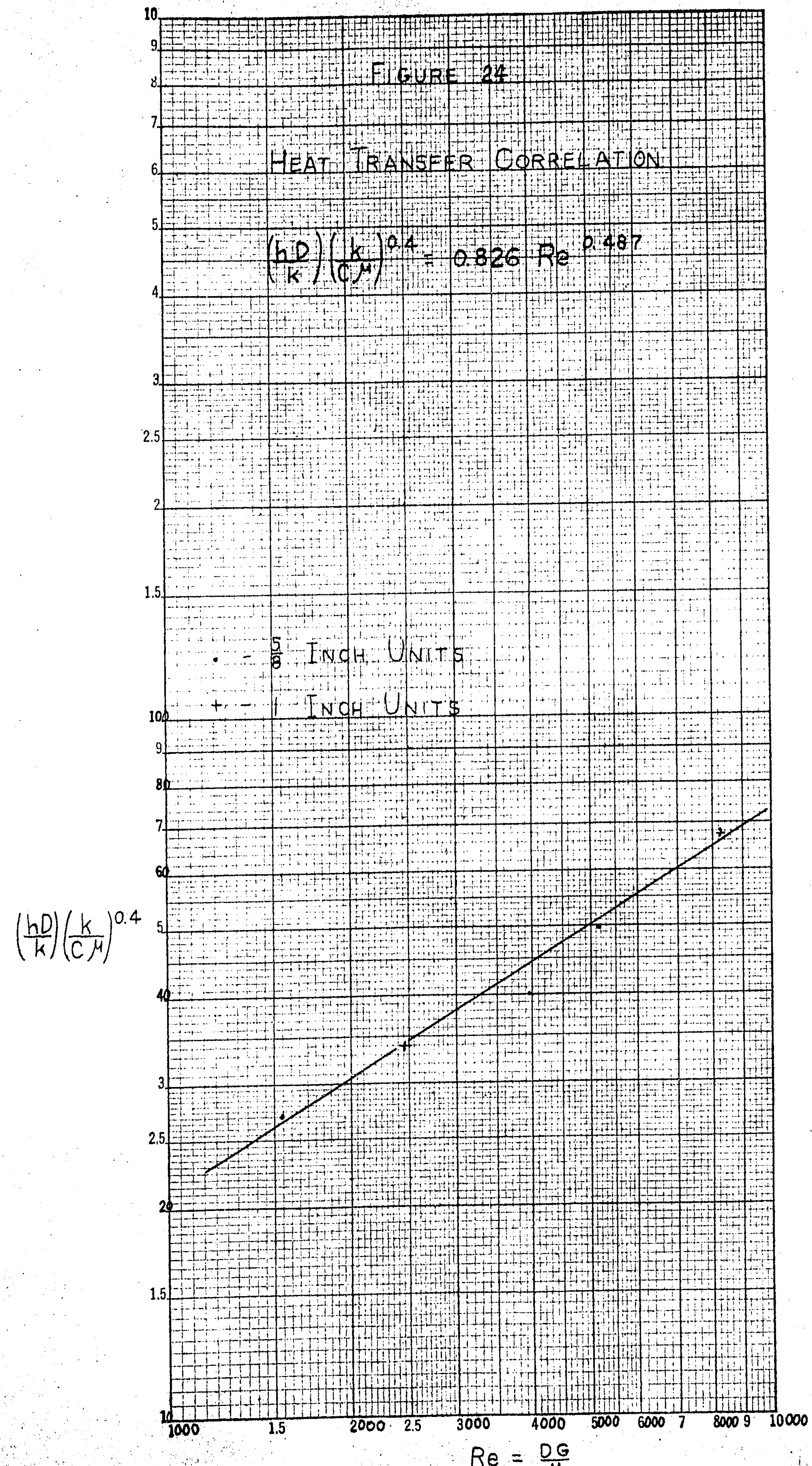


FIGURE 24
HEAT TRANSFER CORRELATION
$\left(\frac{hD}{k}\right)\left(\frac{k}{C\mu}\right)^{0.4} = 0.826\ Re^{0.487}$
• - $\frac{5}{8}$ INCH UNITS
+ - 1 INCH UNITS
$\left(\frac{hD}{k}\right)\left(\frac{k}{C\mu}\right)^{0.4}$
$Re = \frac{DG}{\mu}$
1000
2000
3000
4000
5000
6000
8000
10000

