ENGINEERING RESEARCH INSTITUTE

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Project M592

WOLVERINE TUBE DIVISION
CALUMET AND HECLA CONSOLIDATED COPPER COMPANY

August, 1953

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ABSTRACT

The lack of a generalized correlation for the outside convection coefficient of gases in crossflow through banks of finned tubes necessitated a study of the available data. The purpose of this work was to provide a procedure for the design of finned-tube banks. Such units have been used either to cool process water or to heat or cool gases.

The data for air are correlated by using the tube characteristics and arrangement of the tubes in the units to relate the outside heat-transfer coefficient and the pressure drop with the rate of flow of air.

The heat-transfer and pressure-drop correlations for air are generalized for any gas by using the physical properties of the fluid. Two sample calculations are included to illustrate the use of the proposed correlations.

TABLE OF CONTENTS

LIST OF FIGURES

LIST OF TABLES

CORRELATION OF HEAT TRANSFER AND PRESSURE DROP FOR AIR FLOWING ACROSS BANKS OF FINNED TUBES

Finned tubes are often used for transferring heat from a fluid inside the tubes to air flowing across a bank of the finned tubes. The purpose of these heat exchangers is either to cool the fluids inside or to heat the air outside the tubes. The size of such units varies from bus heaters and automobile radiators up to large industrial cooling units. The latter, which might be 24 feet square, are used to replace cooling towers in process plants.

At the time this work was started, no general correlation of data for any group of tubes was available in the literature.

For purposes of commercial design, individual companies often obtain experimental data on particular tubes and tube arrangements for specific installations. The recent paper by Schmidt[7] presents an equation of the outside heat-transfer coefficient for banks of finned tubes in terms of the coefficient for plain tubes, the fin height, and the number of fins per inch.

It is the purpose of this report to correlate available heat-transfer and pressure-drop data for air flowing across banks of finned tubes with helical fins. The data are on tubes of the type manufactured by the Wolverine Tube Division of the Calumet and Hecla Consolidated Copper Company. These tubes are constructed of copper, extruded by the Trufin process, or made of bimetallic construction with a liner metal which is different from that of the fin metal. Experimental data on 30 different units for air flowing outside the tubes have been provided by 11 organizations. In some cases complete test reports were provided, while in other cases only performance curves were made available.

Banks of plain tubes have been studied rather thoroughly by Grimison[1]. Data on specific banks of finned tubes have been reported by Katz, Beatty, and Foust[4] and recently by Kays and London.[5,6] These papers do not provide a correlation to allow the prediction of heat-transfer coefficients or pressure drop for finned tubes of a given dimension or for the effect of tube spacing and bundle arrangement in tube banks. Jameson[3] has reported heat-transfer and pressure-drop data obtained from extended surfaces made of fins soldered to bare tubes. Gunter and Shaw[2] have proposed a generalized pressure-drop correlation based on experimental data obtained from units using various types of extended surfaces, including helical fins.

This report proposes correlations for evaluating heat-transfer coefficients and pressure drops between air on the outside surface of finned tubes and water or steam on the inside of the tubes. Within the limit of the available dimensions of the tubes and arrangements in the 30 tests reported the various variables of fin spacing, fin height, and tube arrangement are investigated. In order that these data might be applied to fluids other than air in transferring heat to or from the outside of finned tubes, the correlation has been placed on a dimensionless basis. Such a correlation requires more careful data and includes the properties of the fluid at the mean bulk temperature of the fluid passing across the tube banks.

In addition to the correlations of heat-transfer coefficients and pressure drops, sample problems illustrating the use of the correlations for the design of a water cooler and of a hydrogen cooler are presented.

Data Available:

Data are available on 30 different units which employed a total of 27 different types of finned tubes. Most of the tube-bank units consisted of tubes on equilateral, triangular pitch. Some tube-banks had nonequilateral triangular tube arrangements and 3 units had square pitch layouts. The frontal face area varied from 0.605 sq ft to 480 sq ft. The 30 units referred to above represent data on banks of finned tubes of monometallic (copper) or bimetallic construction.

The dimensions of the various tubes on which the data are available and their arrangement in the tube banks are summarized in Tables I and II. Figure 1 illustrates the application and indicates the nature of the dimensions used. The available heat-transfer data are plotted as outside coefficients of heat transfer between the air and the outside surface of the tube as a function of the velocity of the air at the minimum cross-sectional area in Figures 2 through 6. This velocity is computed on the basis of air at 70°F and 50 per cent relative humidity. Wherever possible, actual data are shown. However, in some cases only performance curves were available (necessitating calculation of the individual heat-transfer coefficients). The range of the experimental data on both the heat-transfer coefficients and the velocity of the air are indicated on Figure 7. This plot summarizes the smooth curves of Figures 2 through 6. The lines in Figure 7 terminate at the reported minimum and maximum air velocities. The reported pressure-drop data are plotted versus the maximum air velocity on Figures 9, 10, and 11.

The effect of the number of vertical rows of tubes normal to the direction of flow on heat transfer and pressure drop should be mentioned. For plain tubes the added turbulence caused by the air passing the initial rows of tubes causes an increase in heat transfer in the first two or three rows.

Schmidt[7] studied the effect of the number of rows in finned-tube air heaters in which the fan was located downstream from the tubes. He reported heat-transfer coefficients as a function of the mass velocity raised to some power and he found that this power varied slightly with the number of rows. The constant of proportionality also varied with the number of rows of finned tubes. He states that for units having blowers upstream from the tubes the effect of the number of rows is less than when the fan is downstream. Jameson[3] studied units containing wrapped-on finned tubes with a blower located upstream so that there was a considerable calming section prior to the finned-tube unit. This is nearly equivalent to a downstream fan. Jameson also found some slight effect of the number of rows on the heat-transfer coefficient. The data studied in the present report show no effect of the number of rows of tubes in the unit. For a bank of finned tubes with an upstream blower there is little increase in heat transfer from the first to the second or later rows.

A considerable portion of the heat-transfer data were reported as overall coefficients of heat transfer between air on the outside of the tube and either water or steam inside the tubes. The outside film coefficients were evaluated from these overall coefficients. The inside film resistance and the metal resistance were subtracted from the overall resistance to heat-transfer in order to compute the outside film resistance, which is the reciprocal of the outside film coefficient. In the case of condensing steam the heat-transfer coefficient on the inside of the tube was estimated. A coefficient such as 1500 Btu/hr-°F-sq ft was assumed. For some of the data, a series of the reported fluid velocities on the inside of the tube were plotted against the overall resistance and extrapolated to an infinite velocity. This results in an overall resistance having no film resistance on the inside of the tubes (Wilson plot). It should be appreciated that these computations result in outside film coefficients which are higher numerically than the overall coefficient by 5 to 20 per cent. Thus, an overall coefficient of 15 Btu/hr-°F-sq ft might give a computed outside coefficient of 17 Btu/hr-°F-sq ft. Therefore, it follows that even though the conversion from overall coefficient to an outside film coefficient may include these approximations, the film coefficient error due to this is relatively small.

<u>Heat-Transfer Correlation For The Outside Film Coefficient</u>:

A correlation of the heat-transfer data involves description of the bank of finned tubes, properties of the flowing air, and the conditions of flow, such as temperature, pressure, and velocity. For convection between solid surfaces and a fluid, studies have shown that Equation 1 is normally suitable for correlating the data.

$$Nu = cRe^{m} Pr^{o}, \qquad (1)$$

where

Nu = Nusselt number;
Re = Reynolds number;
Pr = Prandtl number; and
c, m, and o = constants.

A definition of these dimensionless groups, used in Equation 1, is given in Equation 2:

$$\frac{hD}{k} = c\left(\frac{DG}{\mu}\right)^{m}\left(\frac{Cp\,\mu}{k}\right)^{o}, \qquad (2)$$

where

h = film coefficient;
D = diameter for single tubes;
k = thermal conductivity of fluid;
Cp = specific heat of fluid;
μ = viscosity of fluid; and
G = mass rate of fluid flow.

The heat-transfer coefficient between the air and the outside surface of the tube is included in the Nusselt number as the convection coefficient. The condition of flow of the fluid across the tube bank is indicated by the Reynolds number and includes an equivalent diameter for the tube bank, mass velocity of the air, and viscosity of the air. The thermal properties of the fluid passing through the tube bank are included in the Prandtl number as viscosity, specific heat, and thermal conductivity.

For tube banks containing plain tubes, the significant dimension which is used as D is either the diameter of the tube or the spacing between the tubes. Actually, it has become necessary to use different correlations depending on the spacing between the tubes. For finned tubes the problem is even more complex in that it is desirable to use a single dimension to represent the mechanical structure of the tubes and their arrangement in the tube bank. It must be admitted that any single dimension which is used is likely to be relatively empirical. To prove the validity of such a dimension, the only test which can be applied is that the use of this dimension in the general equation will bring test data on different tube banks into a single correlating relationship. It should be noted that different tube banks give different coefficients of heat-transfer at a given velocity, as indicated in Figures 2-6.

Since there are numerous variables in a tube bank such as number of fins per unit length, the fin diameter, the root diameter, and the pitch of the tube, the grouping of the variables to find an equivalent diameter must be rather arbitrary. After considerable study, the selected equivalent diameter is given by Equation 3, in which the equivalent diameter is for a tube bank composed of a specific kind of finned tube and is a function of the number of fins per inch, the root diameter, the fin diameter, and the spacing between the tubes based on an equilateral triangular arrangement. In case the longitudinal and transverse spacing of the tubes were not the same, an average of the two spacings was used.

$$D_e = \frac{(ND_o)}{(12)}\left(\frac{S}{D_r}\right)^2 \qquad (3)$$

where

N	=	number of fins per inch;
D_o	=	fin diameter, in.;
D_r	=	root diameter, in.; and
S	=	tube pitch for an equilateral triangular arrangement, in.

Correlation Of Heat Transfer From Air To Banks Of Finned Tubes:

In all the reported experimental data the air was being heated from approximately room temperature up to a value around 300°F. Under these conditions the Prandtl number of air, i.e., the viscosity of the air, the thermal conductivity of the air, and the heat capacity of the air do not vary appreciably, so that Equation 2 can be simplified to Equation 4 by introducing a constant average Prandtl number. It will be noted that in Figures 2-6, the maximum linear velocity was used in preference to the mass velocity; in other words, V_{max} was used instead of G_{max}.

$$h_o = f\ (D_e^{\,r},\ G^m), \qquad (4)$$

The exponents "r" and "m" as obtained from the correlation are -0.5 and 0.56 respectively. This results in Equation 5:

$$h_o = f\ (D_e^{\,-0.5},\ G_{max}^{0.56}) \qquad (5)$$

A further simplification of Equation 5 is possible by replacing the mass velocity by the maximum linear velocity based on air at a standard pressure, temperature, and humidity. Under standard conditions of 70°F, 1-atm. pressure, and 50% relative humidity, air has a density of 0.074 lb/cu ft. It should be noted that the maximum velocity is computed on the basis of the minimum free-flow area in the transverse direction. The data for individual units are plotted in Figures 2 through 6 as h_o versus V_{max}. In Figure 12, the data for all units, except units 11 through 18 and unit 22, are plotted as $h_o(S/D_r)(ND_o)^{0.5}$ versus V_{max}. The single line drawn through the points is represented by Equation 6. It is apparent that a group of the original data does not fit the curve. No adequate explanation has been found for the discrepancies between the results of the various groups of tube banks. It is believed that the points which lie below the average curve indicate the presence of some resistance, either on the inside of the tube or in the compound tube wall, that is not properly accounted for. The curve on Figure 12 is based on tube banks having triangular staggered pitches.

$$h_o = 1.9 \frac{(D_r)}{(S(N D_o)^{0.5})} V_{max}^{0.56} . \qquad (6)$$

Equation 6 represents a correlation of the bulk of the heat-transfer data for air flowing across banks of finned tubes in units having triangular tube arrangements. The outside coefficient is a function of the tube dimensions, the tube spacing, and the maximum velocity through the minimum free transverse area. When it is necessary to determine the outside coefficient for a unit having similar tubes and spacing to the units represented on Figures 2 through 6 it is advisable to use the curves on these figures. However, when it is desired to find the effect on the heat-transfer coefficient of changing the tube dimensions or the tube spacing, Equation 6 is recommended. Equation 6 is also recommended when determining coefficients for tubes and spacings not covered by the original test data which are included in this report.

An equally good correlation of the data was obtained by plotting the group h_o (S/D_r) $(ND_o)^{0.5}$ versus the face velocity. Table II has been included to facilitate the conversion of maximum velocity to face velocity. The second column of Table II gives the ratio of the free-flow area to the face area for each unit studied. The product of this ratio and V_{max} gives the corresponding face velocity. The third column of Table II gives the ratio of outside to inside surface area for each unit and one can then convert the heat-transfer coefficient based on outside surface to one based on inside surface if it is desired.

The meager data available for tube arrangements other than triangular staggered pitch are presented in Figure 13, together with triangular arrangements not indicated in Figure 12. Equation 6 is also plotted on this figure.

Generalized Heat-Transfer Correlation:

The data on air have been computed in the form of a generalized dimensionless correlation of heat transfer, employing a modification of Equation 2. Most of the reported data were in the form of inlet and outlet bulk air temperatures, so that the properties of the air could be employed in the correlation. In the case of such available information, it was possible to compute the Prandtl number, the Nusselt number, and the Reynolds number using the equivalent diameter given in Equation 3. Accordingly, these calculated groups have resulted in the correlation indicated on Figure 14. It should be noted that those points which did not fit the curve on Figure 12, likewise, fall in a similar position on Figure 14.

The data were not sufficient to check the validity of the 1/3 power assumed for the Prandtl number. A viscosity ratio was not used in the correlation because the fluid was gaseous and the effect of temperature on the viscosity was relatively small. Figure 14 may now be used for predicting heat transfer coefficients for any fluid for which the properties are known at the mean bulk temperature in the tube bank.

Correlation of Pressure-Drop Data:

As in the case of heat-transfer, the pressure-drop data are correlated in a simplified form in which the properties of the air are inherent in the correlation and in a generalized form in which the properties of the fluid must be known to obtain the pressure drop.

The individual pressure-drop data as plotted in Figures 9, 10, and 11 show the variation of the pressure-drop in inches of water per row with the maximum air velocity. Figure 8 indicates that pressure-drop varies as the 1/2 power of the number of fins per inch for constant maximum air velocity and physical characteristics of the tube bank.

From theoretical considerations, the pressure drop due to fluid flow past solid surfaces is given by Equation 7:

$$\frac{\Delta P}{\rho} = \frac{f(Re)\, V^2 L}{2 g_c\, D}, \qquad (7)$$

where

ΔP = pressure drop;
ρ = fluid density;
V = fluid velocity;
L = length of flow path;

D = diameter for a single tube;
$f(Re)$ = friction factor; and
g_c = gravitational constant.

For plain-tube banks, D is usually taken as the spacing between the tubes. The function of the Reynolds number is usually referred to as the "friction factor" and varies inversely as the 0.2 power of the Reynolds number, in which D is the length dimension used in Equation 7. In the following correlation the equivalent diameter of the finned tube bank as defined by Equation 3 is substituted for D in Equation 7. Within the range of temperatures studied, the density and viscosity of air are assumed to be constant, whereby Equation 8 is obtained as the simplified form of Equation 7:

$$\Delta P = \frac{F\, D_e^t\ V_{max.}^u}{D_e} \tag{8}$$

where

ΔP = pressure drop, inches water/row; and
F = constant including the physical properties of the fluid.

This equation is the basis for correlating the pressure-drop data in terms of $\Delta P D_e$ as a function of the maximum air velocity as indicated in Figure 15. A parameter, including the root diameter of the tube, the diameter over the fins, and the number of fins per inch as $K = ND_o/D_r^{0.2}$ is also used to complete the pressure-drop correlation for air.

To obtain a generalized pressure-drop correlation for gases, Equation 7 must be modified to include the properties of the fluid in addition to the tube-bank characteristics. Thus Equation 7 may be written as follows:

$$\frac{\Delta P\, D_e}{\rho V_{max}^2} = f(Re) \tag{9}$$

Using the properties of air at the average temperature in the tube bank available from 24 units, the "friction factor" function of the Reynolds number (f) was calculated and plotted against the Reynolds number in Figure 16. This generalized correlation includes also the parameter K, defined previously. Figure 16 indicates a break in the curves at a Reynolds number of 300,000, possibly due to the transition from laminar to turbulent flow. Therefore, for a given set of conditions in any finned-tube bank, the pressure-drop can be evaluated from this figure and the properties of the flowing fluid.

