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INTERCOOLER DESIGN FOR AIRCRAFT

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SUMMARY

Data on heat transfer and pressure losses for flow within circular and rectangular tubes and for flow perpendicular to tube banks are collected and presented. These data, together with a given set of design conditions, are sufficient for the calculation of an optimum intercooler design based on the total power chargeable to the intercooler. The total power is the sum of the powers expended in pumping the charge air and the cooling air through the intercooler and the power used to transport the weight of the intercooler.

The design is accomplished by considering the power chargeable to each of a series of intercoolers obtained by a systematic variation of the variables of the design. All the intercoolers considered satisfy the design conditions.

Three types of intercooler are considered: a counterflow intercooler with indirect-cooling surfaces in the form of fins, a counterflow intercooler with direct-cooling surfaces, and a tube-type cross-flow intercooler.

The optimum designs for the cross-flow and for the counterflow intercoolers are about equally good on a basis of power consumed. The structural rigidity and the practicability of construction of the cross-flow type give it a practical advantage over the other types considered. Although the counterflow intercooler with indirect-cooling surfaces is a reasonably practicable design, the power consumed is somewhat greater than the power chargeable to the cross-flow intercooler.

INTRODUCTION

When an airplane is operating at high altitude, it is necessary to use a supercharger to maintain ground pressure at the carburetor inlet. This maintenance of high intake-manifold pressure tends to keep the power output of the engine at ground-level value. The air, being compressed by the supercharger, however, is heated by adiabatic compression and friction to a temperature that seriously affects the performance of the engine. It is therefore necessary to use an intercooler to reduce the temperature of the air between the supercharger outlet and the carburetor inlet. The amount of cooling required of the intercooler depends on the efficiency of the supercharger installation.

In this investigation, several types of intercooler were compared and a design procedure that will give the best intercooler for a given set of design conditions is indicated. If the cost of the construction, the weight, the size, and the power consumed by the intercooler are disregarded, the design of an intercooler to meet a given set of conditions of temperature and mass flow is straightforward, provided that friction factor and heat-transfer data are available for the type of intercooler selected. The large number of variables involved in intercooler design, however, makes possible an infinite number of intercooler designs, all of which will meet the given conditions of temperature and mass flow. This infinite number of intercooler designs will vary widely in the characteristics of cost, weight, size, and power consumed. It is therefore necessary to decide which of the characteristics named are most important and then to use these characteristics as a basis for the selection of an optimum intercooler design.

In the present investigation of intercooler design, the figure of merit used for the selection of the best design was <u>the total power consumed by the intercooler</u>. This value includes the power required to transport the weight of the intercooler as well as the power used to force the charge air and the cooling air through the intercooler. The cost, the size, and the practicability of construction were not considered, inasmuch as it was thought that a survey of possibilities of improvement in design would be of interest, regardless of whether the improvement could be immediately realized.

All the worth-while types of intercooler are included in the three types considered in this survey: a counterflow intercooler with indirect cooling surface in the form of fins (fig. 1(a)), a counterflow intercooler with direct cooling surfaces (fig. 1(b)), and a cross-flow tube-type intercooler (fig. 2).

SYMBOLS

A_t, total area of the cooling surface on which h_t is based, square feet.

C_D/C_L, the drag-lift ratio of the airplane in the assumed flying attitude.

c_p, specific heat (of air at constant pressure), B.t.u. per pound per $^{\circ}$F.

D, hydraulic diameter of passageway, feet.

$f_1 = (\Delta p/q)(D/4L)$, friction factor for the counterflow intercooler and for flow within the tube for the cross-flow intercooler.

$f_2 = (\Delta p/q)(1/4m)$, friction factor for the flow perpendicular to a tube bank, cross-flow intercooler.

g, the acceleration due to gravity, feet per second per second.

H_t, total heat transfer per second in the intercooler, B.t.u. per second.

h, surface heat-transfer coefficient, B.t.u. per second per square foot per $^{\circ}$F.

h_s, heat-transfer coefficient from air to metal for the counterflow indirect-cooling case based on the total wetted surface, B.t.u. per second per square foot per $^{\circ}$F.

h_t, over-all heat-transfer coefficient from fluid to fluid, B.t.u. per second per square foot per $^{\circ}$F.

k, thermal conductivity of air, B.t.u. per square foot per second per $^{\circ}$F. per foot.

k_m, thermal conductivity of the metal used in the construction of the intercooler, B.t.u. per square foot per second per $^{\circ}$F. per foot.

L, length of tubes in the cross-flow intercooler, or length of passageways in the counterflow intercooler, feet.

m, number of rows of tubes in the cross-flow intercooler.

M, flow of air per unit time, pounds per second.

N, counterflow, direct-cooling type, equal to one-half the total number of spaces.

n, number of tubes per row in the cross-flow intercooler.

Δp, cross-flow, total pressure drop through the intercooler, pounds per square foot. Counterflow, pressure drop per foot length, pounds per square foot per foot.

$P = P_t/A_t$, total power consumed per unit surface, foot-pounds per second per square foot.

P_t, total power consumed by the intercooler, foot-pounds per second.

P_c, P_h, cross-flow, total power required to force the cold, P_c, or the hot, P_h, air through the intercooler, foot-pounds per second; counterflow, power per unit cooling surface required to force the cold or the hot air through the intercooler, foot-pounds per second per square foot.

P_c', counterflow, equal to $P_c + \epsilon \frac{W_1}{2} \frac{C_D}{C_L} V_o$.

P_h', counterflow, equal to $P_h + \epsilon \frac{W_1}{2} \frac{C_D}{C_L} V_o$.

ϵ, a factor to multiply the intercooler weight to account for the additional required structural weight.

$q = \frac{1}{2} \rho V^2$, dynamic pressure in the passageways, pounds per square foot.

R, Reynolds Number, ($\rho \frac{VD}{\mu}$ for flow through a tube; $\rho \frac{V_{max} D}{\mu}$ for cross flow where D is the tube diameter and V_{max} is the flow velocity at the point of minimum width.)

s, spacing, as shown in figures 1(a) and 1(b), feet; and in figure 2, tube diameters.

T, temperature of the hot air, °F.

T', temperature of the cold air, °F.

V, air-flow velocity through the intercooler, feet per second.

V_o, flight velocity of the airplane, feet per second.

w, width of fin, or spacer, feet, as shown in figure 1.

W, total weight of the intercooler, pounds.

$W_1 = W/A_t$, total weight per unit surface, pounds per square foot.

t_m, thickness of metal between hot and cold fluids, feet.

ρV, mass flow of cold or hot air per second per square foot of open area, slugs per square foot per second.

ρVg, weight flow of cold or hot air per second per square foot of open area, pounds per square foot per second.

μ, coefficient of viscosity of air, slugs per foot-second.

η_p, pump efficiency: for the heated-air side, equal to the compressor efficiency; for the cooling-air side, equal to the ratio of the internal work done $(Q\Delta p)$ to the corresponding increase in drag multiplied by the flight velocity $(V_o \Delta D)$.

t_f, fin thickness, feet.

ρ, mass density of air, slugs per cubic foot.

ρ_m, specific weight of the metal used in the construction of the intercooler, pounds per cubic foot.

$\xi = \dfrac{T_i - T_o}{T_i - T_i'}$; relates to charge air.

$\eta = \dfrac{T_o' - T_i'}{T_i - T_i'}$; relates to cooling air.

ζ, the mean over-all temperature difference from charge air to cooling air for cross flow divided by $(T_i - T_i')$, given in table IV as a function of ξ and η.

Subscripts:

t,	total.	i,	inlet.
c,	cold air.	m,	metal.
h,	hot air.	o,	outlet.

SOURCES OF MATERIAL

Cross flow.- Nusselt has given the mathematical solution for heat transfer in a cross-flow intercooler (reference 1); Pierson (reference 2) has determined the heat-transfer coefficient and the friction factor for air flow across tube banks with various spacing arrangements. (See fig. 3.) By the use of this information and the data in figure 4 for heat transfer and friction factor on the inside of pipes (from reference 3), it is possible to calculate the dimensions of a tube-type cross-flow intercooler that will use minimum power for the operating conditions imposed.