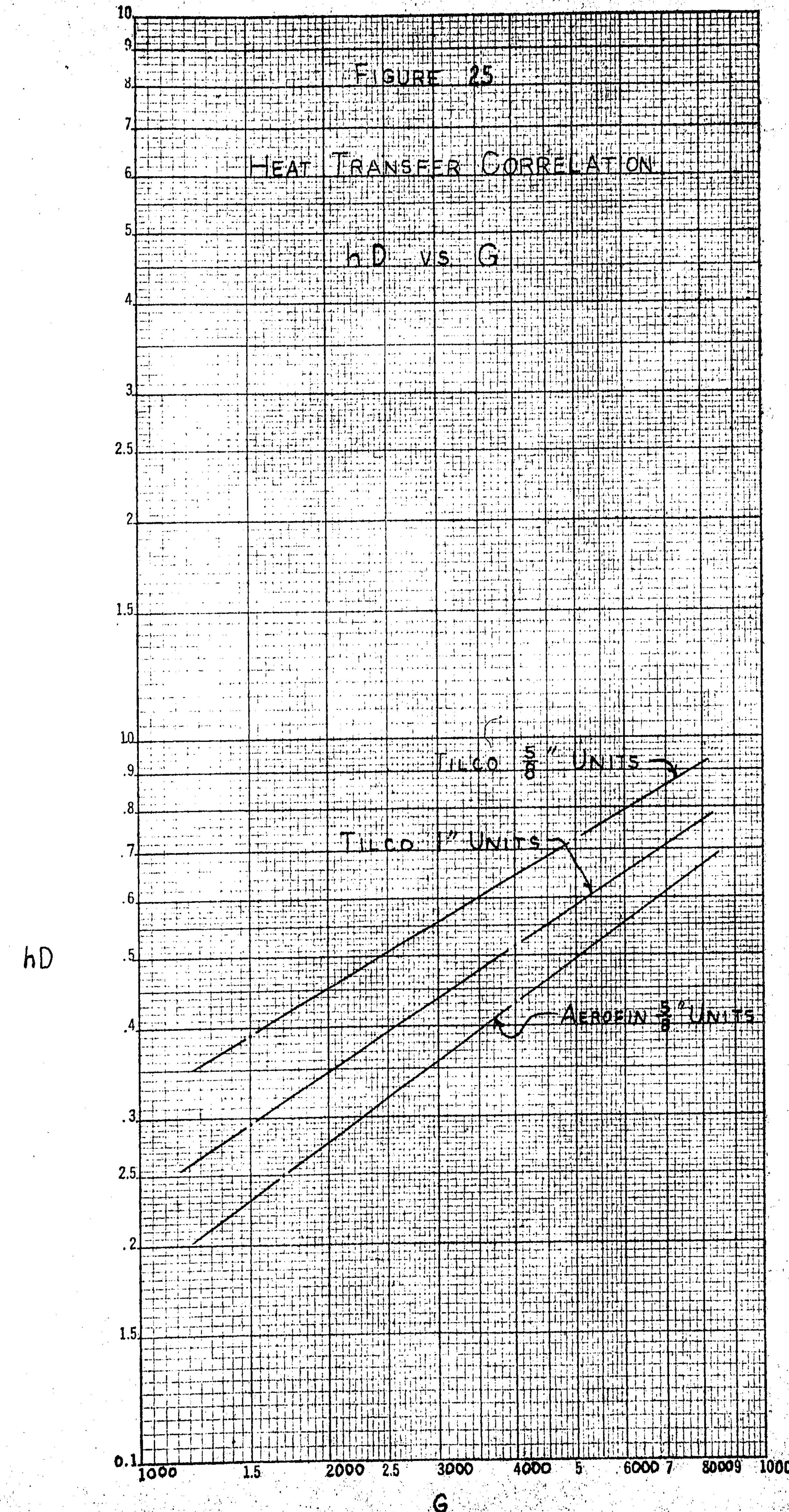


FIGURE 25
HEAT TRANSFER CORRELATION
hD vs. G
TILCO 5/8" UNITS
TILCO 1" UNITS
AEROFIN 5/8" UNITS
hD
G
1000
1.5
2000
2.5
3000
4000
5
6000
7
8000
9
10000
10
9
8
7
6
5
4
3
2.5
2
1.5
.9
.8
.7
.6
.5
.4
.3
.2
0.1