Bond Resistance In Bimetal Finned Tubes:

In the construction of Bimetal finned tubes the liner and the root metal are pressed against one another in the fabrication process. This tends to give a tight metal-to-metal bond. From heat-transfer tests on banks of finned tubes, it has been observed that the tightness of this Bimetal bond varies during the manufacturing process.

The effect of the bond resistance in Bimetal finned tubing is thought to vary up to about 10 percent of the total resistance to heat transfer for low heat fluxes such as for air. This resistance is inherent in the data taken with copper-aluminum Bimetal finned tubing and was not considered in the correlation of data for this report. Therefore, the heat-transfer coefficients reported for Bimetal tubing are already adjusted for the bond resistance which is inherent in the tubing.

Bond resistance is a factor which must be considered in the design of heating units made of Bimetal tubing and in some instances[6] it may be fairly large. Work is in progress on this problem and it is considered that at high heat fluxes, bond resistance should not be neglected. However, at low heat fluxes such as are encountered in the heating of air, no correction for bond resistance is necessary with the correlations in this report.

The Effect Of Tube Spacing:

Equation 6 indicates that the heat-transfer coefficient for air is inversely proportional to the tube spacing if all other variables remain constant. This is shown also by the test data for units 12 and 13. These units are identical except that the tube spacings are 0.875 and 1.0 inch respectively. Figure 3 shows that unit 13 has a coefficient about 12 percent below that of unit 12. If the test data are plotted as $h_o(S/D_r)(ND_o)^{0.5}$ versus V_{max}, the points for these two units may be adequately represented by a single curve.

The quantitative effect of tube spacing may be shown by the following calculations for a Bimetal finned tube with

D_o	=	2.00 inches	A_o	=	3.61 sq ft/ft
D_r	=	1.070 inches	A_o/A_i	=	14.8
N	=	9 fins/inch	K	=	17.75

For S = 2.00 inches (fins touching), triangular arrangement D_e = 5.26 ft

Free-flow area/sq ft face area = 0.385 sq ft
Outside surface/sq ft face area = 21.6 sq ft

$S/D_r \, (ND_o)^{0.5} = 7.94$

For a face velocity of 500 ft/min, V_{max} = 500/0.385 = 1300 ft/min

From Figure 12 $h_o \, (S/D_r) \, (ND_o)^{0.5} = 105.6$

Thus h_o = 105.6/7.94 = 13.3 Btu/hr sq ft °F

Heat transfer/sq ft face area = (13.3) (21.6) = 288 Btu/(hr) (°F) (sq ft face) per row of tubes.

From Figure 15 ΔP = 0.16 in water/row

For S = 2.25 inches, triangular spacing

D_e = 6.69 ft

Free-flow area/sq ft face area = 0.452 sq ft
Outside surface/sq ft face area = 19.2 sq ft

$S/D_r \, (ND_o)^{0.5} = 8.93$

For a face velocity of 500 ft/min, V_{max} = 500/0.452 = 1106 ft/min

From Figure 12, $h_o \, (S/D_r) \, (ND_o)^{0.5} = 96.5$

Thus h_o = 96.5/8.93 = 10.8 Btu/hr °F sq ft

Heat transfer/sq ft face area = (10.8) (19.2) = 207 Btu/hr °F sq ft face area per row of tubes.

From Figure 15, ΔP = 0.10 in water/row

For each row of tubes, increasing the tube spacing from 2.00 to 2.25 inches decreases the rate of heat transfer by 28 percent and decreases the power requirements by 37 percent. The tube bank with the larger tube spacing will require more rows of tubes to transfer the same amount of heat as the bank with the smaller tube spacing. From the standpoint of heat transfer alone, the tubes should be as closely spaced as possible.

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APPENDIX

OTHER CORRELATIONS CONSIDERED

Heat-Transfer Correlation:

Figure 17 shows the grouping of heat-transfer data for air on the basis of the recently proposed Schmidt correlation[7]. The outside coefficient for banks of finned tubes is related to the tube dimensions and the corresponding outside coefficient for a bank of plain tubes by the following equation:

$$\frac{h_f}{h_p} = 1-0.18\left[\left(\frac{D_o - D_r}{2}\right) N\right]^{0.63} \qquad (10)$$

where

h_f = outside convection coefficient for finned-tube bank;
h_p = outside convection coefficient for plain-tube bank;
D_o = diameter over the fins, inch;
D_r = root diameter, inch; and
N = fins/inch.

A similar distribution of data for air is obtained by the simple empirical correlation shown in Figure 18. The diameter over the fins (D_o) is eliminated because of the constant ratio of D_o/D_r that was observed within the limits of the available data. The equation fitting the data best on Figure 18 is:

$$h_o = \frac{0.711}{S}\left(\frac{D_r}{N}\right)^{0.3} V_{max}^{0.56} \qquad (11)$$

These correlations may be useful in some applications with air.

Pressure-Drop Correlation

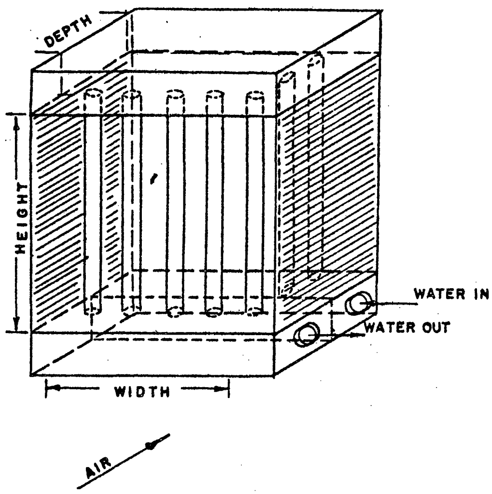
Empirical correlations for the pressure-drop data are shown in Figures 19 and 20 using a different set of dimensions to describe the finned tubes. The correlation for air is satisfactory and could be used advantageously as an alternate to Figure 15. The grouping of data in Figure 20 is not favorable for the generalized pressure-drop correlation, but is shown to indicate how the data spread when these variables are used.

SAMPLE DESIGN CALCULATIONS

Two sample calculations are given to illustrate the use of the recommended heat-transfer and pressure-drop correlations for air as well as for other gases for the design of finned tube heaters and coolers. The first example is of a heat exchanger for cooling water with air. The second example illustrates similar calculations for a hydrogen cooler using the generalized correlations.

Example I.

It is required to cool 400,000 lbs/hr of water from 125°F to 115°F with air in cross flow being heated from 70°F to 110°F on the outside of a bank of finned tubes.



WATER COOLER

Sample Design Calculations

Cooling water with air:

$T_1 = 70°F \qquad T_2 = 110°F$
$t_2 = 115°F \qquad t_1 = 125°F$

Heat Load = 4,000,000 Btu/hr
Heat Exchanger-- 2 tube passes - 1 shell pass.
Tube Specifications-- Wolverine Trufin Copper tube
Catalog No. 0901
D_o = 1.030 in.
D_i = 0.384 in. or 0.032 ft.
Tube wall thickness = 0.038 in.
y = mean fin thickness = 0.019 in.
N = 9 fins/in.
A_o/A_i = 10.8
D_r = 0.384 + (2) (0.038) = 0.460 in.

$$A_i = \frac{(3.14)\ (0.384)}{(12)} = 0.1005 \text{ sq. ft/ft}$$

$$A_o = (0.1005)(10.8) = 1.086 \text{ sq. ft/ft}$$

1st Approximation -- Assume $U_o = 10$: $q = U_o A \Delta T_{LM}$

$$\Delta T_{LM} = \frac{45 - 15}{\ln(45/15)} = 27.3°F$$

A_t, total heat-transfer area $= \dfrac{(4 \times 10^6)}{(10)(27.3)} = 14{,}650$ sq. ft.

L_t, total tube length $= \dfrac{14{,}650}{1.086} = 13{,}500$ ft

Assume tube size and arrangement:

Length of each tube = 20 ft/tube, and 34 tubes/row

n, number of rows $= \dfrac{13{,}500}{(20)(34)} = 19.85$ or 20 staggered rows on equilateral triangular pitch.

Calculation of V_{max} air. (minimum channel-area velocity)

$C_p$air = 0.25 Btu/lb°F at avg air condition

$\Delta T_{air} = T_2 - T_1 = 110 - 70 = 40°F$

$W_{air} = \frac{Q}{Cp\Delta T} = \frac{4,000,000}{(0.25)(40)} = 400,000$ lbs/hr

Specific volume of Std. air (50% R.H. and 70°F) $= 13.5 \frac{ft^3}{lb}$

volume $= \frac{(400,000)(13.5)}{(60)} = 90,000$ cu ft/min

With a clearance of 0.125 in. between fin edges, (equilateral triangular pitch of 1.155 in.):

Width of Unit normal to flow

$= \frac{(1.030 + 0.125)(34)}{(12)} = 3.27$ ft

Unit height = 20 ft

Total face area = (20)(3.27) = 65.5 sq ft

Projected fin area per tube = a_{fins}

$= (\text{fin thickness})(D_o - D_r)(N)(\text{tube length}) \frac{(12)}{144}$

$= \frac{(0.019)(1.030 - 0.460)(9)(20)(12)}{(144)}$

$= 0.163$ sq ft/tube

Projected tube area excluding fins = a_{root}

$= \frac{(20)(0.460)}{(12)} = 0.767$ sq ft/tube

Total projected area per tube = a_t

$= 0.163 + 0.767 = 0.930$ sq ft/tube

Total face area of tube metal (projected area per row)

$= a_{tF} = (0.930)(34) = 31.6$ sq ft.

a_{FF}, free face area = total area - obstructing area

$= 65.5 - 31.6 = 33.9$ sq ft

$V_{max} = \frac{90,000}{33.9} = 2650$ ft/min

This linear air velocity is excessively high. To reduce the air velocity to a reasonable value of about 1/2 of this velocity, revision of the assumed tube bundle arrangement is necessary.

$S = 0.125 + 1.030 = 1.155$ in.

90 tubes/row, 15-ft tubes

$$\text{Number of rows, } n = \frac{(13{,}500)}{90 \times 15} = 10$$

$$\text{Width of unit} = \frac{(90)(1.155)}{(12)} = 8.66 \text{ ft}$$

Height of unit = 15 ft
Total face area = (15 x 8.66) = 130 sq ft

$$A_{FF} = 130 - \frac{(0.93)(15)(90)}{(20)} = 67.3 \text{ sq ft}$$

$$V_{max.} = \frac{90{,}000}{67.3} = 1340 \text{ ft/min} = 1.34 \times 10^3 \text{ ft/min}$$

From Figure 12, $(h_o\ S/D_r)(ND_o)^{0.5} = 108 = h_o\ D_e^{0.5}$

$$De^{0.5} = (S/D_r)(ND_o)^{0.5} = (1.155/0.460)(9 \times 1.030)^{0.5}$$

$= 7.65$ (Note unit of D_e in Figure 12, in.)

$h_o = 108/7.65 = 14.1$ Btu/hr-°F-sq ft outside area

<u>Metal</u> <u>Resistance</u>:

$$r_m = L_{Cu}\ A_o/k_{Cu}\ A_{avg}$$

where: r_m = metal resistance;
L_{Cu} = thickness of copper;
A_o = outside area of tube;
k_{Cu} = thermal conductivity of copper = 220; and
A_{avg} = average heat transfer area of tube.

d_e, tube equivalent diameter to calculate A_{avg} and L_{Cu}, is:

$$d_e = D_r + (D_o - D_r)(N)(y)$$

$$d_e = 0.460 + (1.030 - 0.460)(9)(0.019) = 0.5575 \text{ in.}$$
$$= 0.0465 \text{ ft}$$

$$L_{Cu} = \frac{0.0465 - 0.032}{2} = 0.0073 \text{ ft}$$

D_{LM}, log-mean diameter

$$= \frac{0.5575 - 0.384}{\ln \frac{0.5575}{0.384}} = 0.466 \text{ in.} = 0.0389 \text{ ft}$$

$A_{avg} = (3.14)(D_{LM})$

$= (3.14)(0.0389) = 0.122$ sq ft/ft length of tube

r_m = metal resistance = $\dfrac{(0.0073)(1.086)}{(220)(0.122)}$

$= 0.000295$ °F-hr-sq ft/Btu

Calculation of the Inside Water-Film Resistance:

$$w_{water} = \frac{Q}{C_p\ \Delta T}$$

$$\Delta T_{water} = (t_1 - t_2) = 125 - 115 = 10°F$$

$$C_p = 1.0 \text{ Btu/lb-°F}$$

$$w_{water} = \frac{4,000,000}{(10)(1)} = 400,000 \text{ lbs/hr}$$

a_i, total internal cross-sectional area per pass,

$$= (3.14)\left(\frac{D_1^2}{4}\right)\left(\frac{1}{144}\right) \text{ (total no. tubes per pass)}$$

$$= \frac{(3.14)(0.384)^2\ (5)(90)}{(4)(144)} = 0.364 \text{ sq ft}$$

V_{water} = water velocity = $\dfrac{400,000}{(3600)(62.4)(0.364)}$

$= 4.90$ ft/sec

Using Equation 9c, p. 183 in McAdams, 2nd edition,

$$h_i = 150\ (1 + 0.011t)(V')^{0.8}/(D_i)^{0.2}$$

$$h_{water} = (150)(1 + 0.011 \times 120)\ \frac{(4.90)^{0.8}}{(0.384)^{0.2}}$$

$= 1510$ Btu/hr - °F-sq ft inside area, where average water temperature = 120°F

r_{water} = film resistance = $A_o/h_{water}\ A_i = 10.8/1510 = .0075$

f_i = fouling factor (water) = $.001\ \dfrac{°F\text{-hr-ft}^2}{Btu}$ (TEMA)

$f_iA_o/A_i = .001(10.8) = .0108$

Calculation of the actual overall coefficient, U_o:

$$1/U_o = r_{air} + r_{metal} + f_i + r_{water}$$

$$= 0.071 + 0.0003 + 0.0108 + 0.0072$$

$$= 0.0893$$

U_o = 11.2 Btu/hr-°F-sq ft outside, as opposed to an assumed value of 10.0.

From P. 289, Trans. A.S.M.E. (1940),
correction to LMTD = 0.992 (from Figure 10)
Actual LMTD = (0.992)(27.3) = 27.1°F

Required heat-transfer area $= \dfrac{4{,}000{,}000}{(11.2)(27.1)} = 13{,}200$ sq ft

Total required tube length $= \dfrac{13{,}200}{1.086} = 12{,}150$ ft

Actual tube length used = (90)(15)(10) = 13,500 ft

Excess heat-transfer area $= \dfrac{(13{,}500 - 12{,}150)}{12{,}150}(100) = 11.1\%$

Pressure-Drop Calculation:

Using the Figure 15 correlation, where

$$D_e = (S/D_r)^2 \frac{(ND_o)}{12} = \frac{(7.65)^2}{12} = 4.9 \text{ ft},$$

$$K = ND_o/D_r^{0.2}$$

$$= \frac{(9)(1.030)}{(0.460)^{0.2}} = 10.8$$

From Figure 15, at V_{max} = 1340 ft/min

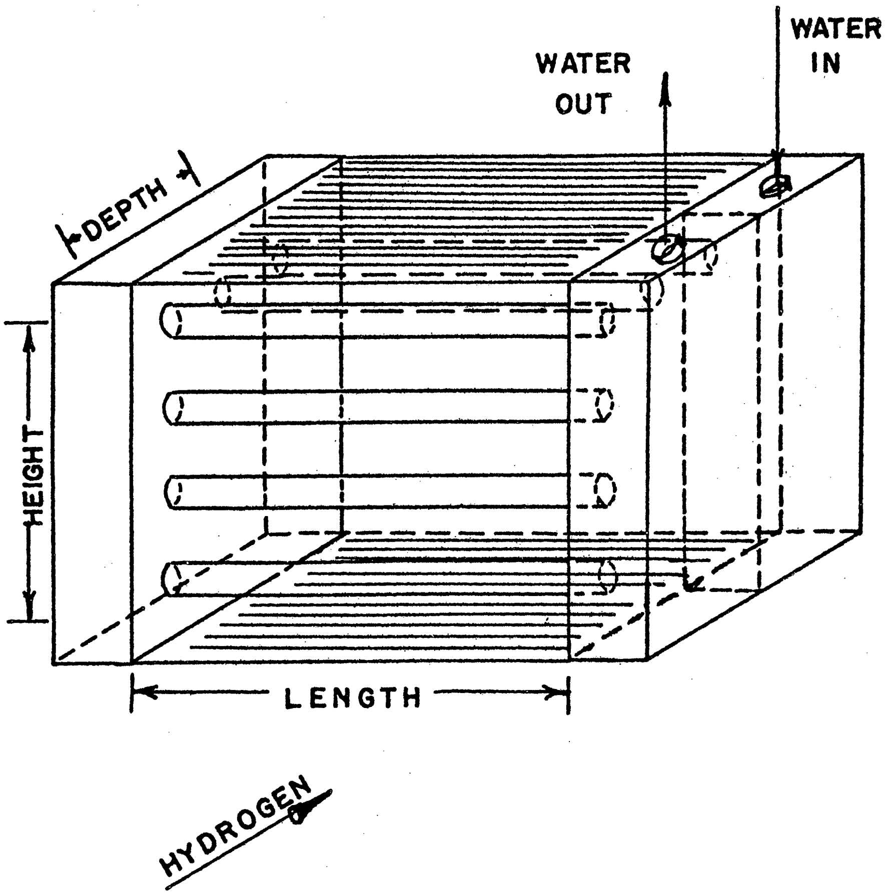
$$\Delta PD_e = 0.440, \therefore \Delta P = 0.440/4.9 = 0.0898 \ \frac{\text{in. water}}{\text{row}}$$

Total pressure drop = (0.0898)(10)

= 0.898 in. water or 0.0325 lb/sq in.