Counterflow.- The mathematical equations for heat transfer in counterflow are given in reference 4, and the friction factors used for the counterflow intercoolers are included in figure 5. The solid line is for round tubes. The broken line is the theoretical curve for laminar flow between parallel plates, using $D = 2s$. The composite curve A may be used, with only a small percentage error, for rectangular ducts when the ratio of the lengths of the two sides is 20 or more. For smaller values of the ratio of the lengths of the sides, the friction factor for laminar flow may be obtained from reference 5 (p. 116).

McAdams (reference 5, p. 117) states that the available data on air, water, and oil indicate that the friction factor for turbulent flow in round pipes may be used to calculate the pressure drop in rectangular ducts by using the hydraulic diameter. (See also reference 3.) Data from pressure-loss tests on the segments of finned cylinders used for the heat-transfer tests reported in reference 6 are shown as points in figure 5.

The friction factor for turbulent flow being the same for round as for rectangular ducts, it follows from Reynolds analogy that the heat-transfer coefficient is the same in both kinds of tube. The same analogy indicates that, for laminar flow, the rectangular ducts should have a heat-transfer coefficient somewhat higher than that of a round tube. The heat-transfer data for round tubes (fig. 4) were used for both counterflow intercoolers because no heat-transfer data for laminar flow through rectangular ducts were available.

DESIGN CONDITIONS

In the design of an intercooler, there are known: the mass flow of hot air through the intercooler, the temperatures at which the hot and the cold air enter the intercooler, and the temperature at which the hot air is required to leave the intercooler.

In order to design the intercooler that will use least power, the flight velocity, the drag-lift ratio of the airplane for this viscosity, and the factor ϵ must be known to obtain the power used in transporting the intercooler weight. In addition, the pumping efficiencies must be known in order to calculate the total pumping power chargeable to the intercooler, and the altitude at which the maximum demand is made upon the intercooler must be known in order to determine the fluid constants of the cooling air.

For the purposes of demonstrating the method of design used in this investigation, the following conditions are assumed:

1. The brake horsepower of the engine is 1,000.

2. The engine uses 6,600 pounds of air per hour, or 1.833 pounds per second.

3. $T_i = 280^\circ$ F., $T_o = 80^\circ$ F., $T_i' = -30^\circ$ F.

4. The airplane is operating at the rated height of the engine, which is 25,000 feet.

5. V_o, the flight velocity of the airplane, is 300 miles per hour or 440 feet per second.

6. $\epsilon(C_D/C_L) = 0.075$.

7. η_p = 75 percent for both the cooling-air and the charge-air sides.

These assumed design conditions are based on the information given in reference 8 and are for an airplane with reasonable performance.

Fluid Constants Used

	Charge air	Cooling air
Density, slugs/ cu. ft.	0.001965 at 180° F.	0.000907 at 20° F.
Viscosity, slugs/ft.-sec.	0.443×10^{-6} at 180° F.	0.355×10^{-6} at 20° F.
Thermal conductivity, B.t.u./sec./ft.^{2}/°F./ft.	4.19×10^{-6} at 140° F.	3.61×10^{-6} at 40° F.
Specific heat, B.t.u./lb.	0.238	0.238
Pressure, in Hg	30.5	11.1

GENERAL CONSIDERATIONS

The design of an intercooler of any one of the three types mentioned is accomplished by considering the changes occurring in total power chargeable to the intercooler as the independent variables are systematically changed. The cross plot of the total power against each of these variables then enables the optimum intercooler design to be selected. The variables may include such quantities as weight flow of cooling air, M_c; the air flow per unit open frontal area, $(\rho Vg)_c$ or $(\rho Vg)_h$; dimensions such as tube length, L; tube diameter, D; spacing between fins or tubes, s; and so on. A careful choice of variables, however, will reduce computation to a minimum.

COUNTERFLOW INTERCOOLER

The equation connecting ξ, M_h, M_c, and h_tA_t for heat transfer in a counterflow intercooler was obtained from equations given in reference 4.

$$\xi = \frac{1 - e^{-h_tA_t\left(\frac{1}{M_hc_p} - \frac{1}{M_cc_p}\right)}}{1 - \frac{M_h}{M_c} e^{-h_tA_t\left(\frac{1}{M_hc_p} - \frac{1}{M_cc_p}\right)}} = \frac{T_i - T_o}{T_i - T_i'} \qquad (1)$$

Solutions for this equation, over a limited range, may be obtained from figure 6 (fig. 52 of reference 4).

Inasmuch as ξ and M_h are determined by the design conditions, h_tA_t may be determined in terms of M_c from from equation (1) for the counterflow intercooler. This relationship is shown in figure 7 for the design conditions assumed in this paper.

For each assumed value of M_c, it is required that the intercooler be so designed that P_t, the total power consumed by the intercooler, shall be a minimum, P_t/h_tA_t shall be a minimum or, since $P = P_t/A_t$, P/h_t shall be a minimum, where

$$\frac{P}{h_t} = \left(\frac{1}{h_h} + \frac{1}{h_c} + \frac{t_m}{k_m}\right)\left[P_h + P_c + W_1 \left(\epsilon \frac{C_D}{C_L} V_o\right)\right] \qquad (2)$$

Two types of counterflow intercooler have been considered in the present investigation. One type uses an indirect-cooling surface in the form of fins on each side of a central dividing plate (fig. 1(a)) and the other type uses only direct-cooling surfaces in the form of thin plates arranged in layers (fig. 1(b)). Both types are made of aluminum. The indirect-cooling-surface type will be considered first.

Counterflow intercooler with indirect-cooling surface (fig. 1(a)).- Six variables are necessary to determine the design. This case was not completely worked out because the large number of variables meant a very large number of designs to be calculated. Accordingly, the problem was simplified by assuming the fin width and spacing to be identical for both sides of the intercooler. The variables selected were $(\rho Vg)_c$, $(\rho Vg)_h$, fin spacing, and fin width. The advantage in selecting the weight flow of air per unit open frontal area $(\rho Vg)_c$ and $(\rho Vg)_h$, in preference to intercooler dimensions such as length, is that the resulting calculations are somewhat more straightforward.

Table I outlines the calculations for the design of a counterflow intercooler with indirect-cooling surface. Various simplifications are possible owing to the repetitions encountered. These simplifications are noted beneath the table and result in a decrease of more than three-fourths of the indicated calculations. The fin widths used in the calculations were 0.0417, 0.0833, and 0.1666 foot with fin spacing of 0.0042, 0.0083, 0.0208, and 0.0416 foot for each fin width used. The quantities $(\rho Vg)_h$ and $(\rho Vg)_c$ were given values of 1, 4, 7, and 10.

The hydraulic diameter, $D = \frac{4sw}{2(s+w)}$, can be calculated for any given fin spacing and width; then Reynolds Number, $R = \frac{\rho VD}{\mu}$, can be calculated for any value of ρV. After the Reynolds Number is known, Nusselt's number, $h_s D/k$, and hence h_s, can be obtained from fig. 4. The quantity h_s is the surface heat-transfer coefficient between air and surface in a rectangular channel of section s by w feet.

The formula for the heat-transfer coefficient based on unit area of the dividing plate for such a finned cooling surface is

$$h_h \text{ (or } h_c) = \frac{h_s}{s + t_f}\left(s + 2w' \frac{\tan h\ a\ w'}{a\ w'}\right) \qquad (3)$$

where

$$w' = w + t_f/2 \quad \text{and} \quad a = \sqrt{2h_s/k_m\ t_f}$$

$\frac{1}{s+t_f}$, number of fins per unit width of dividing plate.

$\left(2w' \frac{\tan h\ a\ w'}{a\ w'}\right)$, equivalent cooling area of one fin of unit length if the entire equivalent area is assumed to be at base temperature.

For each combination of fin spacing and fin width, the maximum value of h_h (or h_o) was obtained from a curve of h_h (or h_c) against t_f, thus giving the optimum fin thickness for each case. Illustrative curves of h_h (or h_c) against t_f are shown in figure 8.