Discussion of Results

It was found that the data are correlated by the first two forms since $\left(\frac{h}{C_p G}\right)\left(\frac{C_p \mu_f}{K}\right)^{2/3}$ and $\left(\frac{hD}{K}\right)\left(\frac{k}{C\mu}\right)^{0.4}$ both yielded straight lines when plotted versus Re. See Figures 22, 23, and 24.

The data are not correlated by the form $hD = \phi\, G$, since the values of hD vs G do not plot as a straight line. See Figure 25.

The correlations may be expressed mathematically as:

$$\left(\frac{h}{C_p G}\right)\left(\frac{C_p \mu_f}{K}\right)^{2/3} = 0.927\ Re^{-0.534} \quad \text{and} \quad \left(\frac{hD}{K}\right)\left(\frac{K}{C\mu}\right)^{0.4} = 0.826\ Re^{-.487}$$

The maximum deviation of any experimental point from the line $\left(\frac{h}{C_p G}\right)\left(\frac{C_p \mu_f}{K}\right)^{2/3} = 0.927\ Re^{-0.534}$ is less than nine per cent, while the average deviation is about three per cent. The correlations apply to all units except those having overlapping fins when used in air cooling.

Part IV - Pressure Drop Correlations

Discussion of Gunter-Shaw Correlation Theory

It was desired, to test the applicability of the pressure drop correlation which Gunter and Shaw propose for all types of surfaces in crossflow (4). In this correlation, Gunter and Shaw found half the friction factor,

$$\frac{f}{2} = \frac{\Delta P g D_v \rho}{G^2 L} \left(\frac{\mu}{\mu_w}\right)^{0.140} \left(\frac{D_v}{S_t}\right)^{-0.40} \left(\frac{S_l}{S_t}\right)^{-0.6}$$

to be a function of the Reynold's Number, $Re = D_v G/\mu$, to the -0.145 power in the turbulent range, where:

$f/2$ = half friction factor, dimensionless

ΔP = pressure drop due to friction, lb./sq. ft.

g = acceleration due to gravity = 4.18×10^8 ft./hr./hr.

D_v = volumetric hydraulic diameter = $\frac{4 \times \text{Net free volume}}{\text{Friction surface}}$, ft.

ρ = fluid density, lb./cu. ft.

G = fluid mass velocity, lb./hr./sq. ft. net free area.

L = fluid flow length, ft.

S_L = longitudinal pitch, center to center distance from tube in one row to nearest tube in next transverse row.

S_T = transverse pitch, center to center distance from tube to tube in one transverse row.

μ = absolute viscosity at average main stream temperature, lb/ft./hr.

μ_w = absolute viscosity at surface wall temperature, lb./ft./hr.

Values of f/2 and Re were calculated for the serrated-finned tube pressure loss data (See Sample Calculations at bottom) and are plotted in Figure 26.

Gunter-Shaw Sample Calculations

Configuration Ratios:

For Unit 107-1/8-2.125-3-32:

Considering 1 ft. of tube length:

$$V_{total} = 1 \times \left(\frac{17}{8} \times \frac{1}{12}\right)^2 = 0.03133 \text{ ft.}^3$$

$$V_{occ.} = 1 \times \frac{\pi}{4} \times \left(\frac{1.022}{12}\right)^2 + 18.8 \times 7 \times 12 \times \frac{.010}{12} \times \frac{.171}{12} \times \frac{1}{2} \times \left(\frac{17}{8} - \frac{1.022}{12}\right)$$

$$= 0.00656 \text{ ft.}^3$$

$$V_{free} = V_{total} - V_{occ.} = 0.031 - 0.00656 = 0.02474 \text{ ft.}^3$$

$$D_V = \frac{4 \times 0.02474}{2.503} = 0.03952 \text{ ft.}$$

$$\left(\frac{D_V}{S_T}\right)^{-0.4} = \left(\frac{0.03952}{2.125}\right)^{-0.4} = 1.961$$

$$\left(\frac{S_L}{S_T}\right)^{-0.6} = 1 \text{ (tubes spaced on equilateral triangles)}$$

$$L = \frac{2(1.840) + 2.125}{12} = 0.484 \text{ ft.}$$

Correlation: $\Delta H = 0.0725$ in. at $G = 3585$ lb./hr.ft.^{2} for Unit 107-1/8-2.125-3-32 at 23°C.