Example II.

It is required to cool 28,600 lb/hr of hydrogen from 150°F to 100°F with water being heated from 90°F to 105°F inside the tubes of a finned-tube cooler.



HYDROGEN COOLER

Cooling Hydrogen with Water:

T_1 = 150°F T_2 = 100°F

t_2 = 105°F t_1 = 90°F

Heat Load = 5,000,000 Btu/hr

Heat Exchanger -- 2 tube passes - 1 shell pass

Tube Specifications -- Wolverine Trufin Bimetal tube

Catalog No. 1106

D_o = 1.272 in.

D_i = 0.504 in.

Tube wall thickness, total = 0.043 in.
copper liner thickness = 0.028 in.

y = mean fin thickness = 0.015 in.

N = 11.0 fins/in.

$A_o/A_i = 16.2$

$D_r = 0.504 + (2)(0.043) = 0.590$ in.

$$A_i = \frac{(3.14)(0.504)}{(12)} = 0.132 \text{ sq ft/ft}$$

$$A_o = (0.132)(16.2) = 2.14 \text{ sq ft/ft}$$

Assume $U_o = 10$: $q = U_o A \Delta T_{LM}$

$$\text{LMTD} = \frac{45 - 10}{\ln \frac{45}{10}} = 23.3°F$$

Approximate total heat-transfer area

$$A = \frac{Q}{U_o \Delta T_{LM}} = \frac{5{,}000{,}000}{(10)(23.3)} = 21{,}500 \text{ sq ft}$$

$$\text{Total tube length} = \frac{21{,}500}{2.14} = 10{,}040 \text{ ft}$$

Assume bank of tubes 10 rows deep

$$\therefore \frac{(10{,}040)}{(10)} = 1004 \text{ ft/row}$$

(with 12-ft tubes, 84 tubes/row)

C_p, specific heat of hydrogen = 3.5 Btu/lb-°F

$$w, \text{hydrogen} = \frac{5{,}000{,}000}{(50)(3.5)} = 28{,}600 \text{ lbs/hr}$$

$$\text{Density of hydrogen} = (\text{density of Std. air}) \frac{(\text{Mol wt } H_2)}{(\text{Mol wt air})}$$

$$= \frac{(0.074)(2.02)}{(29)} = 0.00515 \text{ lb/cu ft}$$

$$\text{Volume of hydrogen} = \frac{28{,}600}{(0.00515)(60)} = 92{,}500 \text{ cu ft/min}$$

S = pitch = 1.75 in.; tube clearance = 0.428 in.

$$\text{Height of unit} = \frac{(84)(1.75)}{(12)} = 12.25 \text{ ft}$$

Length of unit = 12 ft

Total face area = (12)(12.25) = 147 sq ft

$$a_{fins} = \frac{(0.015)(1.272 - 0.590)(11)(12)(12)}{(144)}$$

$$= 0.1127 \text{ sq ft/tube}$$

$$a_{root} = \frac{(12)(0.590)}{(12)} = 0.590 \text{ sq ft/tube}$$

a_{tF}, total projected area

$$= (0.1127 + 0.590)(84) = 59.0 \text{ sq ft}$$

$$a_{FF} = 147 - 59 = 88 \text{ sq ft}$$

$$V_{max} = \frac{92{,}500}{88} = 1050 \text{ ft/min}$$

$$G_{max} = \frac{28{,}600}{88} = 325 \text{ lbs/hr-sq ft}$$

$$D_e = (S/D_r)^2 (ND_o/12) = (1.75/0.590)^2 \frac{(11 \times 1.272)}{(12)}$$

$$= 10.28 \text{ ft}$$

At t_{avg} of hydrogen = 125°F

μ = 0.0227 lb/ft-hr
k = 0.115 Btu-ft/hr-°F-sq ft
C_p = 3.5 Btu/lb-°F

$$Re = G_{max} D_e/\mu = \frac{(325)(10.28)}{(0.0227)} = 147{,}000$$

From Figure 14, $\frac{h_o D_e}{k} \Big/ \left(\frac{C_p \mu}{k}\right)^{0.333} = 1{,}225$

and $(C_p \mu/k)^{0.333} = (3.5 \times 0.0227/0.115)^{0.333} = 0.884$

$$\text{Therefore } h_o = \frac{(1225)(0.884)(0.115)}{(10.28)}$$

$$= 12.1 \text{ Btu/hr-°F-sq ft outside}$$

r_H, hydrogen = 1/12.1 = 0.0826

TEMA fouling factor = 0.001

r_f = (0.001)(16.2) = 0.0162

Metal Resistance:

$$r_m = L_{Al}A_o/k_{AL}A_{avg\ Al} + L_{Cu}A_o/k_{Cu}A_{avg\ Cu}$$

where: r_m = metal resistance,
L = thickness of metal,
A_o = outside area of tube,
k = thermal conductivity, and
A_{avg} = average heat transfer area of tube.

$$L_{Al} = \frac{0.0585 - 0.048}{2} = 0.0053 \text{ ft}$$

$$D_{LM}\text{, log-mean diameter} = \frac{0.0585 - 0.048}{\ln \frac{0.0585}{0.048}} = .0528 \text{ ft}$$

$$A_{avg\ Al} = (3.14)(0.0528) = 0.166 \text{ sq ft/ft}$$

$$L_{Cu} = \frac{0.048 - 0.042}{2} = 0.003 \text{ ft}$$

$$D_{LM} \text{ of copper liner} = \frac{0.048 - 0.042}{\ln \frac{0.048}{0.042}} = 0.0451 \text{ ft.}$$

$$A_{avg\ Cu} = (3.14)(0.0451) = 0.142 \text{ sq ft/ft}$$

$$r_m = r_{Al} + r_{Cu}$$

$$r_m = \frac{(0.0053)(2.14)}{(110)(0.166)} + \frac{(0.003)(2.14)}{(220)(0.142)} = 0.000831$$

Calculation of the Water-Film Resistance.

$$w = \frac{Q}{C_p\ \Delta T} = \frac{5{,}000{,}000.}{(1)(15)} = 333{,}000 \text{ lbs/hr}$$

a_i, total internal cross-sectional water flow area/pass

$$= \frac{(3.14)(0.504)^2(5)(84)}{(4)(144)} = 0.581 \text{ sq ft}$$

$$V_{water} = \frac{333{,}000}{(3600)(62.4)(0.581)} = 2.56 \text{ ft/sec}$$

From - Equation 9c in McAdams, p. 183, at t_{avg} = 97.5°F,

$$h_i = 150(1.0 + 0.011(97.5)) \frac{(2.56)^{0.8}}{(.504)^{0.2}}$$

$$= 754 \text{ Btu/hr-°F-sq ft inside}$$

Calculation of the actual U_o:

$$1/U_o = 0.0826 + 0.000831 + 0.0162 + 0.0215 = 0.1211$$

correction to LMTD for cross flow is negligible.

$$U_o = 1/0.1211 = 8.25 \text{ Btu/hr-°F-sq ft outside}$$

$$\text{Required heat-transfer area} = \frac{5{,}000{,}000}{(8.25)(23.2)} = 26{,}100 \text{ sq ft}$$

$$\text{Required tube length} = \frac{26{,}100}{2.14} = 12{,}200 \text{ ft}$$

$$\text{Actual tube length used} = (12)(84)(10) = 10{,}080 \text{ ft}$$

Add two vertical rows ∴ 12 rows deep.

2nd Approximation:

4 water passes and one hydrogen pass unit
84 horizontal tubes in one vertical row
12 vertical rows deep
No modification in tube size or pitch
The hydrogen gas-film coefficient remains unchanged

Revised Water-Side Coefficient:

a_i, total internal cross-sectional water-flow area

$$= \frac{(3.14)(0.504)^2(3)(84)}{(4)(144)} = 0.35 \text{ sq ft}$$

$$V_{water} = \frac{333{,}000}{(3600)(62.4)(0.35)} = 4.25 \text{ ft/sec}$$

$$h_i = 150(1.0 + (0.011)(97.5)) \frac{(4.25)^{0.8}}{(0.504)^{0.2}} = 1130 \text{ Btu/hr-°F-sq ft inside}$$

$$r_{water} = \frac{16.2}{1130} = 0.01435$$

Overall Coefficient:

$$1/U_o = 0.0826 + 0.000831 + 0.0162 + 0.01435 = 0.114$$

$U_o = 1/0.114 = 8.76$ Btu/hr-°F-sq ft outside

Required heat-transfer area

$$= \frac{5,000,000}{(8.76)(23.3)} = 24,500 \text{ sq ft}$$

Required tube length $= \frac{24,500}{2.14} = 11,440$ ft

Actual tube length used $= (12)(84)(12) = 12,100$ ft

Excess heat-transfer area $= \frac{(12,100 - 11,440)}{11,440}(100) = 5.76\%$

Pressure-Drop Calculation:

$D_e = 10.28$ ft

$$ND_o/D_r^{0.2} = \frac{(11)(1.272)}{(0.590)^{0.2}} = 15.5$$

Re = 147,000

$V_{max} = 1050$ ft/min

$\rho = 0.00515$ lb/cu ft

$V^2_{max} = (1050)^2 = 1.1 \times 10^6$ sq ft/sq min

From Figure 16, $\frac{\Delta P D_e}{\rho V^2_{max}} = 7.8 \times 10^{-6}$

$$\text{Therefore } \Delta P = \frac{(7.8 \times 10^{-6})(0.00515)(1.1 \times 10^6)}{10.28}$$

$= 0.0043$ in. water/row

Total pressure drop $= (12)(0.0043) = 0.0515$ in. water

$= 0.00186$ lb/sq in.

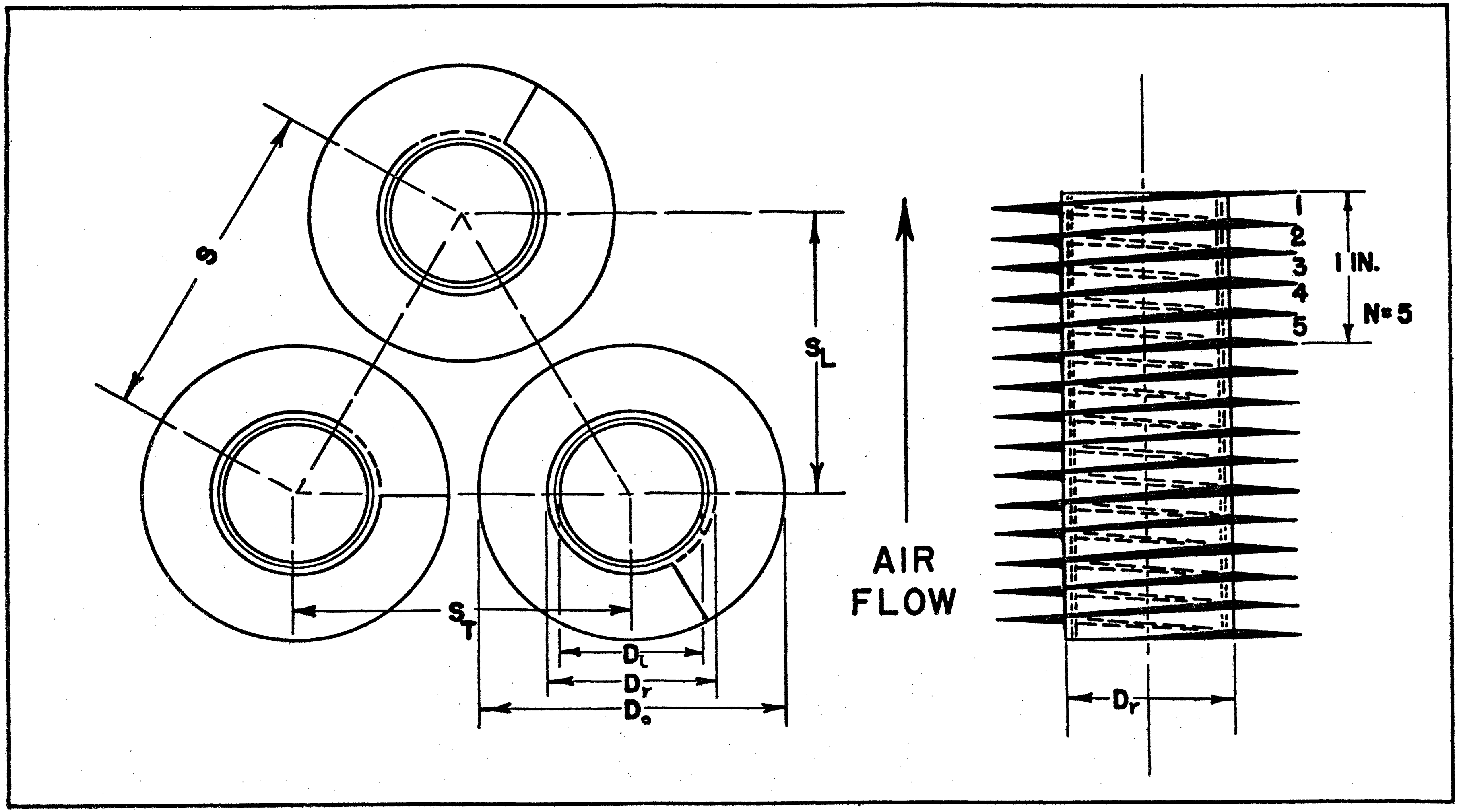


FIGURE 1 NOMENCLATURE FOR DIMENSIONS OF FINNED TUBE AND TUBE SPACING.