The values of P_h' and P_c' are calculated from the curve of friction factor in figure 5 as follows. Obtain a value of $f_1 = (\Delta p/q)(D/4L)$ from figure 5 for the assumed values of ρV and D. Calculate the value of Δp per unit length from

$$\frac{\Delta p}{L} = \frac{4f_1 q}{D} \qquad (4)$$

Then

$$P_c = \frac{V_c \, \Delta \, p_c}{\eta_p}\left(\frac{s \; w}{s + t_f}\right) \qquad (5)$$

and

$$P_c' = \left[\frac{V_c \, \Delta \, p_c}{\eta_p}\left(\frac{s \; w}{s + t_f}\right) + \frac{W_1}{2}\left(\epsilon \, \frac{C_D}{C_L} \, V_o\right)\right] \qquad (6)$$

with similar equations for the hot-air side.

It is seen that the equation for P_c' (equation (6)) includes the power necessary to transport the part of the total weight of a square foot of dividing plate and its fins and cover that is chargeable to the cooling-air side. The weight was included in this manner merely as a convenience in computation. An equation similar to (6) holds for P_h' and

$$(P_c' + P_h') = P_h + P_c + W_1 \; \left(\epsilon \, \frac{C_D}{C_L} \, V_o\right) = P \qquad (7)$$

which is the last factor in equation (2).

If the value of M_c is taken from column 31 of table I, $h_t A_t$ may be found from figure 7, and h_t for the particular fin arrangement is given in column 29. Then A_t may be calculated, and $PA_t/550$ is the total horsepower consumed by the counterflow intercooler. This total power as a function of $(\rho Vg)_c$ and $(\rho Vg)_h$ is given in figure 9.

The smallest power consumption calculated for this type of intercooler was about 16 horsepower. This power could probably be further reduced by using a larger intercooler and a smaller air velocity through the intercooler. It is later shown (see table VI) that an intercooler of this type, which used 16 horsepower, was about as large as could reasonably be used. If this intercooler were used in a cross-flow instead of a counterflow type, the mean temperature difference available for cooling would be about 20 percent smaller; a corresponding increase of about 20 percent in cooling area, and hence in power consumed, would therefore be required.

Counterflow intercooler with direct-cooling surface (fig. 1(b)).- Four variables are necessary to determine the design: $(\rho Vg)_h$, $(\rho Vg)_c$, s_h, and s_c. In all calculations made on this type of intercooler, the spacings s_h and s_c are assumed to be equal. Table II shows the calculations required for the case of the counterflow intercooler with direct-cooling surface. Repetitions similar to those occurring in table I reduce the indicated work to about one-fourth. It is advantageous, from considerations of weight and heat flow through the metal, to have the individual metal plates as thin as possible. A thickness that seemed reasonable (0.0025 ft.) was selected and spacings of 0.0021, 0.0042, 0.0062, and 0.0083 foot were investigated. The quantities $(\rho Vg)_h$ and $(\rho Vg)_c$ were given values of 1, 4, 7, and 10.

The calculation for this type of intercooler is similar to that for the counterflow intercooler with indirect-cooling surface, except for the calculation of P and h_t, which are now based on the total heat-transfer surface. For this type intercooler, $D = 2s$.

Figure 10 shows the results obtained with the direct-cooling-surface type of intercooler. The smallest spacing used (0.0021 ft.) gives an intercooler that will consume about 13 horsepower, using values of $(\rho Vg)_c$ and $(\rho Vg)_h$ between 1 and 2. This power consumption might be reduced to about 8 horsepower if the weight were reduced by using 0.000417-foot metal in place of the 0.0025-foot metal used in this calculation. It seems practically impossible to use such thin metal in an intercooler of this type.

THE CROSS-FLOW INTERCOOLER

The required heat-transfer and friction data to compute the case of the cross-flow intercooler are shown in figures 3 and 4. It may be noted that, for the flow perpendicular to a tube bank (fig. 3), the Nusselt number was found to be independent of the spacing for all the cases where the spacing between tubes in a row equaled the spacing between rows (reference 2). For the present design, only cases in which these spacings are equal will be considered. Five variables are necessary to determine the design. For this case, the intercooler dimensions are selected as the variable factors. These dimensions are the tube length, the tube diameter, the spacing between the tubes, the number of tubes per row, and the number of rows of tubes. (See fig. 2.)

The procedure used in the calculations for the cross-flow intercooler may be most easily followed by reference to table III, which is a sample calculation for an intercooler with tubes 2 feet long and 1/48 foot in diameter.

As a preliminary step, figure 11 (which is of value in obtaining column 9 of table III) is derived as follows:

For a cross-flow intercooler, the quantity of heat transferred per second is given (reference 1) as

$$H_t = M_h c_p \left(T_i - T_o\right) = h_t A_t \zeta \left(T_i - T_i'\right) \qquad (8)$$

After a range of values of M_c has been assumed and ξ and η have been calculated from their definitions, ζ may be read directly from table IV (table 3 of reference 1). Thus, from equation (8), a value of $h_t A_t$ may be obtained for each assumed value of M_c; a curve of these is given in figure 7.

TABLE IV

Variation of ζ with η and ξ

(from reference 1, table 3)

ξ \ η	0	0.1	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.9	1.0
0	1.0	0.947	0.893	0.838	0.781	0.721	0.657	0.586	0.502	0.388	0
.1	.947	.893	.840	.786	.729	.670	.605	.533	.448	.338	0
.2	.893	.840	.785	.734	.677	.617	.552	.480	.398	.292	0
.3	.838	.786	.734	.682	.625	.565	.502	.430	.348	.247	0
.4	.781	.729	.677	.625	.569	.513	.449	.378	.300	.206	0
.5	.721	.670	.617	.565	.513	.456	.394	.326	.251	.167	0
.6	.657	.605	.552	.502	.449	.394	.334	.271	.201	.128	0
.7	.586	.533	.480	.430	.378	.326	.271	.213	.151	.089	0
.8	.502	.448	.398	.348	.300	.251	.201	.151	.100	.052	0
.9	.388	.338	.292	.247	.206	.167	.128	.089	.052	.022	0
1.0	0	0	0	0	0	0	0	0	0	0	0

For a given tube length and diameter, such as is assumed in this calculation, the variation of $1/h_c$ with $(\rho V)_c$ (fig. 11) is readily calculated from the heat-transfer data of figure 4 for the inside of the tube. The curve thus obtained holds for any number of tubes.

For any given number of tubes and weight flow on the cold side, M_c, A_t, and A_2 are known and h_t can be calculated by use of figure 7. Then $(\rho V)_c$ can be calculated from the equation

$$(\rho V)_c = \frac{M_c}{A_1 g} \tag{9}$$

and the variation of $1/h_t$ with $(\rho V)_c$ for several values of (mn) may therefore be obtained (fig. 11).

The data contained in figure 11, obtained in the foregoing manner, were used for the calculations in table III, where L and D were held constant and a series of values of m, n, and s were assumed.

The results of this sample calculation are presented in figures 12 and 13. It may be seen from these figures that the minimum power consumption calculated is about 10 horsepower. If this power calculation is repeated for different lengths and diameters of tubing and the results are cross-plotted, the best combination of the five intercooler variables and the mass flow of cooling air will be obtained.

DISCUSSION OF THE RESULTS

The curves given in figure 7 showing h_tA_t against M_c are the result of a mathematical analysis and will apply to any conceivable type of cross-flow or counterflow intercooler that is to meet the design conditions. This figure shows that any intercooler with the same values of friction factor and heat-transfer coefficient for all directions of air flow, such as the intercooler shown in figure 1(b), will be smaller, hence lighter and more efficient, if the air is in counterflow than if it is in cross flow. The counterflow intercooler has about a 20-percent advantage over the cross-flow intercooler under the operating conditions chosen for this analysis.

The foregoing considerations would seem to indicate that the best method of building an intercooler is to make it of the counterflow type. The assumption of equal friction factor and heat-transfer coefficient for both cross flow and counterflow, however, does not hold, as can be seen from figures 3 and 4.