$$Re = \frac{D_V G}{\mu} = \frac{0.03952\ (3585)}{0.01825 \times 2.42} = 1531$$

$$\frac{f}{2} = \frac{\Delta P\, g\, D_V\, \rho}{G^2 L} \left(\frac{\mu}{\mu_w}\right)^{0.14} \left(\frac{D_V}{S_T}\right)^{-0.4} \left(\frac{S_L}{S_T}\right)^{-0.6}$$

$$= \frac{0.0725 \times \frac{62.4}{12} \times 4.18 \times 10^{+8} \times 0.03952 \times .0745 \times 1 \times 1.961 \times 1}{(3585)^2 \times 0.484}$$

$$= 8.44 \times 10^{-2}$$

The Sieder-Tate factor, $\left(\frac{\mu}{\mu_w}\right)^{0.14}$, was assumed to be unity for all runs. For isothermal runs $\mu = \mu_w$ and the factor is equal to unity. For runs with the air being heated it is extremely difficult to evaluate the quantity μ_w since the wall temperature is unknown. Surface pyrometer measurements indicate fin surface temperatures of the order of 180°F. At this temperature the Sieder-Tate factor differs from unity by less than two per cent, and was therefore assumed to be unity for all runs.

Table IX

Gunter-Shaw Pressure Drop Correlation

Unit 107-7/8-1.875-3-35 at 27°C.

Re	f/2
1634	1.374×10^{-1}
1760	1.284
1940	1.208
2100	1.152
2260	1.190
2425	1.270
2575	1.239
2720	1.178
2868	1.166
3015	1.231
3207	1.152
3320	1.161
3447	1.152
3600	1.110
3752	1.145
3900	1.151