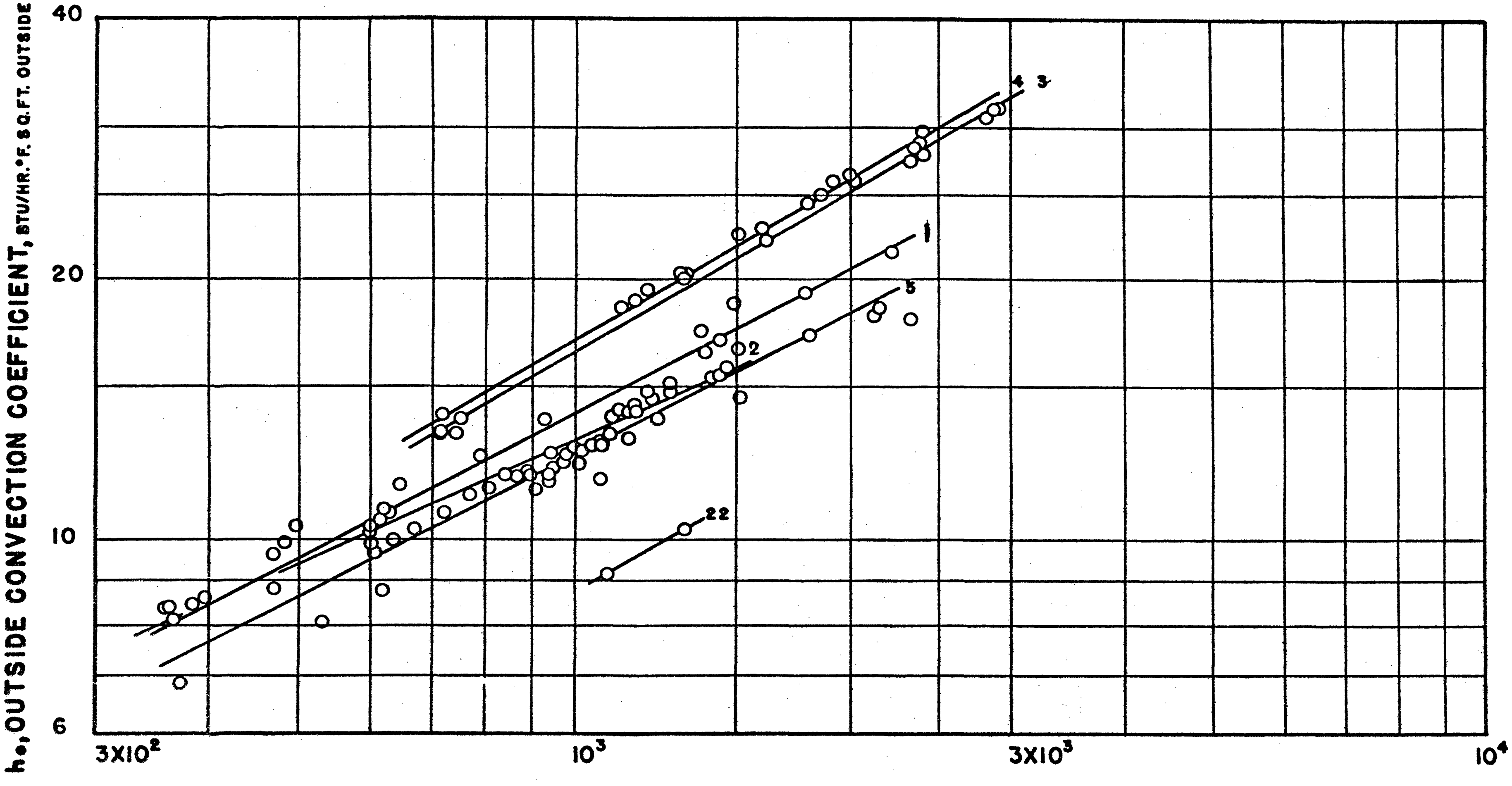


FIGURE 2 OUTSIDE CONVECTION COEFFICIENT AS A FUNCTION OF THE MAXIMUM AIR VELOCITY.

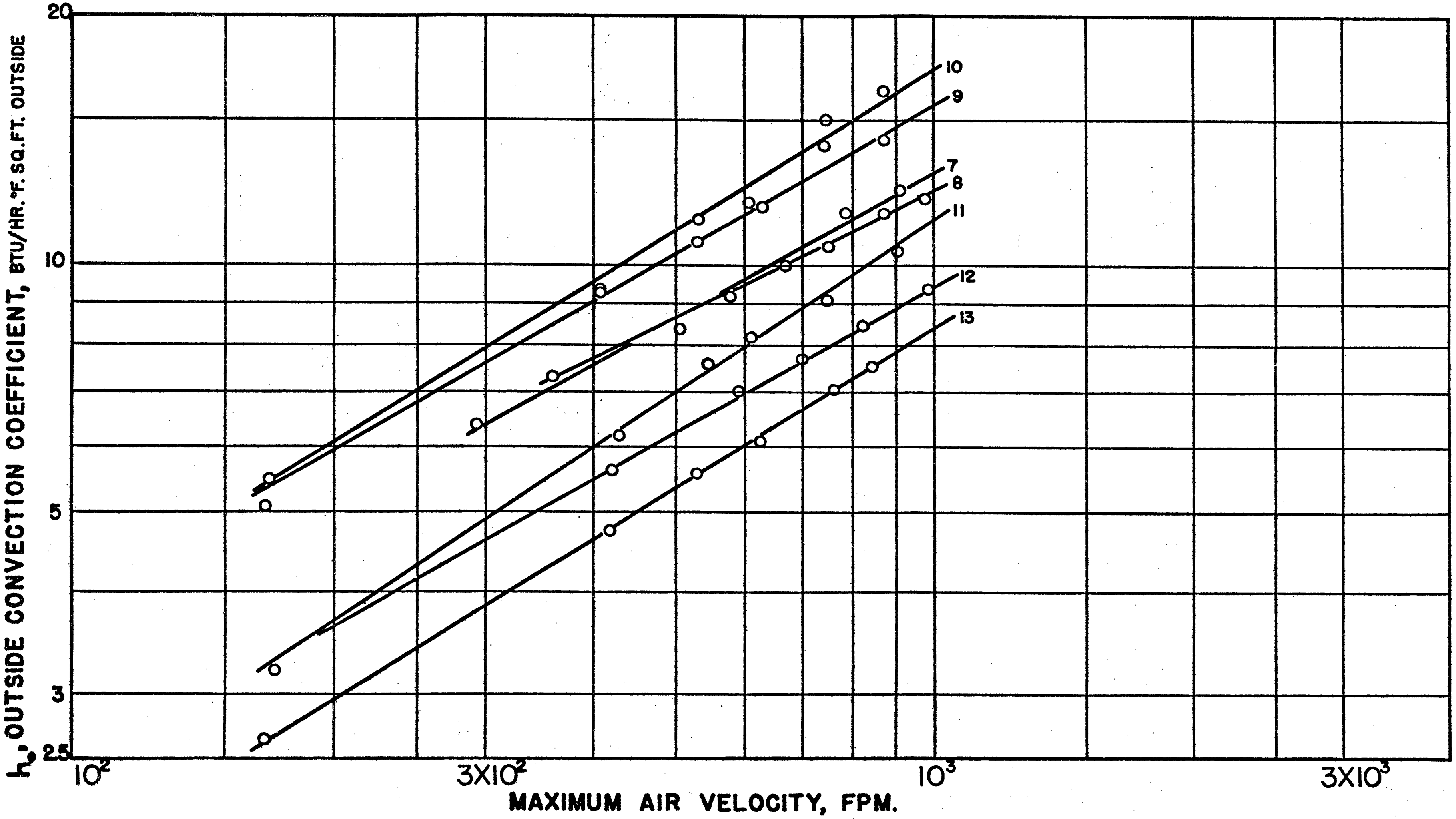


FIGURE 3 OUTSIDE CONVECTION COEFFICIENT
AS A FUNCTION OF THE MAXIMUM AIR VELOCITY.

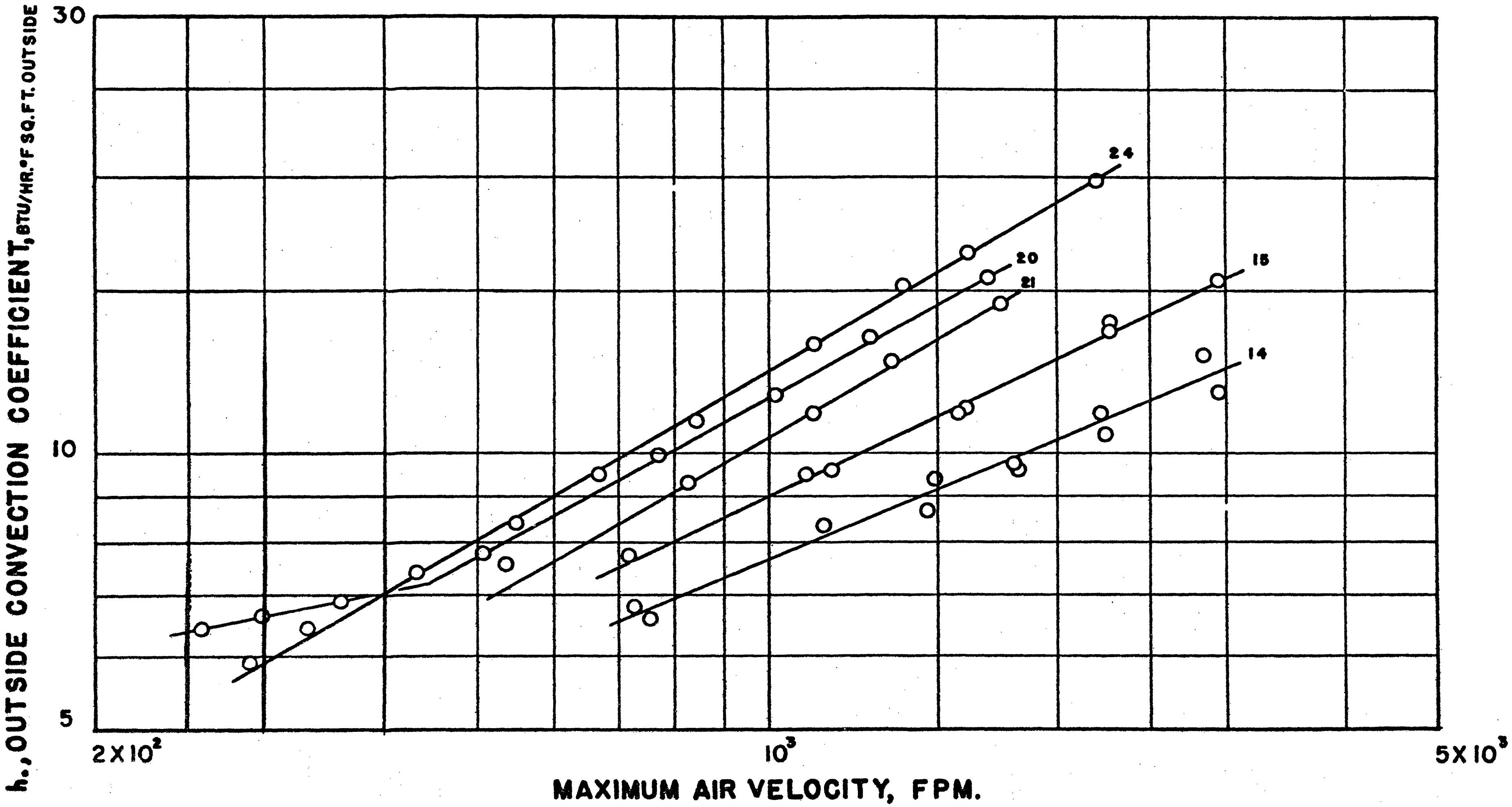


FIGURE 4 OUTSIDE CONVECTION COEFFICIENT AS A FUNCTION OF THE MAXIMUM AIR VELOCITY.

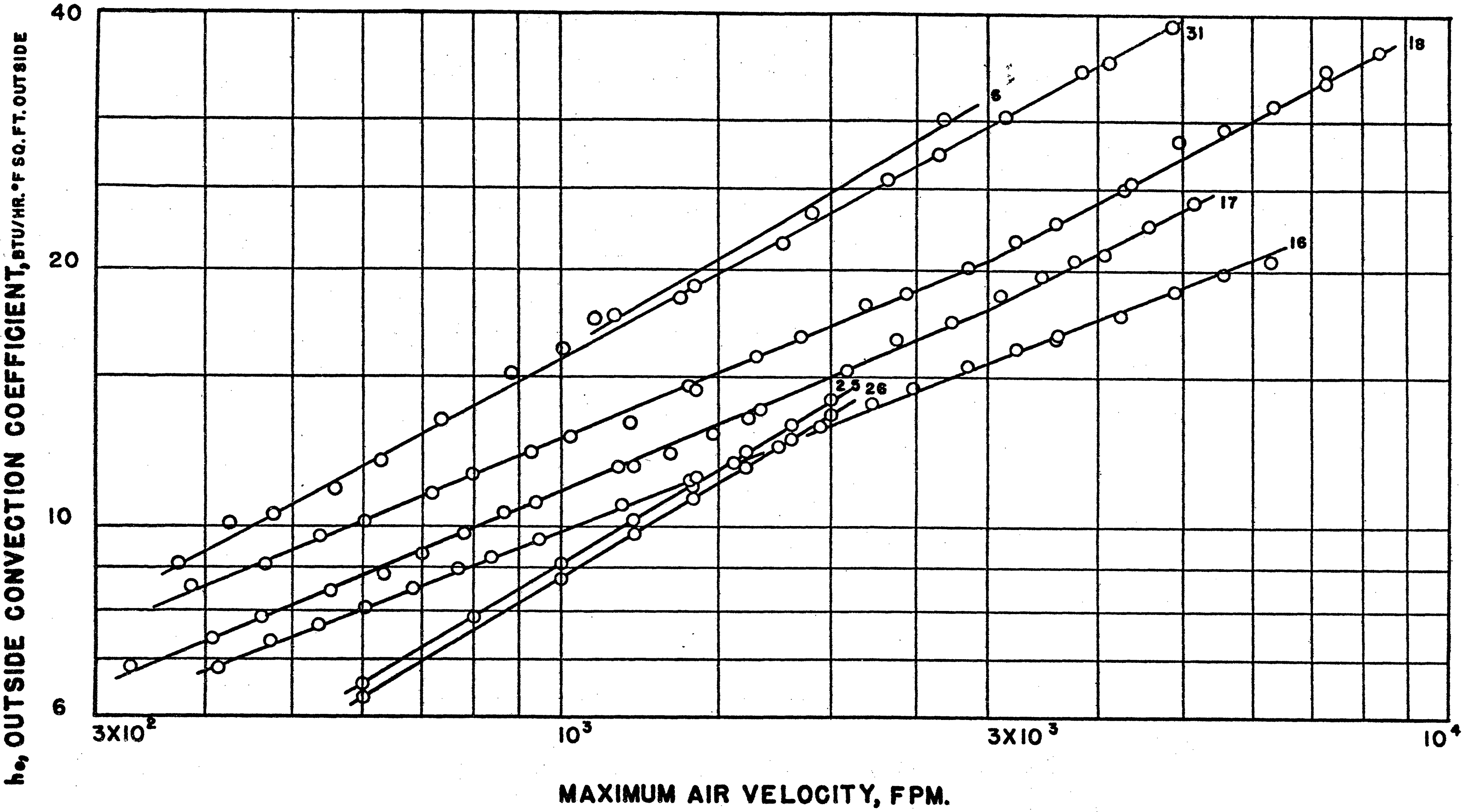


FIGURE 5 OUTSIDE CONVECTION COEFFICIENT AS A FUNCTION OF THE MAXIMUM AIR VELOCITY.

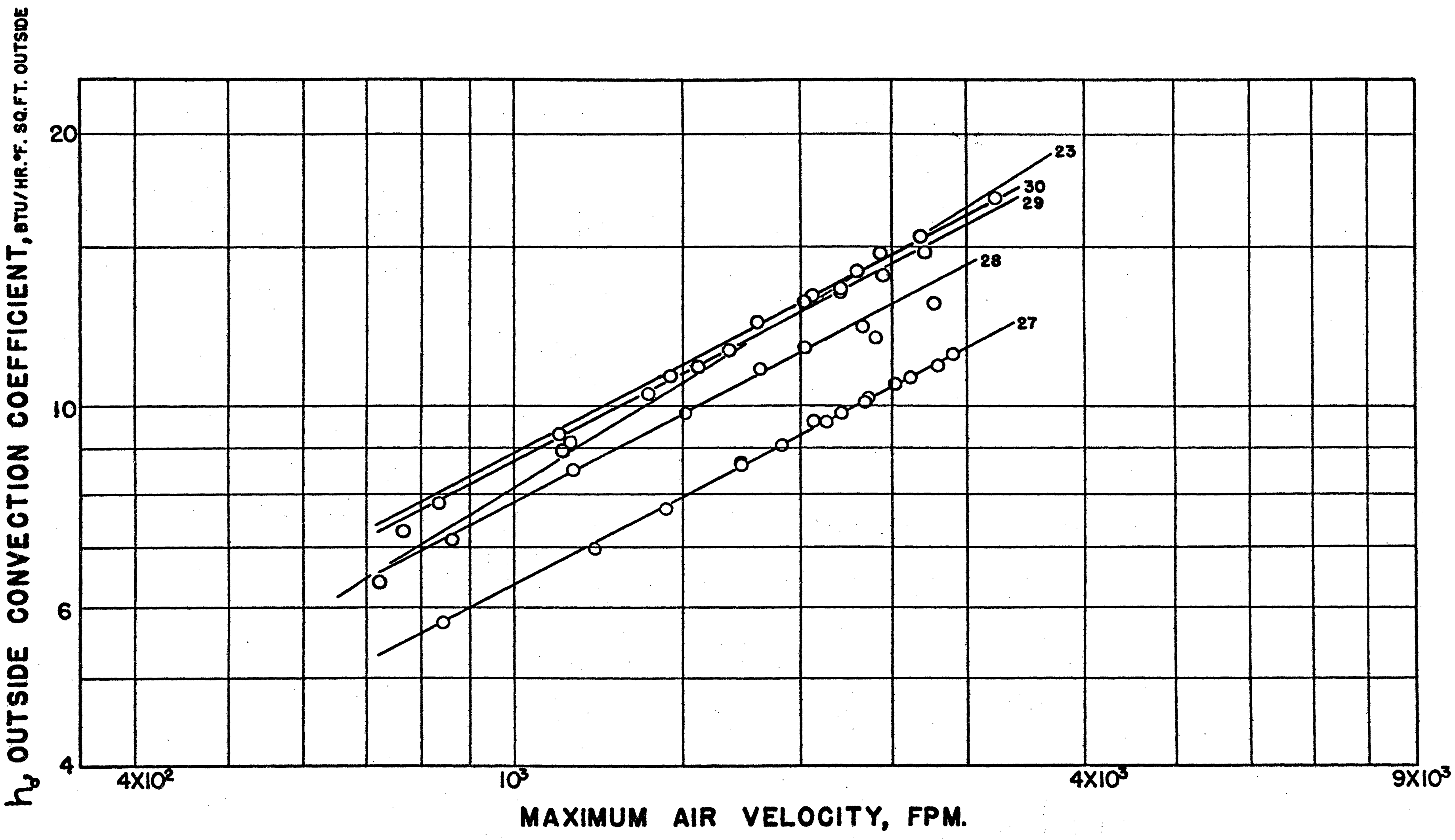


FIGURE 6 OUTSIDE CONVECTION COEFFICIENT AS A FUNCTION OF THE MAXIMUM AIR VELOCITY.