The question remains whether flow across a bank of tubes is more or less efficient than flow parallel to a tube surface. Accordingly, calculations were made for several types of simplified heat exchangers for the purpose of comparing the optimum designs for each type under the same operating conditions. Table V gives the results of these calculations. The following points should be noted:

(a) Much higher heat dissipation per square foot of cooling surface is achieved in cross flow. That is, much less metal surface is required for equal amounts of heat dissipation.

(b) Cooling efficiencies are comparable for flow within tubes and for flow across a tube bundle.

TABLE V

A Comparison of Optimum Heat-Exchanger Designs for Air Flow within and across a Tube Bundle

(Δp = 25.6 lb./sq.ft.; energy dissipated, 250 hp.; [a]T_w - T_{ia} = 70° F.; η_p = 100 percent; air-metal heat-transfer coefficient taken equal to h_t)

Design type	Heat dissipation per unit cooling surface (hp./sq.ft.)	Required frontal area (sq.ft.)	Length (or depth) of design (ft.)	Cooling efficiency heat dissipated ÷ [b]pumping power cost
Air flow within round tubes, D = 1/48 ft.	0.56	3.74	1.04	19.0
Air flow within hexagonal tubes, D = 1/48 ft.	.51	2.92	1.17	20.7
Air flow across round tubes, D = 1/48 ft. s = 1/96 ft.	.98	12.2	.31	18.9
Air flow across round tubes, D = 1/48 ft. s = 1/48 ft.	.74	9.80	.92	21.8

[a]T_w is the temperature of the metal surfaces and is assumed to be constant throughout the heat exchanger, since water is on the other side of the heat-transfer surface.

[b]Pumping power cost is only the power required to push the air through; the power used in pumping water is not considered.

The calculations in table V show that the cross-flow tube-type heat exchanger overcomes its initial disadvantage of lower mean-temperature difference by dissipating more heat per unit area owing to a higher heat-transfer coefficient. The ratio of heat transfer to pumping power is about the same for flow either within or across the tubes. The smaller cooling area required by the cross-flow tube-type heat exchanger will thus result in a considerably lighter intercooler. This advantage of weight-saving is enough to overcome the 20-percent disadvantage in mean temperature difference and to give an intercooler requiring less total power.

It is of interest to note that the power loss in the duct between the supercharger and the intercooler bears the same ratio to the power loss through the hot-air side of the intercooler as the pressure drop in the duct bears to the pressure drop through the intercooler. An estimate of the power loss in the duct can be made by use of the numerical results in the sameple calculations, the pressure drop in the duct having been measured.

Although the calculations for the three types of intercoolers show the tube-type cross-flow intercooler to be superior on the basis of total power required, total power may not be the only consideration. Table VI shows that the power consumed by the intercooler varies between about 10 and 25 horsepower in the best designs calculated for the three types of intercooler considered. Whether the actual power consumption is 10 or 25 horsepower for a 1,000-horsepower engine, is not necessarily the determining consideration in the choice of an intercooler. Ruggedness of construction, ease of installation, frontal area, or length might easily be the deciding factor when the difference in total power is so small.

Obviously, any indirect-cooling arrangement can be improved by the substitution of a direct-cooling surface for an indirect-cooling surface. The required cooling surface is thus reduced which, in turn, reduces the weight and the power to pump the air through the exchanger.

TABLE VI

Summary of Designs Using Small Total Power, Taken from Tables I, II, and III

(Counterflow; indirect-cooling-surface-type intercooler)

$(\rho Vg)_c$	$(\rho Vg)_h$	Spacing, s (ft.)	Width, w (ft.)	Length, L (ft.)	Total power, P_t (hp.)	Frontal area (sq. ft.)
1	1	0.0042	0.0417	1.083	17.6	4.52
1	4	.0042	.0417	.529	23.3	4.52

(Counterflow, direct-cooling-surface-type intercooler)

$(\rho Vg)_c$	$(\rho Vg)_h$	Spacing, s (ft.)	Length, L (ft.)	Total power, P_t (hp.)	Frontal area (sq. ft.)
1	1	0.0021	0.39	18.6	8.02
1	1	.0042	1.10	25.3	5.85
1	4	.0021	.18	18.7	8.02
1	4	.0042	.51	18.1	5.85
4	4	.0021	.79	22.8	2.01
4	4	.0042	2.03	20.5	1.50

(Cross-flow, tube-type intercooler)

Length, L (ft.)	Diameter, D (ft.)	Spacing, s	Number of tubes per row, n	Number of rows, m	Total power P_t, (hp.)	Frontal area (sq.ft.)
2	1/48	1/4 D	50	40	[a]15.3	1.4
		1/2 D	45	47	10.5	2.1
		3/4 D	47	48	10.2	3.0

[a]Lowest value reached, not a minimum.

CONCLUDING REMARKS

The results show the importance of using the intercooler having optimum dimensions for the operating conditions specified. They further show the relative power cost associated with pumping the cooling and the charge air through the intercooler and the power to transport the intercooler. The relative importance of the various dimensions of the intercooler is also easily determined by an examination of the tables and the figures.

The total power to operate a well-designed intercooler is so small that other considerations, such as ease of installation or ruggedness of construction, might easily be the determining consideration in the choice of a type of intercooler.

Langley Memorial Aeronautical Laboratory,
National Advisory Committee for Aeronautics,
Langley Field, Va., May 26, 1939.

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FIGURE LEGENDS

(a) Fin-type counterflow intercooler.
(b) Counterflow intercooler with direct cooling surfaces.
Figure 1.- Lay-out dimensions of two counterflow types of intercooler.

Figure 2.- Tube lay-out dimensions of a cross-flow type of intercooler.

Figure 3.- Variation of Nusselt number hD/k and friction factor $\frac{\Delta p}{q}\frac{1}{4m}$ with Reynolds Number $D\rho V_{max}/\mu$ for air flow across tube banks (reference 2).

Figure 4.- Variation of Nusselt number hD/k and friction factor $\frac{\Delta p}{q}\frac{D}{4L}$ with Reynolds Number $D\rho V/\mu$ for air flow through circular tubes (reference 3).

Figure 5.- Variation of friction factor $\frac{\Delta p}{q}\frac{D}{4L}$ with Reynolds Number $D\rho V/\mu$ for fully developed air flow through circular and rectangular tubes.

Figure 6.- Graphical presentation of solutions to equation (1) (reference 4).

Figure 7.- Required values of heat-transfer coefficient and cooling surface as a function of the mass flow of cooling air for cross flow and for counterflow intercoolers. ξ, 0.645; M_h, 1.833 lb./sec.

Figure 8.- Illustration of the method of using equation (3) to determine the optimum fin thickness for a fin-type counterflow intercooler. Fin width, 1/24 ft.; (ρVg), 10.

Figure 9.- Variation of total horsepower consumed with $(\rho Vg)_c$ and $(\rho Vg)_h$ for a fin-type counterflow intercooler.

(a) Spacing, 0.0083 ft. (b) Spacing, 0.0063 ft.
(c) Spacing, 0.0042 ft. (d) Spacing, 0.0021 ft.
Figure 10.- Variation of horsepower consumed with $(\rho Vg)_c$ for several values of $(\rho Vg)_h$ and spacing in a direct-cooling-surface counterflow intercooler.

Figure 11.- Variation of $1/h_t$ and $1/h_c$ with $(\rho V)_c$ in a tube-type cross-flow intercooler.

Figure 12.- Variation of horsepower consumed with m for several values of n and s in a tube-type cross-flow intercooler.

Figure 13.- Variation of horsepower consumed with n for several values of spacing, and with spacing for several values of n in a tube-type cross-flow intercooler.