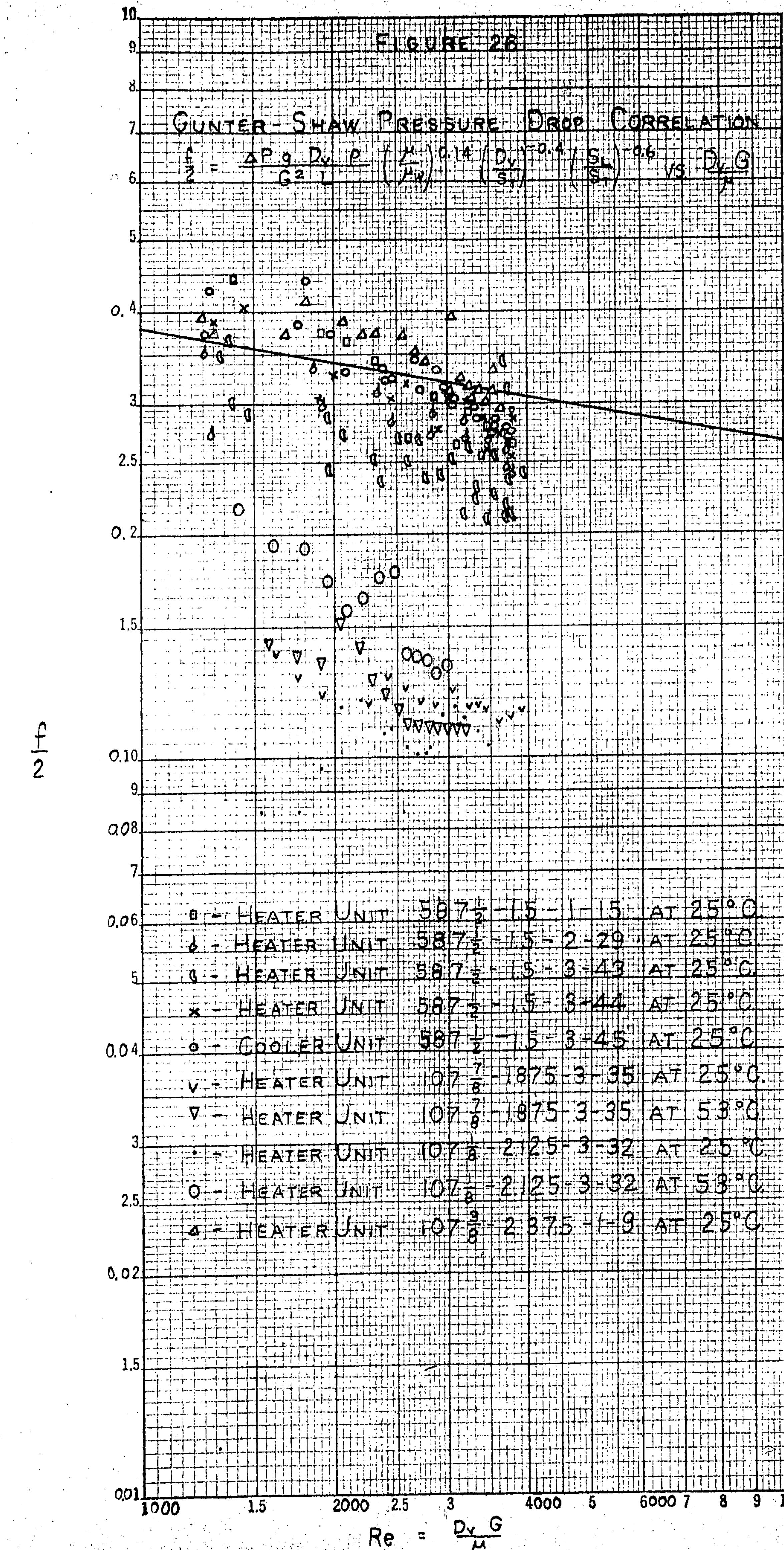


FIGURE 26
GUNTER-SHAW PRESSURE DROP CORRELATION
$\frac{f}{2} = \frac{\Delta P\, g\, D_v\, \rho}{G^2\, L}\left(\frac{\mu}{\mu_w}\right)^{0.14}\left(\frac{D_v}{S_T}\right)^{-0.4}\left(\frac{S_L}{S_T}\right)^{0.6}$ VS. $\frac{D_v G}{\mu}$
□ - HEATER UNIT 587½ - 1.5 - 1 - 15 AT 25°C
♁ - HEATER UNIT 587½ - 1.5 - 2 - 29 AT 25°C
◖ - HEATER UNIT 587½ - 1.5 - 3 - 43 AT 25°C
× - HEATER UNIT 587½ - 1.5 - 3 - 44 AT 25°C
o - COOLER UNIT 587½ - 1.5 - 3 - 45 AT 25°C
v - HEATER UNIT 107⅞ - 1875 - 3 - 35 AT 25°C
▽ - HEATER UNIT 107⅞ - 1875 - 3 - 35 AT 53°C
• - HEATER UNIT 107⅛ - 2125 - 3 - 32 AT 25°C
O - HEATER UNIT 107⅛ - 2125 - 3 - 32 AT 53°C
△ - HEATER UNIT 107⅜ - 2.375 - 1 - 9 AT 25°C
$\frac{f}{2}$
$Re = \frac{D_v G}{\mu}$

Discussion of Results for Gunter-Shaw Correlation

From Figure 26 it is seen that the Gunter-Shaw correlation fails to correlate pressure drop data for Tilco-type serrated-fin tubes in crossflow. The line drawn in Figure 26 is $\frac{f}{2} = 0.96\ Re^{-0.145}$, the correlation proposed by Gunter and Shaw for the turbulent range.

It is not surprising that the Gunter-Shaw correlation fails here. The correlation is proposed for three or more rows, while many of the cases considered here are for one or two rows. There is no allowance for the effect of baffles, and in these tests baffling is a major factor. The most significant conclusion to be drawn, however, is that use of the Gunter-Shaw configuration ratios does little or nothing to bring together data for five-eighths and one inch tubes.