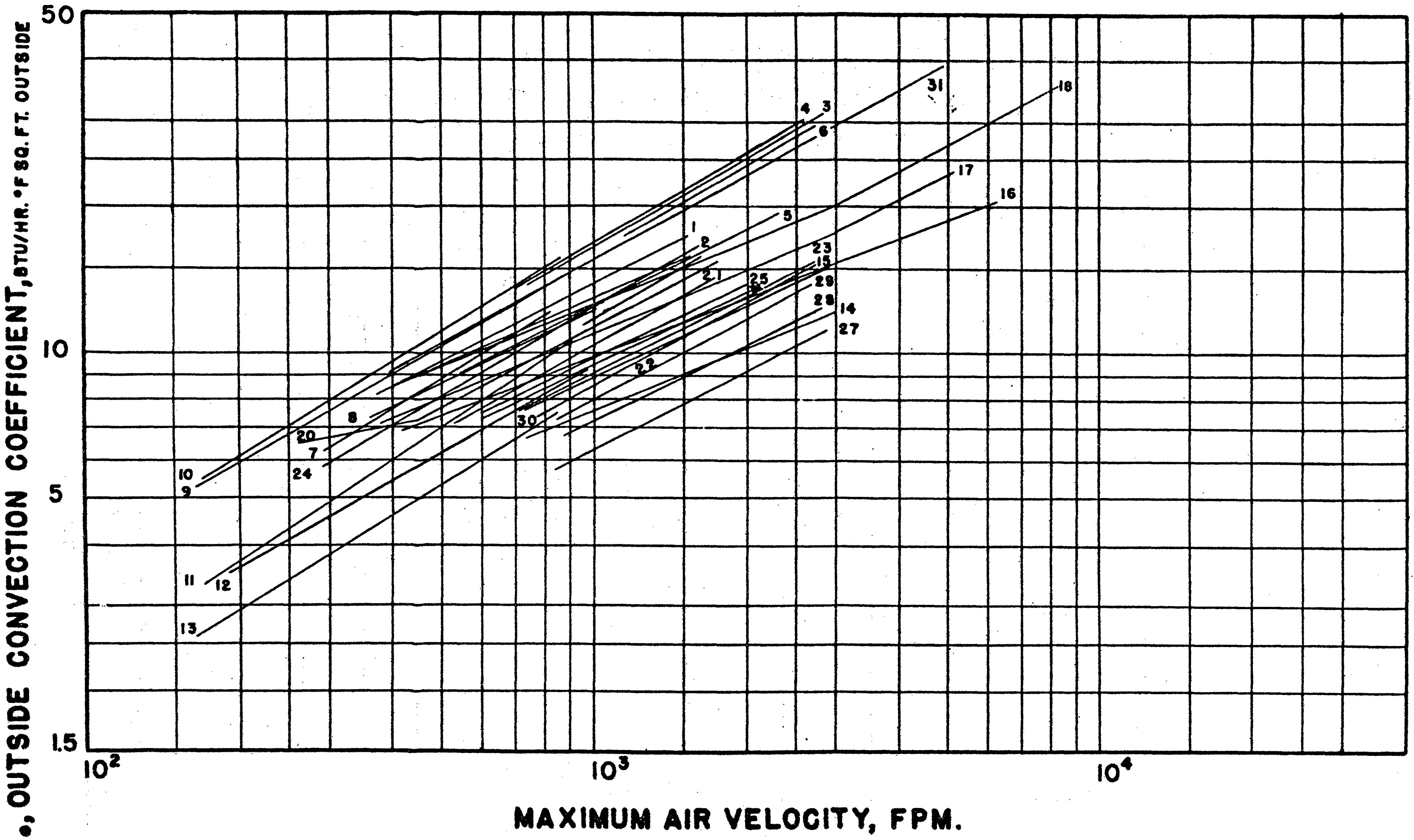


FIGURE 7 RANGE OF HEAT TRANSFER DATA.

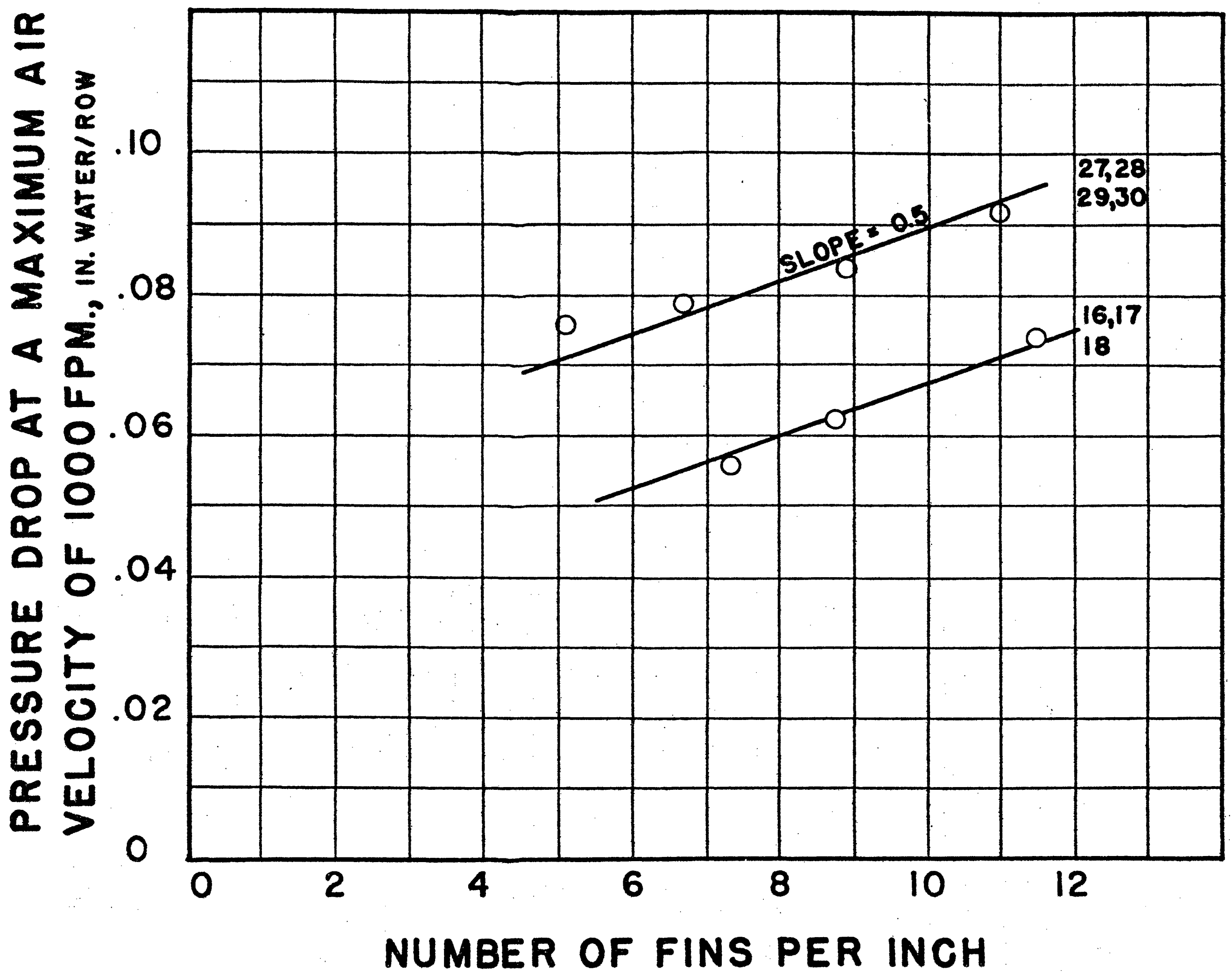


FIGURE 8 VARIATION OF PRESSURE DROP WITH THE NUMBER OF FINS PER INCH.

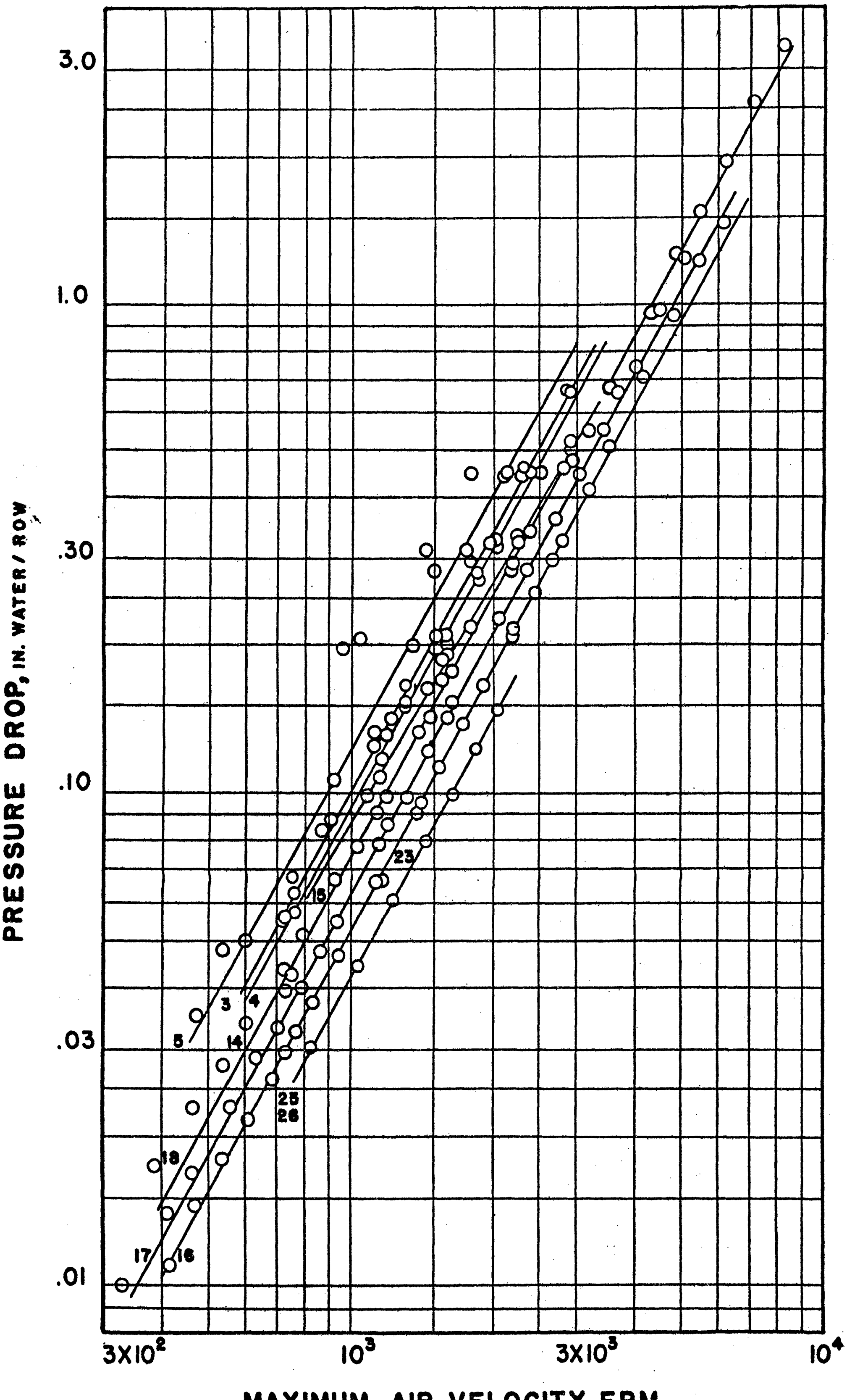


FIGURE 9 PRESSURE DROP AS A FUNCTION OF THE MAXIMUM AIR VELOCITY.

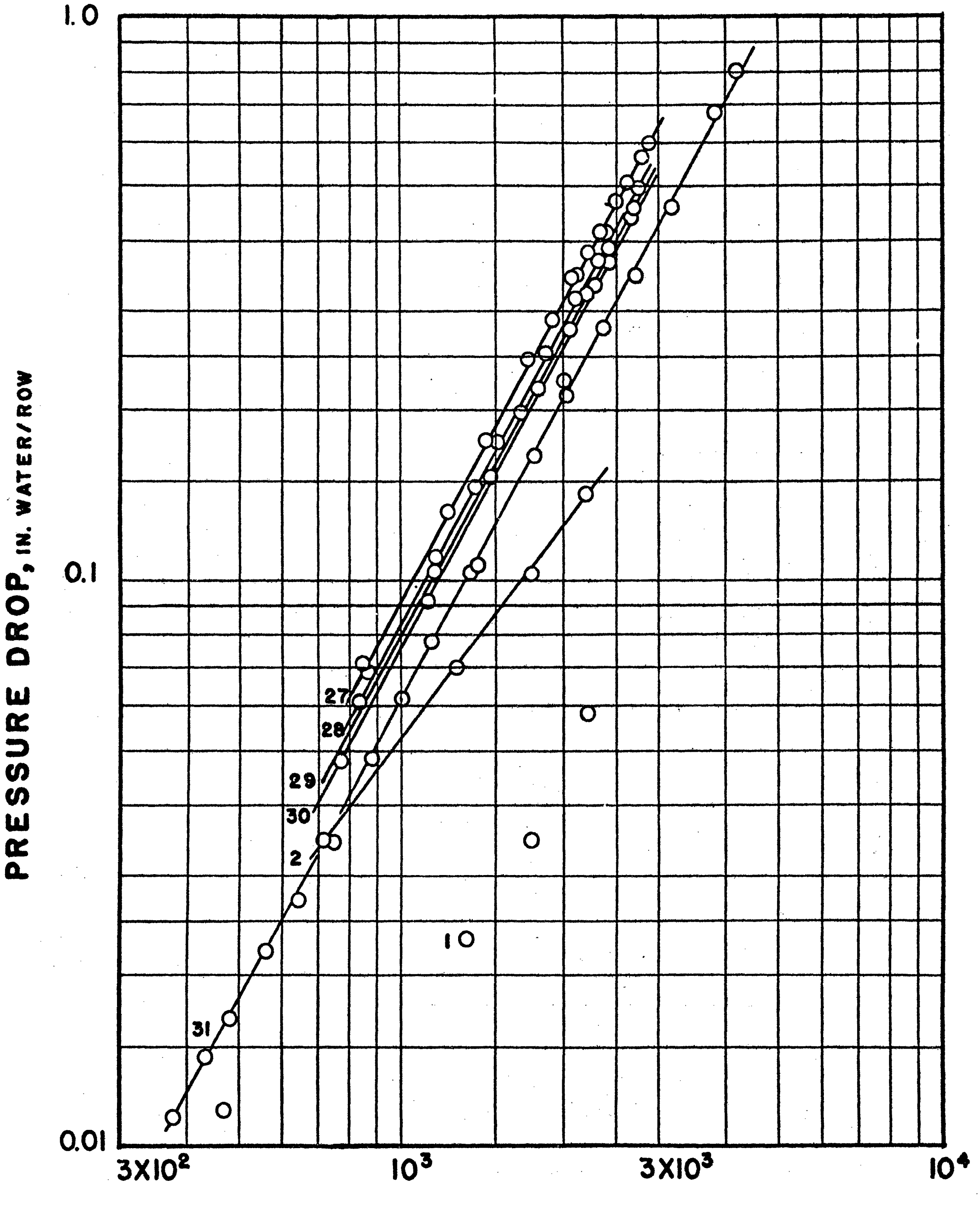


FIGURE 10 PRESSURE DROP AS A FUNCTION OF THE MAXIMUM AIR VELOCITY.