TABLE I. OUTLINE OF CALCULATIONS FOR THE DESIGN OF A COUNTERFLOW INDIRECT-COOLING-SURFACE TYPE OF INTERCOOLER

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16
					Charge-air side										
$(\rho Vg)_h$ (a)	$(\rho Vg)_c$ (a)	w (ft.)	s (ft.)	$D = \frac{2sw}{s+w}$ (ft.)	$R_h = \frac{(\rho V)_h D}{\mu}$	$h_s D/k_h$ (b)	$h_s = \left(\frac{h_s D}{k_h}\right)\frac{k_h}{D}$	$1/h_h$ (c)	t_{f_h} (c) (ft.)	f_1 (d)	$q_h = \frac{(\rho Vg)^2}{2\rho_h g^2}$ $\left(\frac{lb.}{sq.ft.}\right)$	$\Delta p_h = \frac{4f_1 q_h}{D}$ (per ft. length) (lb./sq.ft./ft.)	$P_h = \frac{V_h}{\eta_p}\frac{sw}{s+t_{f_h}}\Delta p_h$ (ft.-lb./sec.)	$W_h = \rho_m\left(\frac{t_m}{2} + \frac{wt_{f_h}}{s+t_{f_h}}\right)$ (lb.)	$P_h' = P_h + W_h(\epsilon V_o C_D/C_L)$ (ft.-lb./sec.)
1	1	0.0417	.0042	.0076	552	3.29	1.81×10^{-3}	32.9	.0004	.0434	.245	5.596	4.612	2.486	86.65
			.0083	.0139	1010	4.40	1.33	74.6	.0005	.0239	.245	1.685	1.412	2.136	71.90
			.0208	.0278	2020	6.30	.95	224.1	.0008	.0117	.245	.412	.350	1.987	65.92
			.0416	.0417	3030	8.35	.84	410.0	.0010	.0106	.245	.249	.213	1.913	63.34
		.0833	.0042	.0079	574	3.34	1.77	23.9	.0010	.0418	.245	5.185	7.352	4.343	150.67
			.0083	.0151	1097	4.59	1.27	48.1	.0011	.0222	.245	1.441	2.228	3.366	113.36
			.0208	.0333	2420	7.03	.88	143.8	.0016	.0112	.245	.340	.551	2.731	90.67
			.0416	.0556	4040	10.95	.82	265.1	.0022	.0098	.245	.173	.289	2.446	81.01
		.1666	.0042	.0081	588	3.39	1.75	15.0	.0018	.0410	.245	4.960	12.190	10.178	348.06
			.0083	.0159	1145	4.69	1.24	31.8	.0023	.0210	.245	1.294	3.549	7.794	260.75
			.0208	.0370	2690	7.60	.86	88.9	.0030	.0110	.245	.291	.894	5.293	175.56
			.0416	.0666	4840	13.50	.85	220.2	.0038	.0094	.245	.138	.444	4.092	135.48
	4	0.0417	.0042	.0076	552	3.29	1.81	32.9	.0004	.0434	.245	5.596	4.612	2.486	86.65
			.0083	.0139	1010	4.40	1.33	74.6	.0005	.0239	.245	1.685	1.412	2.136	71.90
			.0208	.0278	2020	6.30	.95	224.1	.0008	.0117	.245	.412	.350	1.987	65.92
			.0416	.0417	3030	8.35	.84	410.0	.0010	.0106	.245	.249	.213	1.913	63.34
		.0833	.0042	.0079	574	3.34	1.77	23.9	.0010	.0418	.245	5.185	7.352	4.343	150.67
			.0083	.0151	1097	4.59	1.27	48.1	.0011	.0222	.245	1.441	2.228	3.366	113.36
			.0208	.0333	2420	7.03	.88	143.8	.0016	.0112	.245	.340	.551	2.731	90.67
			.0416	.0556	4040	10.95	.82	265.1	.0022	.0098	.245	.173	.289	2.446	81.01
		.1666	.0042	.0081	588	3.39	1.75	15.0	.0018	.0410	.245	4.960	12.190	10.178	348.06
			.0083	.0159	1145	4.69	1.24	31.8	.0023	.0210	.245	1.294	3.549	7.794	260.75
			.0208	.0370	2690	7.60	.86	88.9	.0030	.0110	.245	.291	.894	5.293	175.56
			.0416	.0666	4840	13.50	.85	220.2	.0038	.0094	.245	.138	.444	4.092	135.48
	7	0.0417	.0042	.0076	552	3.29	1.81	32.9	.0004	.0434	.245	5.596	4.612	2.486	86.65
			.0083	.0139	1010	4.40	1.33	74.6	.0005	.0239	.245	1.685	1.412	2.136	71.90
			.0208	.0278	2020	6.30	.95	224.1	.0008	.0117	.245	.412	.350	1.987	65.92
			.0416	.0417	3030	8.35	.84	410.0	.0010	.0106	.245	.249	.213	1.913	63.34
		.0833	.0042	.0079	574	3.34	1.77	23.9	.0010	.0418	.245	5.185	7.352	4.343	150.67
			.0083	.0151	1097	4.59	1.27	48.1	.0011	.0222	.245	1.441	2.228	3.366	113.36
			.0208	.0333	2420	7.03	.88	143.8	.0016	.0112	.245	.340	.551	2.731	90.67
			.0416	.0556	4040	10.95	.82	265.1	.0022	.0098	.245	.173	.289	2.446	81.01
		.1666	.0042	.0081	588	3.39	1.75	15.0	.0018	.0410	.245	4.960	12.190	10.178	348.06
			.0083	.0159	1145	4.69	1.24	31.8	.0023	.0210	.245	1.294	3.549	7.794	260.75
			.0208	.0370	2690	7.60	.86	88.9	.0030	.0110	.245	.291	.894	5.293	175.56
			.0416	.0666	4840	13.50	.85	220.2	.0038	.0094	.245	.138	.444	4.092	135.48
4	4	0.0417	.0042	.0076	2210	6.65	3.67	17.7	.0007	.0115	3.926	23.763	73.666	2.784	165.53
			.0083	.0139	4040	11.00	3.32	33.6	.0012	.0098	3.926	11.072	34.426	2.640	121.55
			.0208	.0278	8080	23.80	3.59	62.9	.0017	.0082	3.926	4.632	15.118	2.277	90.33
			.0416	.0417	12120	33.90	3.41	105.2	.0025	.0075	3.296	2.824	9.350	2.136	79.84
		.0833	.0042	.0079	2295	6.82	3.62	12.0	.0012	.0114	3.926	22.661	123.955	4.874	284.79
			.0083	.0151	4390	12.00	3.33	21.2	.0017	.0097	3.926	10.088	58.748	4.112	194.49
			.0208	.0333	9670	27.90	3.51	40.7	.0029	.0079	3.926	3.726	22.957	3.460	137.14
			.0416	.0556	16150	42.50	3.20	74.0	.0042	.0069	3.926	1.949	12.518	3.047	113.07
		.1666	.0042	.0081	2350	6.91	3.58	9.3	.0025	.0113	3.926	21.908	193.132	12.318	599.62
			.0083	.0159	4620	12.80	3.37	15.1	.0042	.0095	3.926	9.383	87.429	11.189	456.67
			.0208	.0370	10750	30.90	3.50	27.5	.0058	.0068	3.926	2.716	29.903	7.898	290.53
			.0416	.0666	19350	49.00	3.08	50.0	.0083	.0066	3.926	1.558	18.239	6.414	229.90

a) The values of $(\rho Vg)_h$ and $(\rho Vg)_c$ are varied through the range 1, 4, 7, and 10.

b) Figure 4 shows $h_s D/k$ as a function of R.

c) By the use of equation (3), a curve of $1/h_c$ against t_f is drawn. The minimum value of $1/h_c$ and the corresponding value of t_f are chosen.

d) Figure 5 shows f_1 as a function of R.

e) Multiply columns 28 and 29. By inspection, the minimum value of P/h_t is selected for each combination of $(\rho Vg)_h$ and $(\rho Vg)_c$. Columns 31 to 34 need be calculated for only these minimum values.

f) Figure 7 shows $h_t A_t$ as a function of M_c for counterflow.