Discussion of Prime Tube Size Correlation Theory

Since the Gunter-Shaw correlation fails to correlate data for Tilco-Fin tubes, the search was continued for a suitable correlation. It had been noted previously that the pressure drop per row for two and three row units of 587½ tubing was equal and that of one row of the same tubing was 1.19 times greater. Making use of this fact and of the observation that the pressure drop for five-eighths inch units is approximately 62% of the pressure drop for one inch units a correlation based on prime-tube size suggested itself. A "prime-tube size" correlation was tried and found to be better than the Gunter-Shaw correlation, even though it is far from perfect. This prime tube correlation in its final form is

$$f = \frac{\Delta P}{N}\,\frac{g}{V^2}\,D \quad = Re = \frac{DG'}{\mu} = \frac{DV\rho}{\mu}$$

where f = friction factor, dimensionless

ΔP = pressure drop due to friction, ft. of air

g = acceleration due to gravity, 32.17 ft./sec.^{2}

N = row factor, 1.19 for 1 row unit, 2.0 for 2 row units, 3.0 for 3 row units; ft.

V = air velocity, ft./sec.

ρ = air density, lb./ft.^{3}

μ = air viscosity at average temperature, lb./sec.ft.^{2}

G' = mass rate of air flow, = G/3600, lb./sec.ft.^{2}

Sample Calculation for Prime Tube Size Correlation

For Unit 107 7/8-1.875-3-35 at 25° C.

$\Delta H = 22.6$ in at $G = 2010$ lb./hr./ft.^{2}

$$f = \frac{\Delta P}{N}\frac{g\,D}{V^2} = \frac{\left(22.6 \times \frac{1}{12} \times \frac{1}{.0743 \times 62.4}\right)\left(32.17\right)\left(\frac{1}{12}\right)}{3 \times \left(\frac{2010}{3600} \times \frac{1}{.0743}\right)^2}$$

$$f = 2.49 \times 10^{-2}$$

$$Re = \frac{DG}{\mu} = \frac{\frac{1}{12} \times 2010}{0.0183 \times 2.42} = 3780$$

Table X

Prime Tube Size Pressure Drop Correlation

Unit 107-7/8-1.875-3-35 at 25° C.

Re	f
3780	2.49×10^{-2}
4075	2.33
4490	2.19
4860	2.09
5240	2.64
5610	2.535
5950	2.385
6290	2.14
6640	2.13
6980	2.11
7415	2.09
7670	2.11
7980	2.09
8340	2.01
8675	2.09
9000	2.055