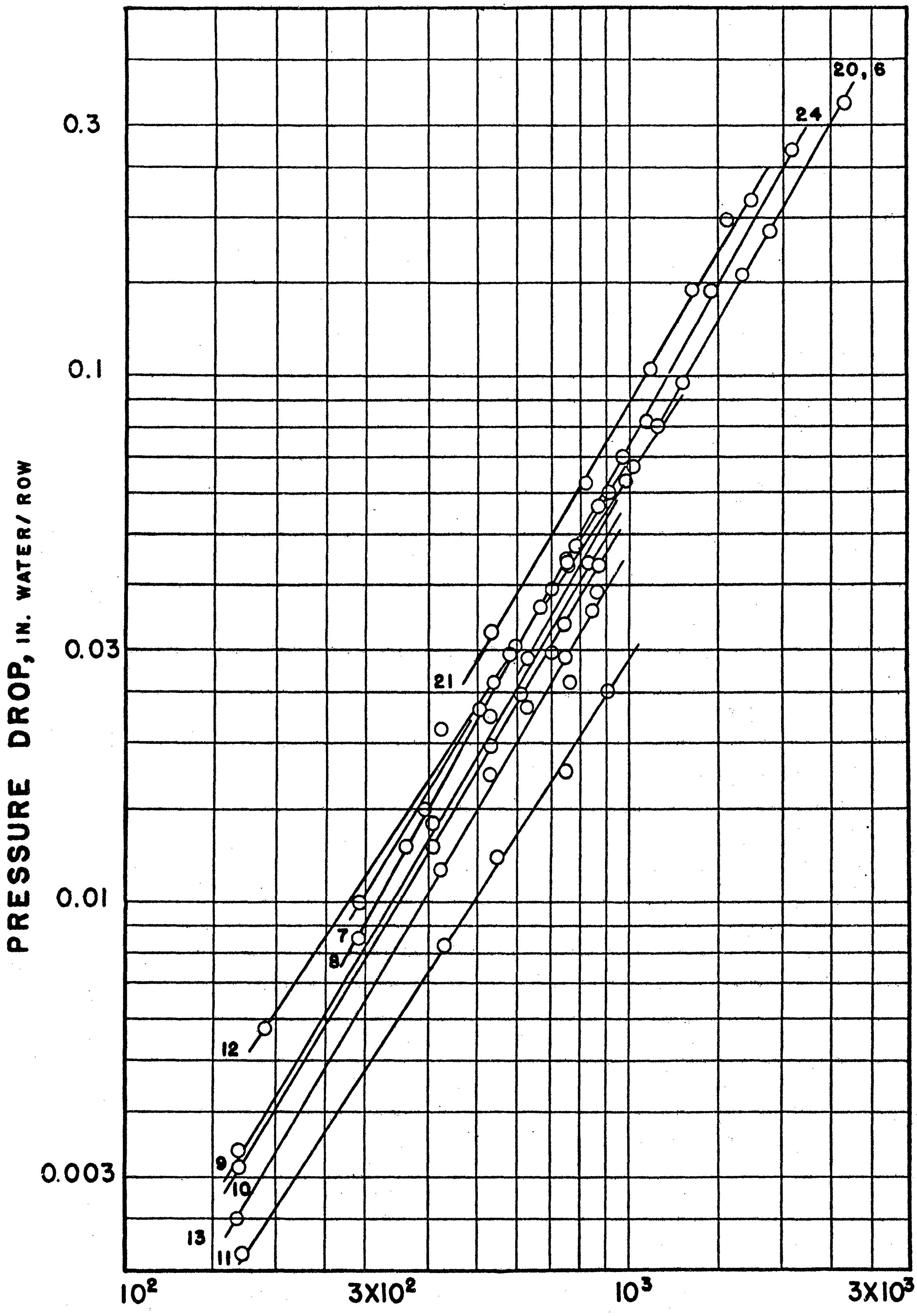
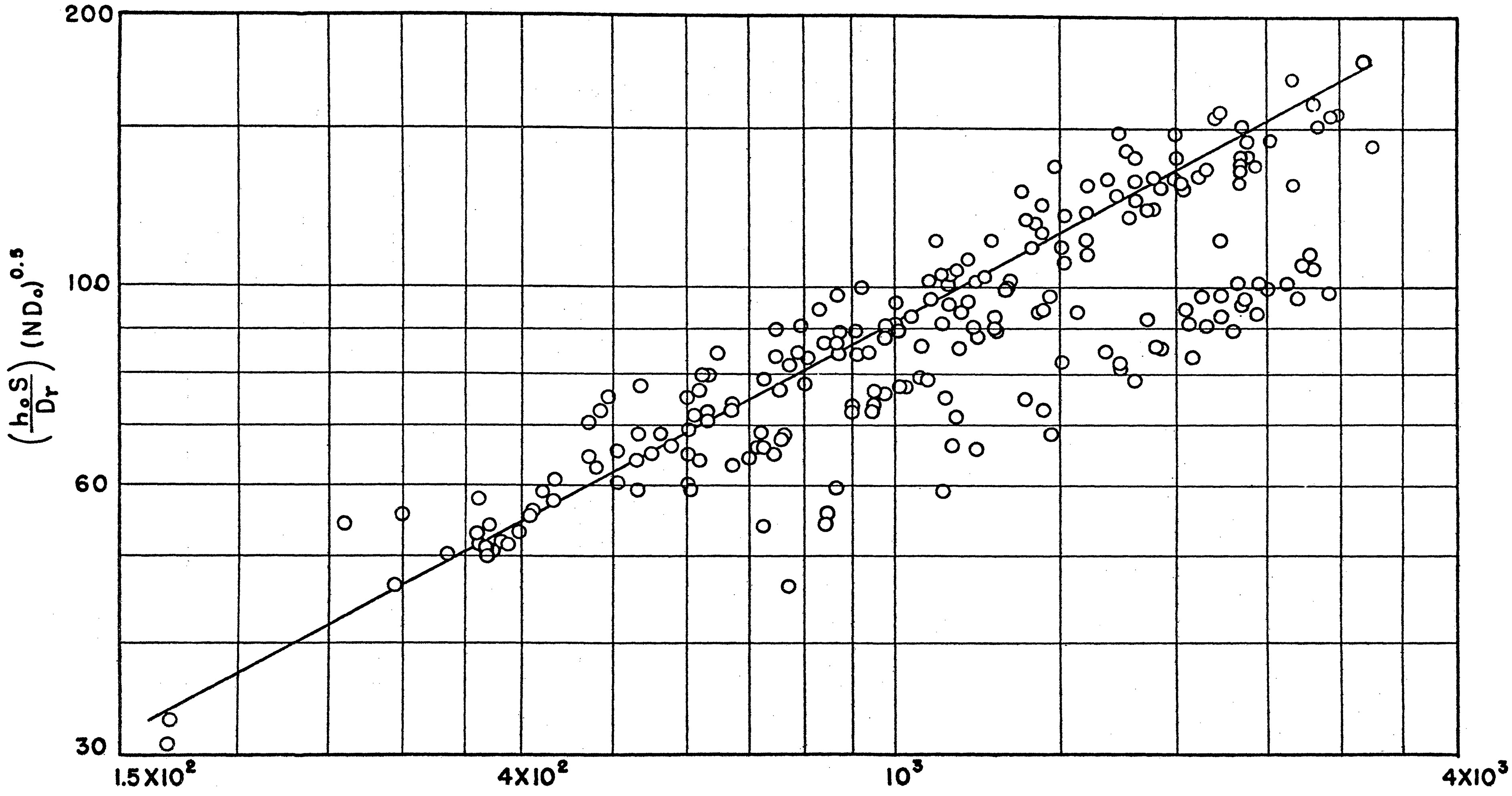


FIGURE 11 PRESSURE DROP AS A FUNCTION OF THE MAXIMUM AIR VELOCITY.



MAXIMUM AIR VELOCITY, FPM
FIGURE 12 HEAT-TRANSFER CORRELATION FOR AIR.

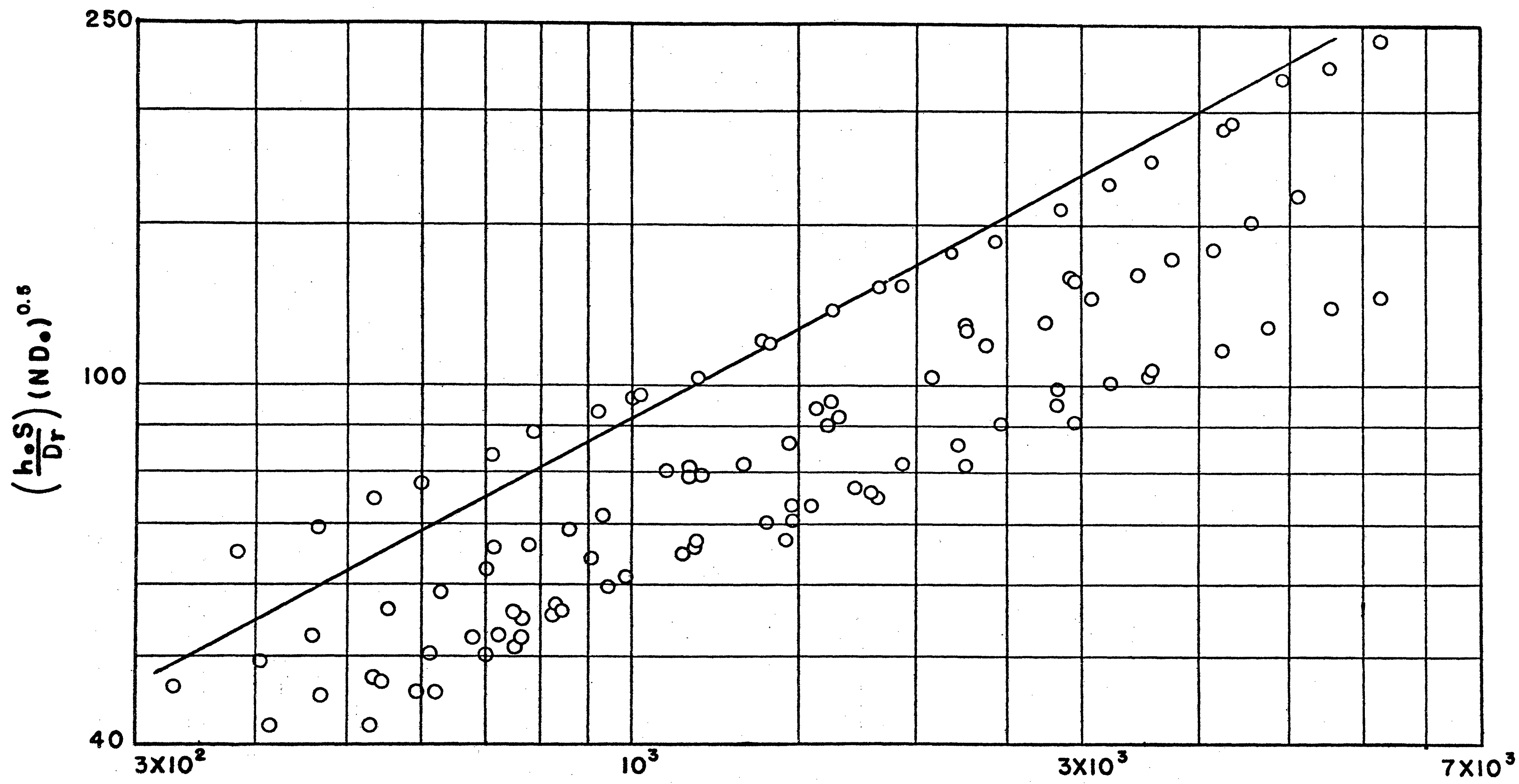


FIGURE 13 COMPARISON OF THE HEAT-TRANSFER CORRELATION WITH NON-TRIANGULAR AND OTHER TUBE BANKS.

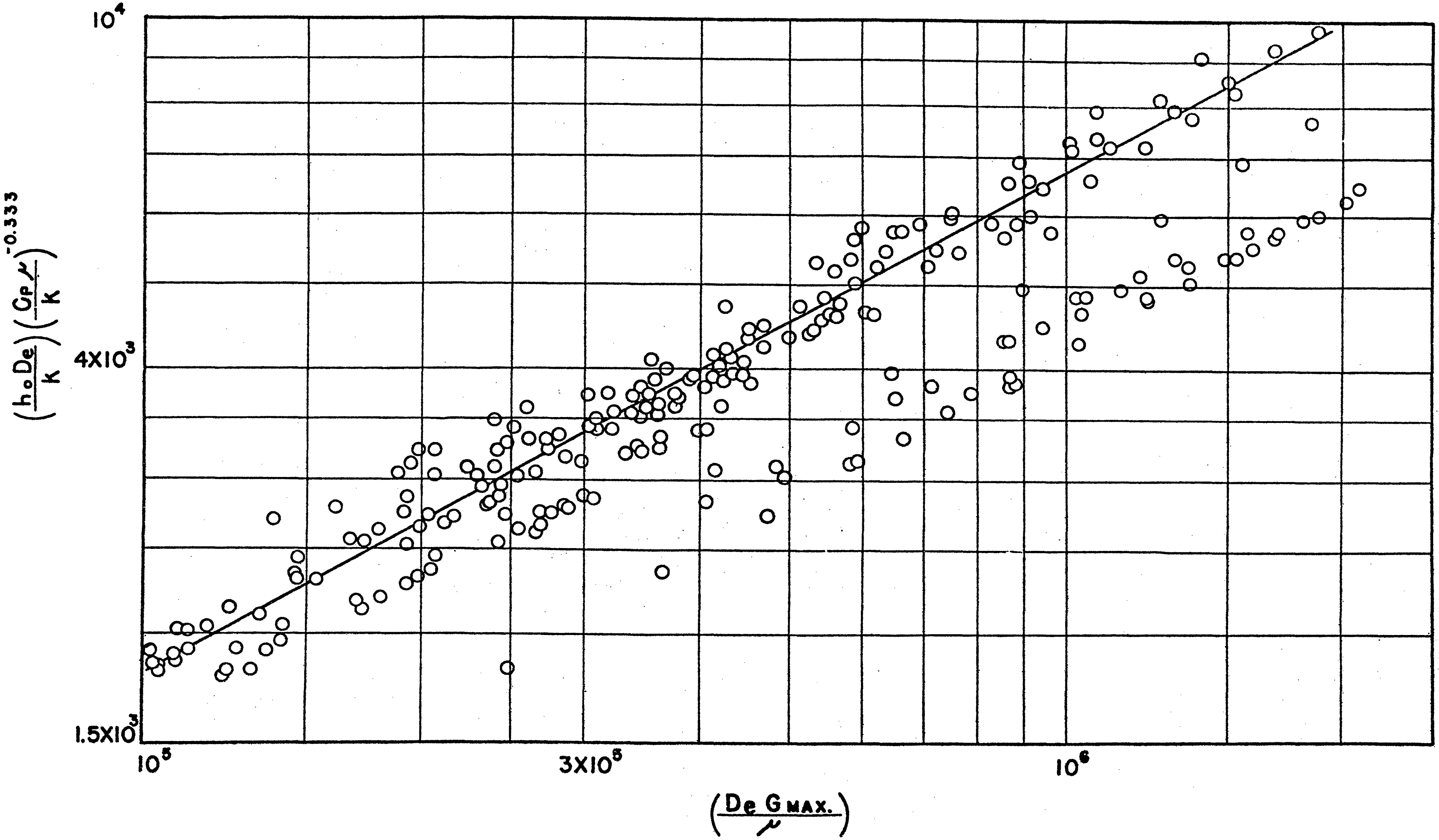
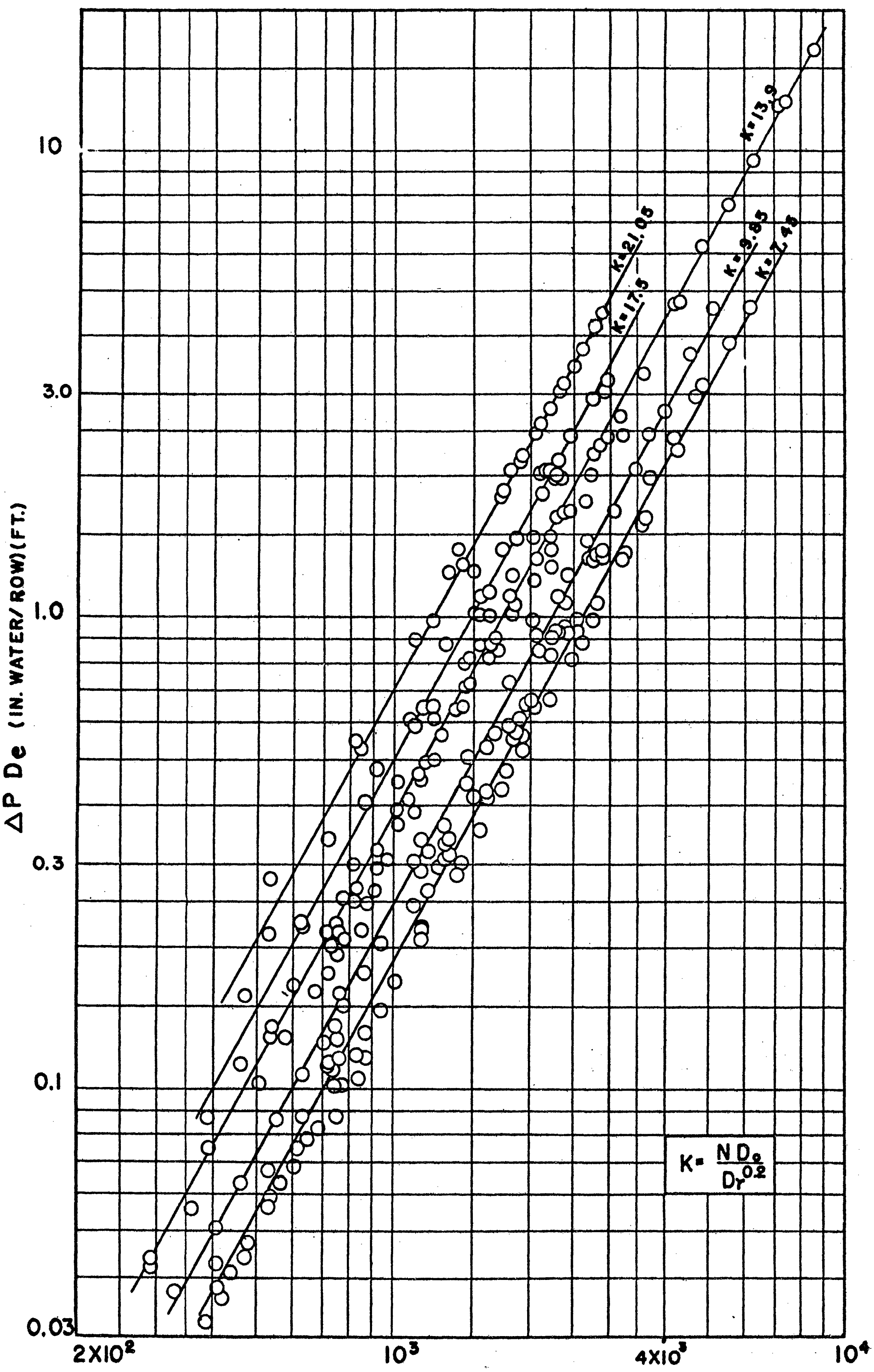