TABLE I (continued)

17	18	19	20	21	22	23	24	25	26	27	28	29	30	31	32	33	34
Cooling-air side																	
$R = \frac{(\rho V)_o D}{\mu}$	$h_s D/k_o$	h_s	$1/h_o$	t_{f_o}	f_2	$q_o = \frac{(\rho Vg)_o^2}{2\rho_o g^2}$	Δp_o (per ft. length)	$P_o = \frac{V_o}{\eta_p} \frac{sw}{s+t_{f_o}} \Delta p_o$	$W_o = \rho_m \left(\frac{t_m}{2} + \frac{wt_{f_o}}{s+t_{f_o}}\right)$	$P_o' = P_o + W_o\ (\epsilon V_o C_D/C_L)$	$P = P_h' + P_o'$	$\frac{1}{h_t} = \frac{1}{h_h} + \frac{1}{h_o} + \frac{t_m}{k_m}$	P/h_t	$M_o = M_h \frac{(\rho Vg)_o}{(\rho Vg)_h}$	$h_t A_t$	A_t	$\frac{P_t}{550} = \frac{PA_t}{550}$
	(b)		(c)	(c) (ft.)	(d)	$\left(\frac{\text{lb.}}{\text{sq.ft.}}\right)$	(lb./ sq.ft./ ft.)	(ft.-lb. /sec.)	(lb.)	(ft.-lb. /sec.)	(ft.-lb. /sec.)		(e)	(lb./ sec.)	(f)	(sq.ft.)	(hp.)
690	3.68	1.75×10^{-3}	32.9	.0004	.0345	.532	9.660	17.237	2.486	99.28	185.93	66.1	8463	1.833	.79	52.2	17.6
1260	4.90	1.27	74.6	.0005	.0190	.532	2.909	5.276	2.136	75.77	147.67	149.5	13422	1.833	.79	118.1	31.7
2520	7.25	.94	224.1	.0008	.0112	.532	.857	1.574	1.987	67.14	133.06	448.5	33718	1.833	.79	354.3	85.7
3780	10.15	.88	410.0	.0010	.0100	.532	.510	.945	1.913	64.08	127.42	820.3	57035	1.833	.79	648.0	150.1
717	3.72	1.70	23.9	.0010	.0334	.532	8.997	27.619	4.343	170.94	321.61	48.1	12685	1.833	.79	38.0	22.2
1370	5.10	1.22	48.1	.0011	.0176	.532	2.480	8.301	3.366	119.38	232.74	96.5	16868	1.833	.79	76.2	32.2
3020	8.35	.90	143.8	.0016	.0106	.532	.677	2.384	2.731	92.50	183.17	287.9	36074	1.833	.79	227.4	75.7
5040	14.15	.92	265.1	.0022	.0093	.532	.356	1.286	2.446	82.01	163.02	530.5	55771	1.833	.79	419.1	124.2
735	3.80	1.69	15.0	.0018	.0327	.532	8.591	45.702	10.178	381.57	729.63	30.3	20354	1.833	.79	23.9	31.7
1440	5.22	1.19	31.8	.0023	.0168	.532	2.248	13.346	7.794	270.55	531.30	63.9	30252	1.833	.79	50.5	48.8
3360	9.05	.88	88.9	.0030	.0103	.532	.592	3.935	5.293	178.60	354.16	178.1	52762	1.833	.79	140.7	90.6
6040	17.50	.95	220.2	.0038	.0089	.532	.284	1.977	4.092	137.02	272.50	440.7	94579	1.833	.79	348.2	172.5
2760	7.80	3.70	17.7	.0007	.0108	8.506	48.350	324.456	2.784	416.33	502.98	50.9	22656	7.332	.50	25.5	23.3
5040	14.15	3.67	33.6	.0012	.0093	8.506	22.764	153.217	2.640	240.34	312.24	108.5	27598	7.332	.50	54.3	30.8
10100	29.10	3.78	62.9	.0017	.0078	8.506	9.546	67.443	2.277	142.58	208.50	287.3	43273	7.332	.50	143.6	54.4
15130	40.20	3.48	105.2	.0025	.0070	8.506	5.711	40.933	2.136	111.42	174.76	515.5	60252	7.332	.50	257.8	81.9
2870	8.00	3.66	12.0	.0012	.0107	8.506	46.083	545.659	4.874	706.50	857.17	36.2	28934	7.332	.50	18.1	28.2
5480	15.50	3.71	21.2	.0017	.0092	8.506	20.730	261.329	4.112	397.03	510.39	69.6	31490	7.332	.50	34.8	32.3
12080	33.90	3.68	40.7	.0029	.0074	8.506	7.561	100.842	3.460	215.02	305.69	184.8	45795	7.332	.50	92.4	51.3
20180	50.90	3.30	74.0	.0042	.0066	8.506	4.039	56.156	3.047	156.71	237.72	339.4	61038	7.332	.50	169.7	73.3
2940	8.15	3.63	9.3	.0025	.0107	8.506	44.945	857.686	12.318	1264.17	1612.23	24.6	38237	7.332	.50	12.3	36.0
5770	16.50	3.75	15.1	.0042	.0090	8.506	19.259	388.456	11.189	757.70	1018.45	47.2	45338	7.332	.50	23.6	43.7
13430	36.60	3.57	27.5	.0058	.0073	8.506	6.713	159.992	7.898	420.62	596.18	116.7	62821	7.332	.50	58.4	63.3
24180	58.80	3.19	50.0	.0083	.0063	8.506	3.218	81.652	6.414	293.31	428.79	270.5	100331	7.332	.50	135.2	105.4
4830	13.50	6.41	11.1	.0008	.0094	26.050	128.879	1482.830	2.877	1577.77	1664.42	44.3	71170	12.831	.47	20.8	62.9
8830	25.70	6.67	17.9	.0015	.0081	26.050	60.721	714.933	2.818	807.92	879.82	92.8	76277	12.831	.47	43.6	69.7
17650	45.50	5.91	36.2	.0021	.0068	26.050	25.488	309.465	2.415	389.15	455.07	260.6	103510	12.831	.47	122.5	101.3
26500	63.00	5.46	66.6	.0029	.0061	26.050	15.243	189.410	2.208	262.27	325.61	476.9	127680	12.831	.47	224.1	132.7
5020	14.05	6.42	7.8	.0017	.0093	26.050	122,666	2325.976	5.752	2515.80	2666.47	32.0	83475	12.831	.47	15.0	72.7
9590	27.50	6.57	12.4	.0025	.0079	26.050	54.515	1113.155	5.029	1279.11	1392.47	60.8	81139	12.831	.47	28.6	72.4
21150	52.50	5.69	26.7	.0042	.0065	26.050	20.339	449.824	4.112	585.52	676.19	170.8	105607	12.831	.47	80.3	98.7
35300	79.80	5.18	47.2	.0054	.0057	26.050	10.682	252.053	3.366	363.13	444.14	312.6	120745	12.831	.47	146.9	118.6
5140	14.45	6.44	6.5	.0028	.0093	26.050	119.637	3823.596	13.066	4254.78	4602.84	21.8	99080	12.831	.47	10.2	85.4
10100	29.10	6.60	9.5	.0042	.0078	26.050	51.117	1803.604	11.189	2172.84	2433.59	41.6	98830	12.831	.47	19.6	86.7
23500	57.50	5.61	19.1	.0075	.0064	26.050	18.024	706.464	9.196	1009.93	1185.49	108.3	122120	12.831	.47	50.9	109.7
42300	92.00	4.99	33.4	.0100	.0037	26.050	5.789	248.477	7.206	486.28	621.76	253.9	143167	12.831	.47	119.3	134.9
2760	7.80	3.70	17.7	.0007	.0108	8.506	48.350	324.456	2.784	416.33	581.86	35.7	18706	1.833	.79	28.2	29.8
5040	14.15	3.67	33.6	.0012	.0093	8.506	22.764	153.217	2.640	240.34	361.89	67.5	20520	1.833	.79	53.3	35.1
10100	29.10	3.78	62.9	.0017	.0078	8.506	9.546	67.443	2.277	142.58	232.91	126.1	22062	1.833	.79	99.6	42.2
15130	40.20	3.48	105.2	.0025	.0070	8.506	5.711	40.933	2.136	111.42	191.26	210.7	28103	1.833	.79	166.5	57.9
2870	8.00	3.66	12.0	.0012	.0107	8.506	46.083	545.659	4.874	706.50	991.29	24.3	22682	1.833	.79	19.2	34.6
5480	15.50	3.71	21.2	.0017	.0092	8.506	20.730	261.329	4.112	397.03	591.52	42.7	22784	1.833	.79	33.7	36.2
12080	33.90	3.68	40.7	.0029	.0074	8.506	7.561	100.842	3.460	215.02	352.16	81.7	24043	1.833	.79	64.5	41.3
20180	50.90	3.30	74.0	.0042	.0066	8.506	4.039	56.156	3.047	156.71	269.78	148.3	31425	1.833	.79	117.2	57.5
2940	8.15	3.63	9.3	.0025	.0107	8.506	44.945	857.686	12.318	1264.17	1863.79	18.9	34132	1.833	.79	14.9	50.5
5770	16.50	3.75	15.1	.0042	.0090	8.506	19.259	388.456	11.189	757.70	1214.37	30.5	35273	1.833	.79	24.1	53.2
13430	36.60	3.57	27.5	.0058	.0073	8.506	6.713	159.992	7.898	420.62	711.15	55.3	36127	1.833	.79	43.7	56.5
24180	58.80	3.19	50.0	.0083	.0063	8.506	3.218	81.652	6.414	293.31	523.21	100.3	46673	1.833	.79	79.2	75.3