FIGURE 27

PRIME TUBE SIZE

PRESSURE DROP CORRELATION

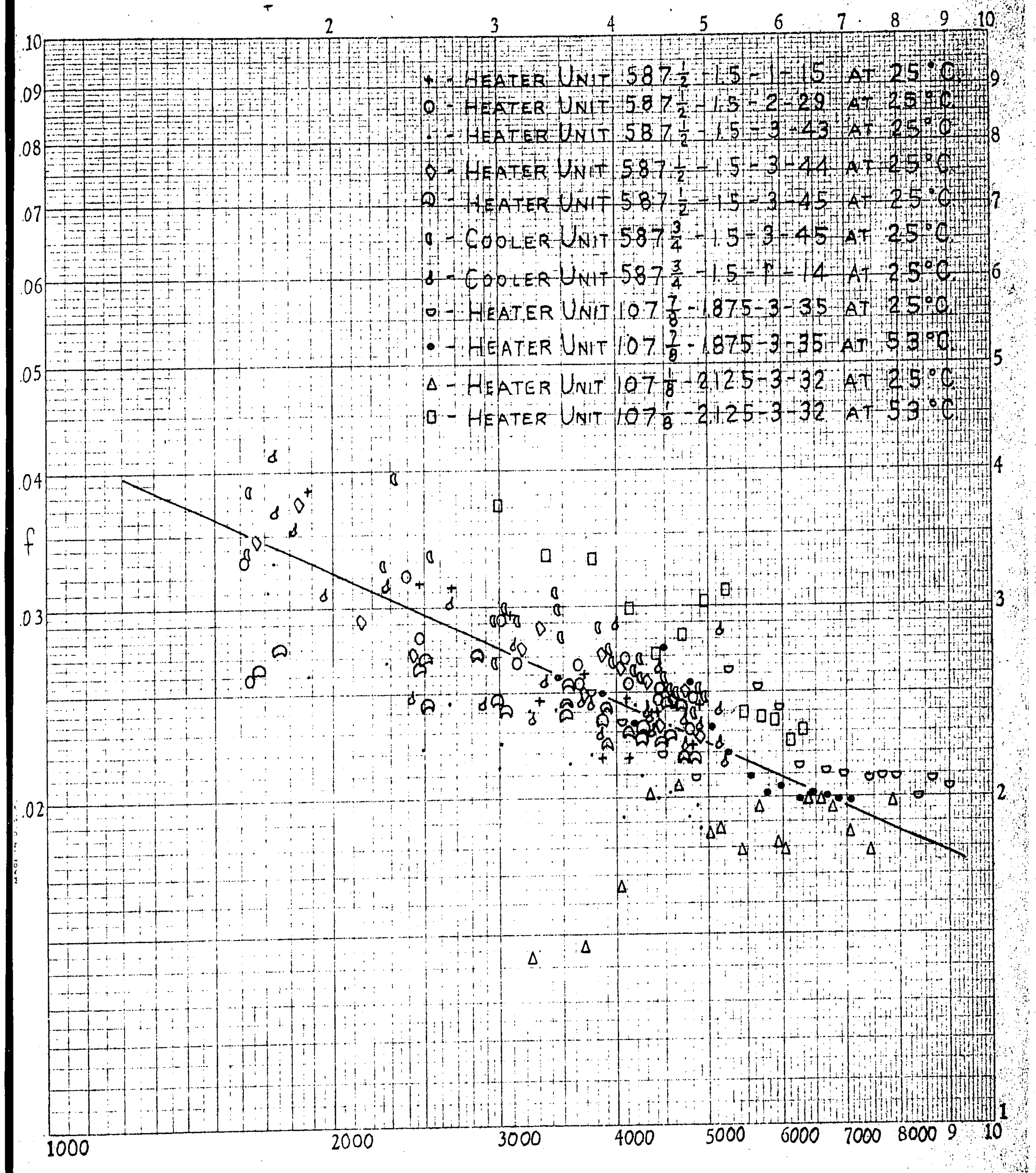
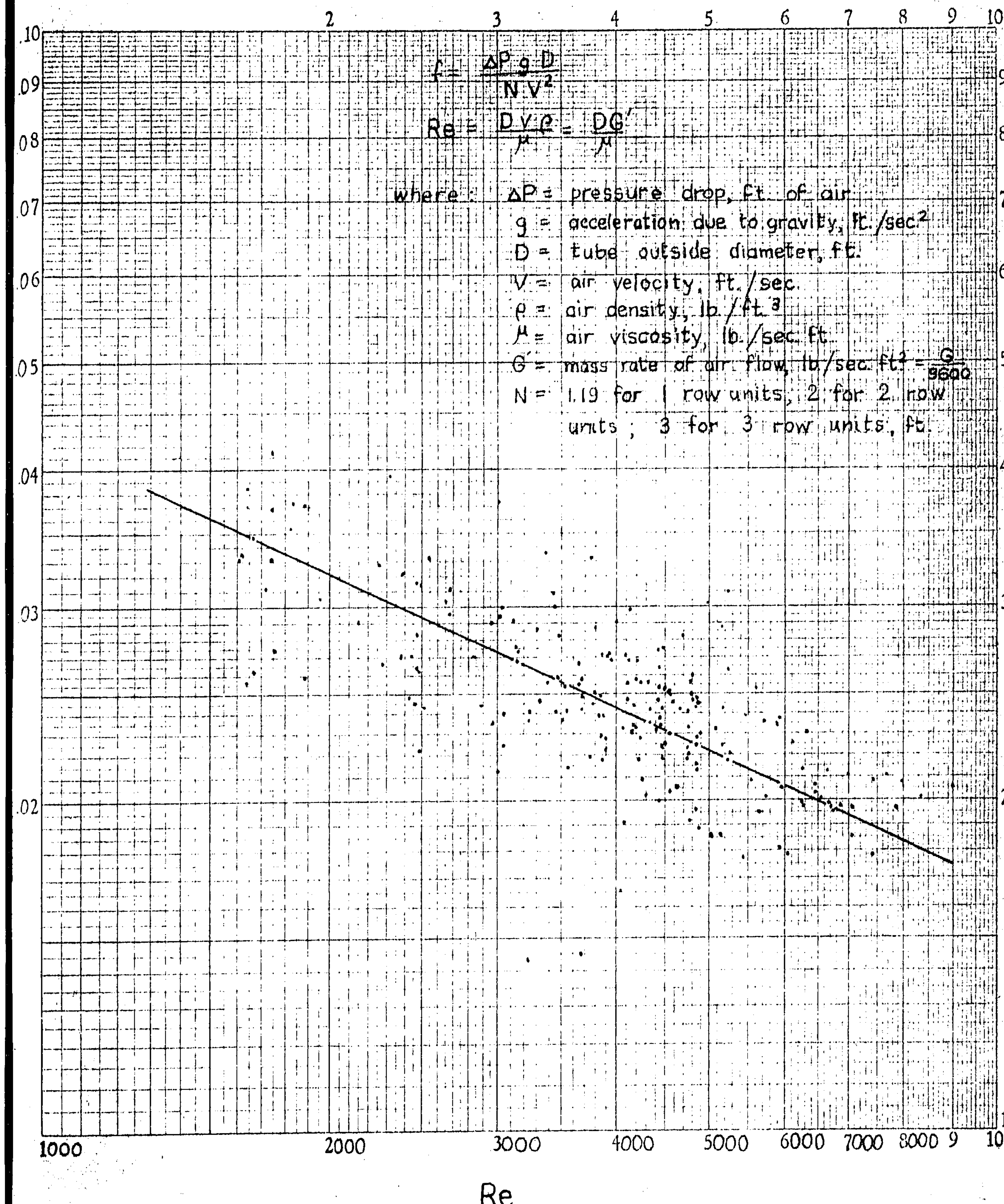