FIGURE 14 GENERALIZED HEAT-TRANSFER CORRELATION.



MAXIMUM AIR VELOCITY, FPM

FIGURE 15 PRESSURE-DROP CORRELATION FOR AIR.

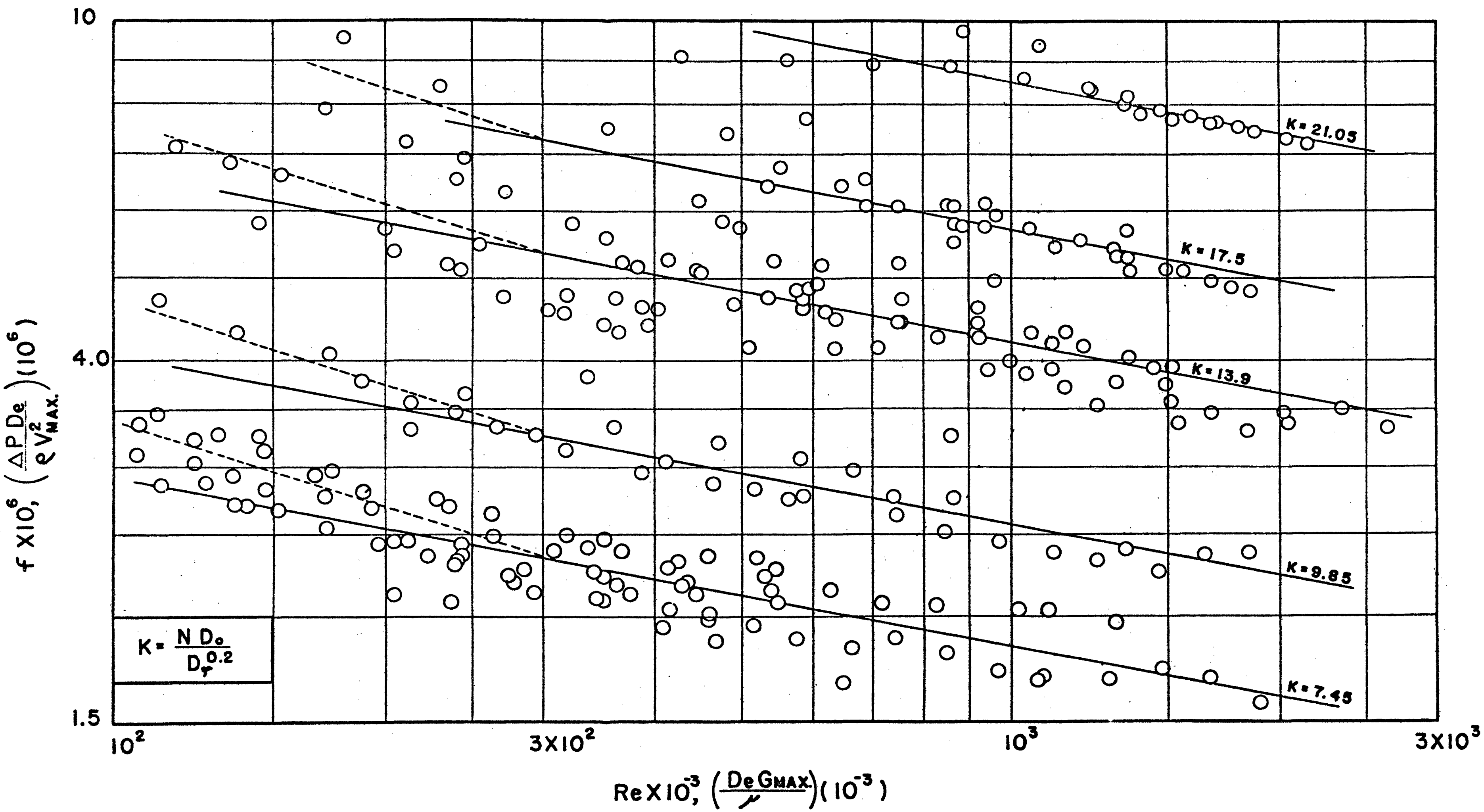


FIGURE 16 GENERALIZED PRESSURE DROP CORRELATION.

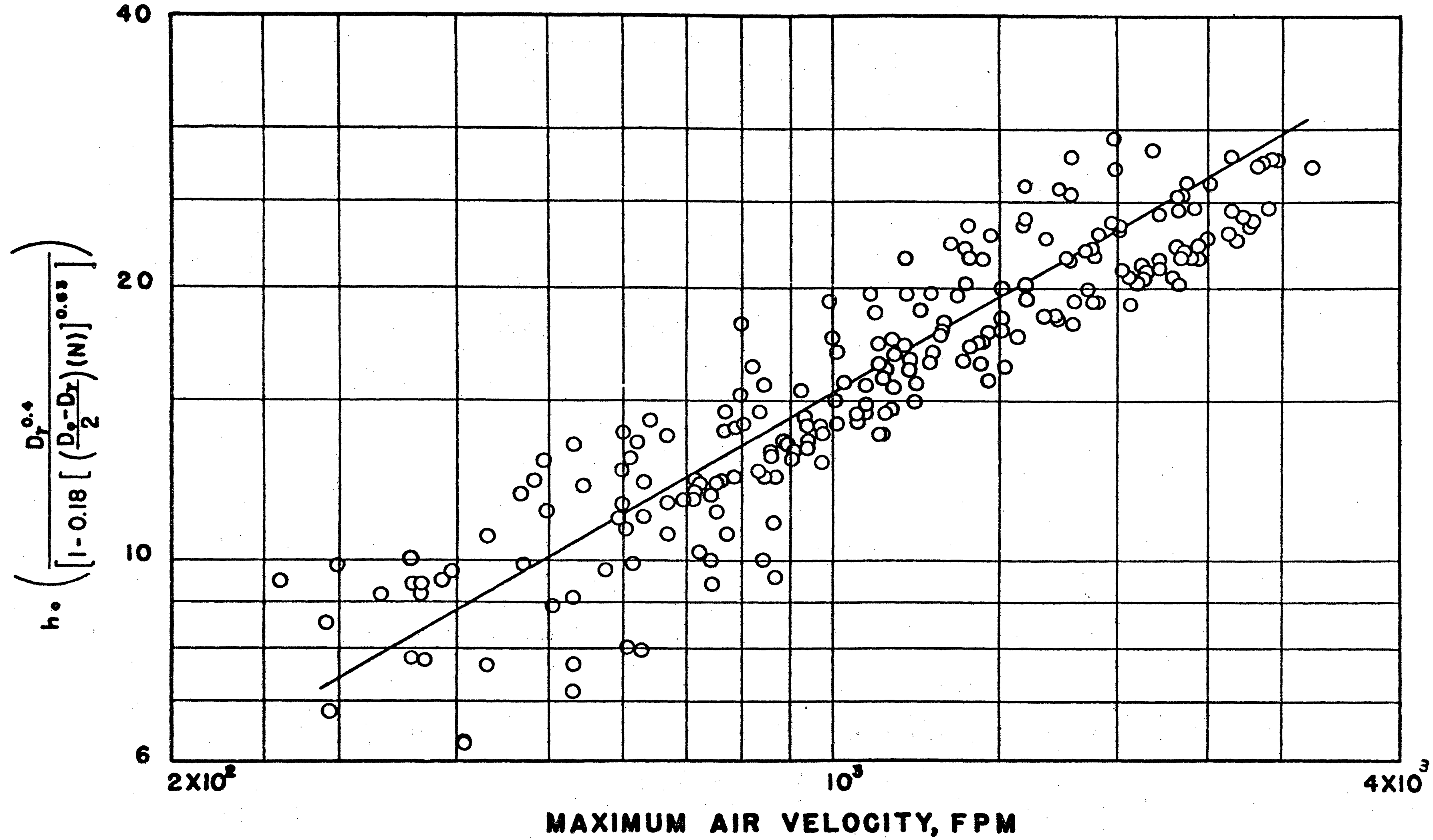


FIGURE 17 HEAT-TRANSFER CORRELATION OF T. E. SCHMIDT APPLIED TO AIR.

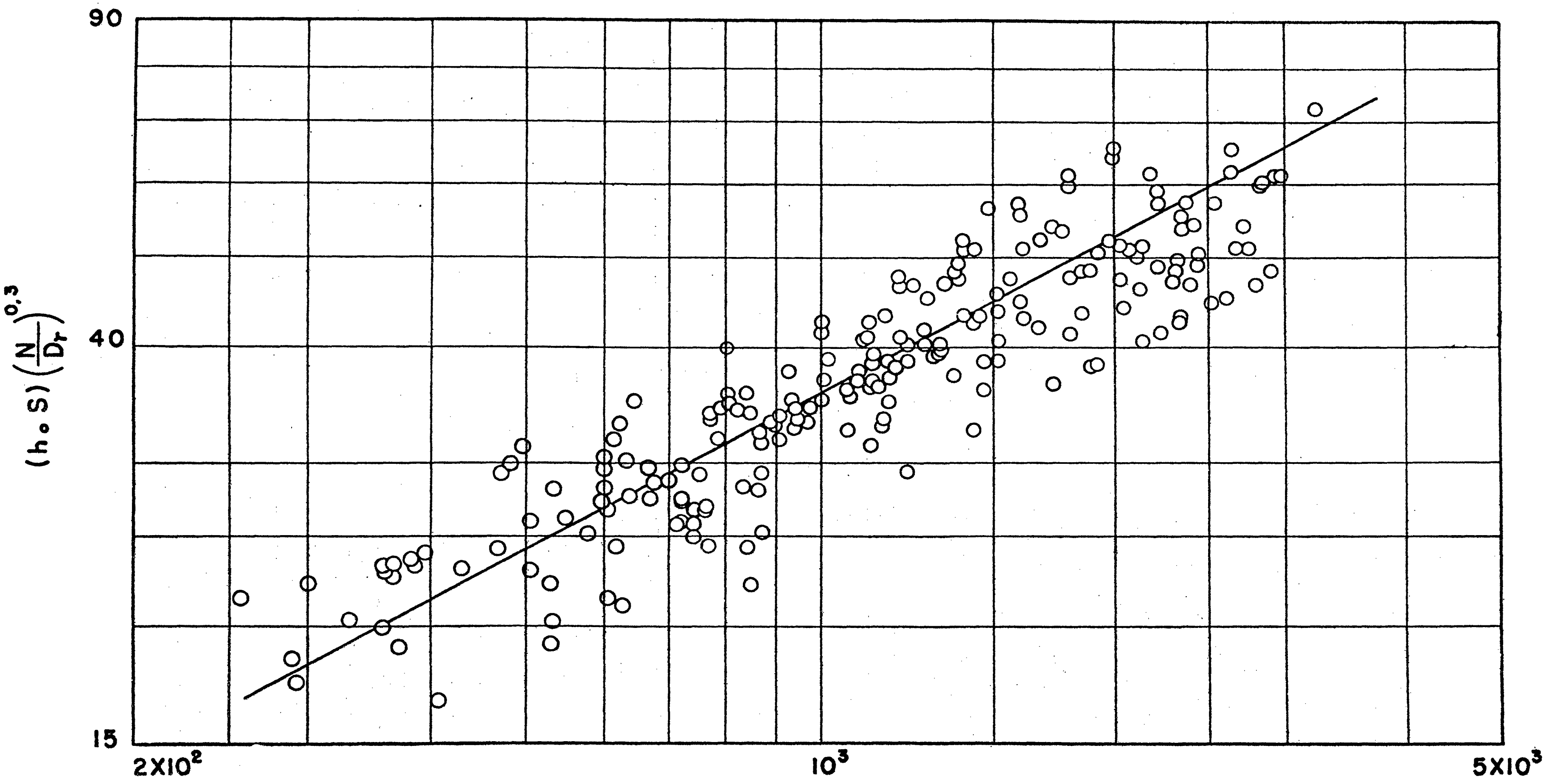


FIGURE 18 HEAT-TRANSFER CORRELATION FOR AIR AS A FUNCTION OF S, N, AND D_r.