Simplifications

1. Note from column 31 that $M_o = M_h \frac{(\rho Vg)_c}{(\rho Vg)_h}$. Figure 7 shows that the design conditions do not permit values of M_o below 1.182 lb. per sec. Since M_h = 1.833 lb. per sec., *the following combinations* may be omitted: $(\rho Vg)_h$, $(\rho Vg)_c$ = 4,1; 7,1; 7,4; 10,1; and 10,4.
2. Columns 6 to 16 depend only on $(\rho Vg)_h$ and may be used in combination with any value of $(\rho Vg)_c$.
3. Columns 17 to 27 depend only on $(\rho Vg)_c$ and may be used in combination with any value of $(\rho Vg)_h$.

TABLE II. OUTLINE OF CALCULATIONS FOR THE DESIGN OF A COUNTERFLOW DIRECT-COOLING-SURFACE TYPE OF INTERCOOLER

(For simplifications, see Table I)

1	2	3	4	5	6	7	8	9	10	11	12	13	14
					Charge-air-side								
$(\rho Vg)_h$ (a)	$(\rho Vg)_c$ (a)	s_h (a) (ft.)	s_c (a) (ft.)	$N_W = \frac{W_h}{(\rho Vg)_h s_h}$	$R = \frac{2s_h(\rho V)_h}{\mu}$	hD/k_h (b)	$1/b_h$	f_1 (c)	$q_h = \frac{(\rho V)_h^2}{2\rho_h}$ $\left(\frac{\text{lb.}}{\text{sq.ft.}}\right)$	$\Delta p_h = \frac{2f_1 q_h}{s_h}$ (per ft. length) (lb./sq.ft./ft.)	$P_h = \frac{\Delta p_h V_h s_h}{2\eta_p}$ (ft.-lb./sec.)	V_h (ft./sec.)	$P_h' = P_h + W_h \epsilon V_o(C_D/C_L)$ (ft.-lb./sec.)
1	1	0.0021	.0021	872	298	2.42	414.1	.0533	.25	12.56	.28	15.8	7.21
		.0042	.0042	436	595	3.40	589.3	.0275	.25	3.24	.14	15.8	7.07
		.0063	.0063	291	893	4.18	718.4	.0185	.25	1.45	.10	15.8	7.03
		.0083	.0083	221	1190	4.75	835.4	.0142	.25	.84	.07	15.8	7.00
	4	.0021	.0021	872	298	2.42	414.1	.0533	.25	12.56	.28	15.8	7.21
		.0042	.0042	436	595	3.40	589.3	.0275	.25	3.24	.14	15.8	7.07
		.0063	.0063	291	893	4.18	718.4	.0185	.25	1.45	.10	15.8	7.03
		.0083	.0083	221	1190	4.75	835.4	.0142	.25	.84	.07	15.8	7.00
	7	.0021	.0021	872	298	2.42	414.1	.0533	.25	12.56	.28	15.8	7.21
		.0042	.0042	436	595	3.40	589.3	.0275	.25	3.24	.14	15.8	7.07
		.0063	.0063	291	893	4.18	718.4	.0185	.25	1.45	.10	15.8	7.03
		.0083	.0083	221	1190	4.75	835.4	.0142	.25	.84	.07	15.8	7.00
	10	.0021	.0021	872	298	2.42	414.1	.0533	.25	12.56	.28	15.8	7.21
		.0042	.0042	436	595	3.40	589.3	.0275	.25	3.24	.14	15.8	7.07
		.0063	.0063	291	893	4.18	718.4	.0185	.25	1.45	.10	15.8	7.03
		.0083	.0083	221	1190	4.75	835.4	.0142	.25	.84	.07	15.8	7.00
4	4	0.0021	.0021	218	1190	4.75	211.0	.0142	3.93	53.55	4.75	63.3	11.68
		.0042	.0042	112	2381	7.00	286.3	.0072	3.93	13.58	2.41	63.3	9.34
		.0063	.0063	73	3571	9.60	312.8	.0059	3.93	7.42	1.97	63.3	8.90
		.0083	.0083	55	4762	13.20	300.7	.0083	3.93	7.84	2.75	63.3	9.68
	7	.0021	.0021	218	1190	4.75	211.0	.0142	3.93	53.55	4.75	63.3	11.68
		.0042	.0042	112	2381	7.00	286.3	.0072	3.93	13.58	2.41	63.3	9.34
		.0063	.0063	73	3571	9.60	312.8	.0059	3.93	7.42	1.97	63.3	8.90
		.0083	.0083	55	4762	13.20	300.7	.0083	3.93	7.84	2.75	63.3	9.68
	10	.0021	.0021	218	1190	4.75	211.0	.0142	3.93	53.55	4.75	63.3	11.68
		.0042	.0042	112	2381	7.00	286.3	.0072	3.93	13.58	2.41	63.3	9.34
		.0063	.0063	73	3571	9.60	312.8	.0059	3.93	7.42	1.97	63.3	8.90
		.0083	.0083	55	4762	13.20	300.7	.0083	3.93	7.84	2.75	63.3	9.68
7	7	0.0021	.0021	125	2083	6.42	156.1	.0082	12.03	94.75	14.70	110.8	21.63
		.0042	.0042	62	4167	11.80	169.8	.0071	12.03	41.00	12.72	110.8	19.65
		.0063	.0063	42	6250	18.08	166.1	.0093	12.03	35.81	16.66	110.8	23.59
		.0083	.0083	32	8333	24.15	164.3	.0086	12.03	24.86	15.24	110.8	22.17
	10	.0021	.0021	125	2083	6.42	156.1	.0082	12.03	94.75	14.70	110.8	21.63
		.0042	.0042	62	4167	11.80	169.8	.0071	12.03	41.00	12.72	110.8	19.65
		.0063	.0063	42	6250	18.08	166.1	.0093	12.03	35.81	16.66	110.8	23.59
		.0083	.0083	32	8333	24.15	164.3	.0086	12.03	24.86	15.24	110.8	22.17
10	7	0.0021	.0021	87	2976	8.20	122.2	.0058	24.60	136.80	30.30	158.2	37.23
		.0042	.0042	44	5952	17.00	117.9	.0095	24.60	112.10	49.66	158.2	56.59
		.0063	.0063	29	8929	25.85	116.2	.0084	24.60	66.00	43.85	158.2	50.78
		.0083	.0083	22	11905	33.40	118.8	.0078	24.60	46.00	40.27	158.2	47.20
	10	.0021	.0021	87	2976	8.20	122.2	.0058	24.60	136.80	30.30	158.2	37.23
		.0042	.0042	44	5952	17.00	117.9	.0095	24.60	112.10	49.66	158.2	37.23
		.0063	.0063	29	8929	25.85	116.2	.0084	24.60	66.00	43.85	158.2	37.23
		.0083	.0083	22	11905	33.40	118.8	.0078	24.60	46.00	40.27	158.2	37.23

a)The values of $(\rho Vg)_h$ and $(\rho Vg)_c$ are varied through the range 1, 4, 7, and 10; the values of s_h and s_c are varied through the range 0.0021, 0.0042, 0.0062, and 0.0083.

b)Figure 4 shows hD/k as a function of R.

c)Figure 5 shows f_1 as a function of R.