FIGURE 28

PRIME TUBE SIZE

PRESSURE DROP CORRELATION



Discussion of Results for Prime Tube Size Correlation

This "prime-tube size" correlation, analagous to the heat transfer correlation presented earlier in this report, works better than the Gunter-Shaw correlation. The average deviation is about 10 per cent. The equation of the line drawn in Figures 27 and 28 is $f = 0.66\ Re^{-0.40}$. It is to be noted that the high values of f occur mainly when the air is being heated, rather than in isothermal runs at 25 degrees Centigrade. Use of the Sieder-Tate correction factor, then, will help to bring these values more in line, but only by a factor of about two per cent, while some of the points deviate by as much as 35 per cent from the average. There are both positive and negative deviations for most units, and a great spread of the points for each unit, indicating that the original data are poor. It is difficult to correlate the pressure drop data since great differences in the pressure drop are caused by small changes in the arrangement of baffles or small changes in the number of tubes, e.g. between Units 587-1/2-1.5-3-43, 587-1/2-1.5-3-44, and 587-1/2-1.5-3-45, while adding another row of changing the tube or fin size has a relatively small effect on the pressure drop.

The pressure drop data for Unit 107-3/8-2.375-1-9 are not brought into line by this correlation. The values of f are high by a factor of approximately 2.5. This may be caused by: 1. an increased pressure drop resulting from increase of fin length, a factor omitted in the correlation; and/or 2. the failure of the constant $N = 1.19$ for one row which was determined for five-eighths inch tubes, to hold for one inch units.

Conclusions

As a result of the investigations reported here it may be concluded:

1. The performance of serrated-fin tubing under conditions of icing has been established.

2. The overall heat transfer coefficient for Wilco-Fins in the size range of ½ inch to 1 inch tubing, the customary range for air heating and cooling work, can be predicted with an expected error of less than five per cent by means of the heat transfer correlation presented. It seems likely that for copper tubes of similar construction this correlation may be applied to tube sizes up to several inches without a large [illegible] error.

3. The prime tube size pressure drop correlation can be applied to predict pressure losses of fin tube units in the range of ½ inch to 1 inch tubing, provided: (1) that the geometry of the tubing is similar to that of 587½ and 107 7/8 tubing, and (2) that the baffling is similar to that of the units studied. An error as great as thirty per cent may be expected.

4. As presently projected, investigation of larger size tubing, proposed for such uses as waste heat recovery from flue gases and oil cooling by air in regions of water scarcity, is the next logical step in the study of serrated-fin tubing.

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