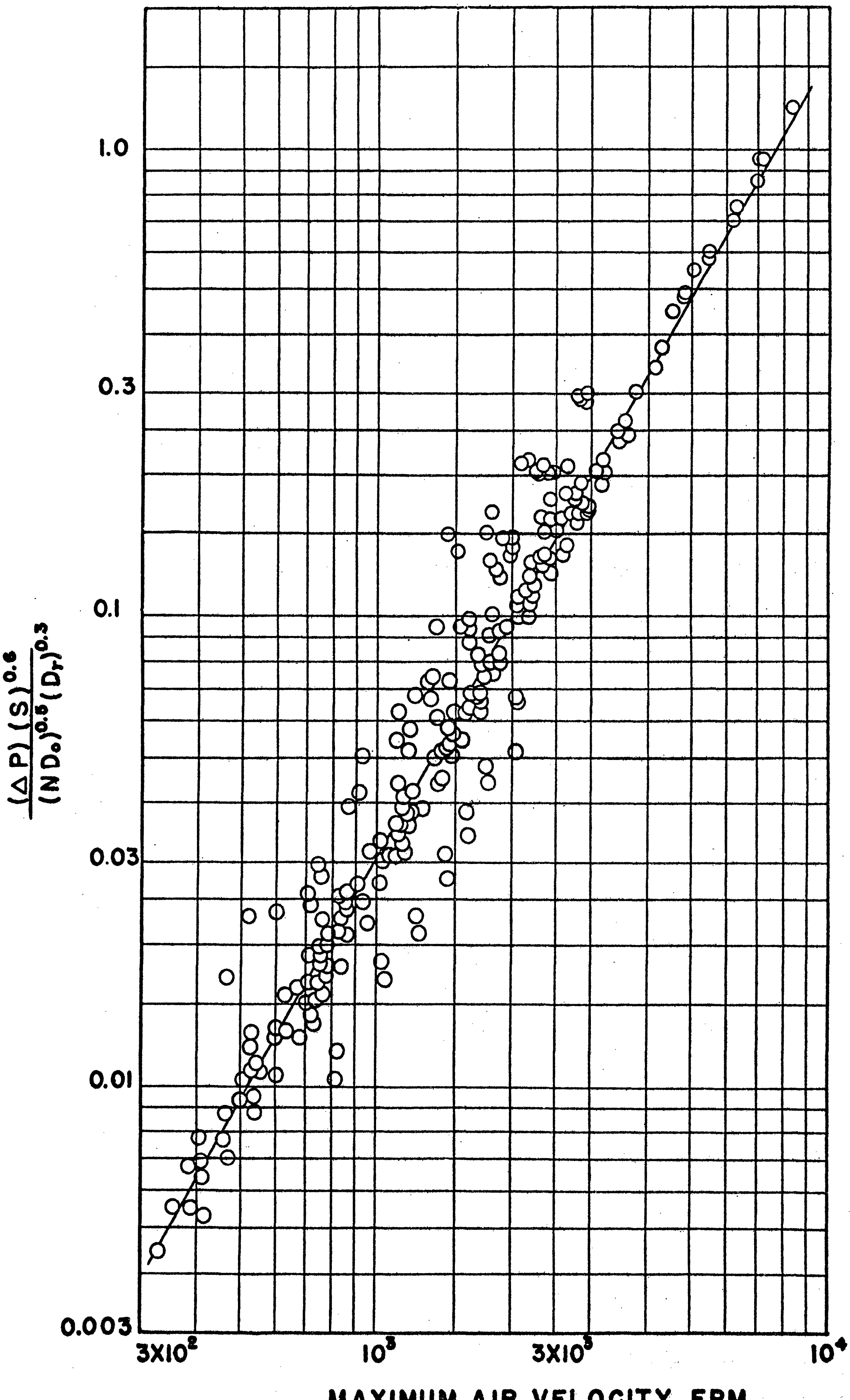


FIGURE 19 EMPIRICAL PRESSURE DROP CORRELATION FOR AIR.

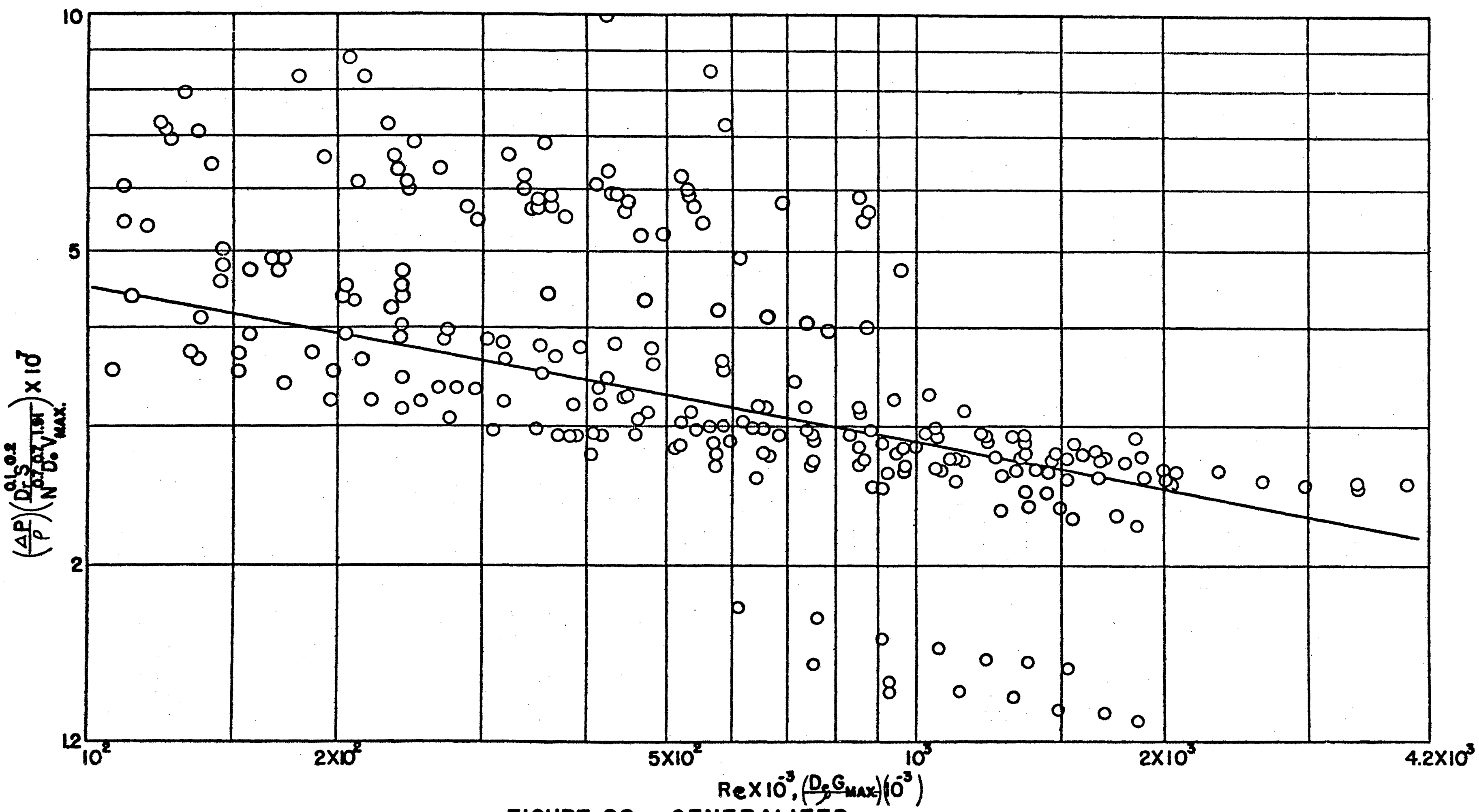


FIGURE 20 GENERALIZED PRESSURE DROP CORRELATION

TABLE I

DIMENSIONS OF TUBES AND UNITS FOR AVAILABLE DATA

UNIT NUMBER	TUBE ARRANGE-MENT	NUMBER OF ROWS	NUMBER OF TUBES PER ROW	FACE AREA IN. HIGH X IN. WIDE	D_r, IN.	D_o, IN.	N FINS/IN.	S_T, IN.	S_L, IN.	S, IN. AVERAGE	TUBE MATERIAL	BLOWER ARRANGEMENT	D_e, FT.	m	TEMPERATURE RANGE OF AIR, °F.	RANGE OF MAX. VELOCITY, FT./MIN.	PRESSURE DROP RANGE IN. WATER/ROW	RANGE OF U_o, BTU/HR. °F. SQ. FT.	RANGE OF h_o, BTU/HR. °F. SQ. FT.
1	-------	1	12	24.5 X 17.6	0.622	1.250	5.80	1.41	1.41	1.41	Cu	U	3.09	0.552	———	420-1500	0.0116-0.058	7.91-15.67	8.2-16.88
2	S	2	12	24.5 X 17.6	0.621	1.313	7.80	1.41	1.41	1.41	Cu	U	4.38	0.477	———	360-1500	0.0695-0.284	8.19-16.1	8.0-15.80
3	S	4	22-23	18.0 X 17.25	0.385	0.700	9.13	0.75	0.75	0.75	Cu	U	2.02	0.620	80.2-272.5	740-2900	0.0575-0.668	13.0 -29.8	13.7-32
4	S	6	22-23	18.0 X 17.25	0.385	0.700	9.13	0.75	0.75	0.75	Cu	U	2.02	0.620	80.6-311.7	700-2500	0.055 -0.46	11.09-28.0	13.6-30
5	S	6	13	18.0 X 16.88	0.5	1.22	7.05	1.25	1.25	1.25	Cu	U	4.48	0.540	80.2-323.2	372-2300	0.0358-0.455	6.84-20.6	7.4-19.8
6	S	6	19-20	18.3 X 15.38	0.375	0.780	9.5	0.78	0.78	0.78	Cu	D	2.67	0.622	———	1140-2600	0.08 -0.33	———	17.7-30.1
7	S	4	7-6	11.4 X 7.75	0.46	0.98	9.25	1.11	1.11	1.11	Al-Cu	D	4.4	0.586	78 - 152.8	292-908	0.01 - 0.06	5.36 - 9	6.4-12.34
8	S	3	7-6	11.4 X 7.75	0.46	0.98	9.25	1.11	1.11	1.11	Al-Cu	D	4.4	0.516	90 - 145	359-971	0.0127-0.07	6.2 - 9.33	7.3-12.1
9	S	8	13	11.6 X 7.5	0.20	0.48	9.5	0.562	0.562	0.562	Al-Cu	D	3.0	0.605	86 - 172	167-869	0.0034-0.044	4.81 -12.3	5.1-14.22
10	S	8	13	11.6 X 7.5	0.20	0.48	9.5	0.562	0.562	0.562	Cu	D	3.0	0.65	93 -174.4	169-866	0.0031-0.039	5.17-13.85	5.49-16.32
11	IL	7	13	11.6 X 7.5	0.20	0.48	9.5	0.576	0.576	0.576	Cu	D	3.16	0.696	92 -158	171-905	0.002 -0.025	3.11- 9.46	3.21-10.44
12	IL	4	8	11.6 X 7.5	0.375	0.875	9	0.875	0.875	0.875	Al-Cu	D	3.56	0.591	96 - 152	190-983	0.006 -0.063	2.64- 7.59	2.83- 9.39
13	IL	4	7	11.6 X 7.5	0.375	0.875	9	1.0	1.0	1.0	Al	D	4.65	0.651	91.5-147	167-844	0.0025-0.036	2.5 - 6.45	2.65-7.55
14	S	3	6	11.9 X 13	1.06	1.97	8.7	2.0	2.0	2.0	Al-Cu	D	5.07	0.433	51.8-113.8	724-2950	0.043 -0.476	4.95-9.2	6.62-12.75
15	S	3	6	11.9 X 13	1.07	2.03	10	2.01	2.01	2.01	Al-Cu	D	5.98	0.498	58.3-124.9	715-2950	0.057 -0.52	6.2 -10.4	7.77-15.4
16	S	5	10-9	8.4 X 9.75	0.380	0.92	7.34	0.975	0.80	0.9	Al-Cu	U	3.15	0.414	80 -190.9	413-6250	0.011 -1.458	———	6.83-20.33
17	S	5	10-9	8.4 X 9.75	0.380	0.92	8.72	0.975	0.80	0.9	Al-Cu	U	3.74	0.445	74.6-206	329-5120	0.01 -1.238	———	6.88-24.09
18	S	5	10-9	8.4 X 9.75	0.380	0.92	11.46	0.975	0.80	0.9	Al-Cu	U	4.93	0.435	70.1-217.7	384-8290	0.0176-3.35	———	8.55-36.19
20	S	4	9	40.5 X 14.4	0.681	1.587	8.36	1.57	1.36	1.57	Al-Cu	U(?)	5.85	0.581	———	258-1690	0.043-0.155	———	6.46-15.62
21	S	4	9	40.5 X 14.4	0.658	1.608	11.4	1.57	1.36	1.57	Al-Cu	U(?)	8.7	0.605	———	534-1748	0.032-0.215	———	7.6 -14.6
22	----	1	3	71.1 X ----	1.032	1.74	7.11	——	——	——	Al	——	——	0.616	———	1089-1312	———	———	9.22-10.27
23	S	4	5-4	36 X 10	1.050	1.939	6.84	2.0625	2.625	2.44	Al-Cu	——	5.96	0.651	86 -147	721-3221	0.03 -0.42	5.2 -12.9	6.4 -17
24	S	4	12	40.3 X 16.75	0.667	1.410	9.83	1.39	1.20	1.39	Al-Cu	U(?)	5.01	0.610	———	290-2195	0.008-0.27	———	5.95-19.88
25	S	----	----	24 FT. X 20 FT.	1.030	2.07	9	2.375	2.375	2.375	Al-Cu	D	8.25	0.64	———	600-2000	0.03 -0.15	———	6.55-14.1
26	S	----	----	24 FT. X 20 FT.	1.030	2.07	11	2.375	2.375	2.375	Al-Cu	D	10.2	0.64	———	600-2000	0.03 -0.15	———	6.3 -13.5
27	S	2	8	30 X 16.5	1.030	2.03	11	2.0625	2.0625	2.0625	Al-Cu	D	7.46	0.55	———	844-2800	0.071 -0.565	5.1 - 8.91	5.78-11.1
28	S	2	8	30 X 16.5	1.030	2.03	8.87	2.0625	2.0625	2.0625	Al-Cu	D	5.85	0.55	———	862-2780	0.07 - 0.5	6.22-10.6	7.13-13.0
29	S	2	8	30 X 16.5	1.030	2.03	6.67	2.0625	2.0625	2.0625	Al-Cu	D	4.4	0.55	———	836-2710	0.061-0.457	7.04-12.42	7.82-14.84
30	S	2	8	30 X 16.5	1.030	2.03	5.07	2.0625	2.0625	2.0625	Al-Cu	D	3.34	0.55	———	768-2690	0.05 - 0.441	6.75-13.20	7.3 -15.45
31	S	5	10-9	8.4 X 9.75	0.420	0.86	8.72	0.975	0.80	0.90	Cu	U	2.86	0.570	———	380-4920	0.0115- 1.085	———	9.1 -39.6

TABLE II
DIMENSIONS OF TUBES AND UNITS

Unit Number	Ratio of Min Free Area to Face Area	Ratio of Outside to Inside Surface Area	Outside Area ft^2/ft	Average Fin thickness, in.
1	0.517	7.44	1.056	0.024
2	0.487	10.8	1.53	0.024
3	0.412	6.45	0.503	0.02
4	0.412	6.45	0.503	0.02
5	0.526	11.56	1.27	0.02
6	0.461	8.80	0.680	0.014
7	0.551	11.44	1.05	0.016
8	0.542	11.44	1.05	0.016
9	0.578	9.23	0.290	0.016 (?)
10	0.578	9.23	0.290	0.016 (?)
11	0.578	9.23	0.290	0.016 (?)
12	0.524	10.84	0.838	0.016 (?)
13	0.582	10.84	0.838	0.016 (?)
14	0.406	14.5	3.42	0.0161
15	0.380	17.2	4.20	0.0173
16	0.537	11.36	0.774	0.018
17	0.517	13.2	0.90	0.018
18	0.510	16.7	1.14	0.016
20	0.486	16.15	2.44	0.019
21	0.461	22.4	3.38	0.019
22	-	8.82	2.10	0.025
23	0.468	11.22	2.65	0.022
24	0.449	14.72	2.17	0.0138
25	0.487	16.67	4.05	0.018
26	0.487	20.12	4.89	0.018
27	0.405	19.05	4.69	0.018
28	0.436	14.48	3.56	0.018
29	0.433	11.12	2.74	0.022
30	0.442	8.75	2.15	0.025
31	0.495	-	0.754	0.019

TABLE III

VARIABLES IN HEAT TRANSFER AND PRESSURE DROP

D_o -- diameter over the fins, in.

D_r -- root diameter, in.

N -- number of fins per in.

S -- center-to-center distance for tubes in an equilateral staggered bank of tubes.

D_e -- equivalent diameter of the bank of finned tubes, $(S/D_r)^2(ND_o/12)$, ft.

n -- number of tube rows normal to direction of flow.

V_{max} -- velocity of gas measured at 70°F, 1 atmosphere, and 50 per cent humidity, through the minimum free-flow area.

C_p -- heat capacity of outside fluid at the average fluid temperature, Btu/lb-°F.

k -- thermal conductivity of outside fluid at the average fluid temperature, Btu-ft/hr-°F-sq ft.

μ -- viscosity of outside fluid at the average fluid temperature, lbs/hr-ft.

h_o -- outside convection coefficient, Btu/hr-°F-sq ft outside.

G_{max} -- mass-flow rate of outside fluid through the minimum free-flow area, lbs/hr-sq ft free area.

Re -- Reynolds number, $D_e G_{max}/\mu$.

Pr -- Prandtl group, $C_p \mu/k$.

Nu -- Nusselt group, $h_o D_e/k$.

ΔP -- pressure drop, in. water/row of tubes.

ρ -- density of fluid at the average fluid temperature, lb/cu ft.

K -- pressure-drop parameter, $ND_o/D_r^{0.2}$.

f -- friction factor or function of the Reynolds number, $\frac{\Delta P D_e}{\rho V_{max}^2}$