TABLE II (continued)

15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30	31
Cooling-air side																
$R = \frac{2s_c (\rho V)_c}{\mu}$	hD/k_c	$1/h_c$	f_2	q_c	Δp_c (per ft. length)	P_c	V_c	P_c'	$P = P_h' + P_c'$	$\frac{1}{h_t} = \frac{1}{h_c} + \frac{1}{h_h} + \frac{t_m}{k_m}$	P/h_t	$M_c = M_h \frac{s_c}{s_h} \frac{(\rho V)_c}{(\rho V)_h}$	$h_t A_t$	A_t	$L = \frac{A_t}{2NW}$	$\frac{P_t}{550} = \frac{PA_t}{550}$
	(b)		(c)	$\left(\frac{\text{lb.}}{\text{sq.ft.}}\right)$	(lb./sq.ft./ft.)	(ft.-lb./sec.)	(ft./sec.)	(ft.-lb./sec.)	$\left(\frac{\text{ft.-lb.}}{\text{sec.ft.}^2}\right)$		(d)	(lb./sec.)	(e)	(sq.ft.)	(ft.)	(hp.)
365	2.68	433.8	.0440	0.53	22.48	1.08	34.2	8.01	15.22	848.0	1.29×10^4	1.833	.794	673.3	0.39	18.6
731	3.78	615.4	.0225	.53	5.75	.55	34.2	7.48	14.55	1204.8	1.75	1.833	.794	956.6	1.10	25.3
1096	4.57	761.6	.0152	.53	2.59	.37	34.2	7.30	14.33	1480.1	2.12	1.833	.794	1175.2	3.01	30.6
1462	5.28	872.6	.0115	.53	1.46	.28	34.2	7.21	14.21	1708.1	2.43	1.833	.794	1356.2	3.07	35.0
1462	5.28	220.2	.0115	8.50	93.95	18.02	137.0	24.95	32.16	634.4	2.04	7.332	.505	320.4	.18	18.7
2924	8.10	287.1	.0054	8.50	22.06	8.46	137.0	15.39	22.46	876.5	1.97	7.332	.505	442.6	.51	18.1
4386	11.98	290.9	.0076	8.50	20.68	11.90	137.0	18.83	25.86	1009.4	2.61	7.332	.505	509.7	.88	24.0
5848	16.75	275.0	.0094	8.50	19.20	14.55	137.0	21.48	28.48	1110.5	3.16	7.332	.505	560.8	1.27	29.0
2558	7.39	157.3	.0067	26.04	167.60	56.24	239.7	63.17	70.38	571.5	4.02	12.831	.477	272.6	.16	34.9
5117	14.45	160.9	.0091	26.04	113.70	76.31	239.7	83.24	90.31	750.3	6.78	12.831	.477	357.9	.41	58.8
7675	22.50	154.8	.0088	26.04	73.30	73.79	239.7	80.72	87.75	873.3	7.66	12.831	.477	416.6	.72	66.5
10234	29.25	157.8	.0082	26.04	51.25	67.97	239.7	74.90	81.90	993.3	8.14	12.831	.477	473.8	1.07	70.5
3655	9.83	118.3	.0060	53.20	306.30	146.87	342.5	153.80	161.01	532.5	8.57	18.330	.463	246.5	.14	72.2
7310	21.25	109.4	.0089	53.20	227.10	217.79	342.5	224.72	231.79	698.8	16.20	18.330	.463	323.5	.37	136.3
10965	31.10	112.0	.0080	53.20	136.10	195.78	342.5	202.71	209.74	830.5	17.42	18.330	.463	384.5	.66	146.6
14620	39.20	117.6	.0074	53.20	94.40	178.90	342.5	185.83	192.83	953.1	18.38	18.330	.463	441.3	1.00	154.7
1462	5.28	220.2	.0115	8.50	93.95	18.02	137.0	24.95	36.63	431.3	1.58	1.833	.794	342.5	.79	22.8
2924	8.10	287.1	.0054	8.50	22.06	8.46	137.0	15.39	24.73	573.4	1.42	1.833	.794	455.3	2.03	20.5
4386	11.98	290.9	.0076	8.50	20.68	11.90	137.0	18.83	27.73	603.8	1.67	1.833	.794	479.4	3.28	24.2
5848	16.75	275.0	.0094	8.50	19.20	14.55	137.0	21.48	31.16	675.8	2.11	1.833	.794	536.6	4.88	30.4
2558	7.39	157.3	.0067	26.04	167.60	56.24	239.7	63.17	74.85	368.4	2.76	3.208	.605	222.9	.51	30.3
5117	14.45	160.9	.0091	26.04	113.70	76.31	239.7	83.24	92.58	447.3	4.14	3.208	.605	270.6	1.21	45.5
7675	22.50	154.8	.0088	26.04	73.30	73.79	239.7	80.72	89.62	467.7	4.19	3.208	.605	283.0	1.94	46.1
10234	29.25	157.8	.0082	26.04	51.25	67.97	239.7	74.90	84.58	458.6	3.88	3.208	.605	277.5	2.52	42.7
3655	9.83	118.3	.0060	53.20	306.30	146.87	342.5	153.80	165.48	329.4	5.45	4.582	.555	182.8	.42	55.0
7310	21.25	109.4	.0089	53.20	227.10	217.79	342.5	224.72	234.06	395.8	9.26	4.582	.555	219.7	.98	93.5
10965	31.10	112.0	.0080	53.20	136.10	195.78	342.5	202.71	211.61	424.9	8.99	4.582	.555	235.8	1.62	90.7
14620	39.20	117.6	.0074	53.20	94.40	178.90	342.5	185.83	195.51	418.4	8.18	4.582	.555	232.2	2.11	82.5
2558	7.39	157.3	.0067	26.04	167.60	56.24	239.7	63.17	84.80	313.5	2.66	1.833	.794	248.9	1.00	38.4
5117	14.45	160.9	.0091	26.04	113.70	76.31	239.7	83.24	102.89	330.8	3.40	1.833	.794	262.7	2.12	49.1
7675	22.50	154.8	.0088	26.04	73.30	73.79	239.7	80.72	104.31	321.0	3.35	1.833	.794	254.9	3.04	48.3
10234	29.25	157.8	.0082	26.04	51.25	67.97	239.7	74.90	97.07	322.2	3.13	1.833	.794	255.8	4.00	45.1
3655	9.83	118.3	.0060	53.20	306.30	146.87	342.5	153.80	175.43	274.5	4.82	2.619	.640	175.7	.70	56.0
7310	21.25	109.4	.0089	53.20	227.10	217.79	342.5	224.72	244.37	279.3	6.83	2.619	.640	178.8	1.44	79.4
10965	31.10	112.0	.0080	53.20	136.10	195.78	342.5	202.71	226.30	278.2	6.30	2.619	.640	178.0	2.12	73.2
14620	39.20	117.6	.0074	53.20	94.40	178.90	342.5	185.83	208.00	282.0	5.87	2.619	.640	180.5	2.82	68.3
2558	7.39	157.3	.0067	26.04	167.60	56.24	239.7	63.17	100.40	279.6	2.81	1.283	1.530	427.8	2.46	78.1
5117	14.45	160.9	.0091	26.04	113.70	76.31	239.7	83.24	139.83	278.9	3.90	1.283	1.530	426.7	4.85	108.5
7675	22.50	154.8	.0088	26.04	73.30	73.79	239.7	80.72	131.50	271.1	3.56	1.283	1.530	414.8	7.15	99.2
10234	29.25	157.8	.0082	26.04	51.25	67.97	239.7	74.90	122.10	276.7	3.38	1.283	1.530	423.4	9.62	94.0
3655	9.83	118.3	.0060	53.20	306.30	146.87	342.5	153.80	191.03	240.6	4.60	1.833	.794	191.0	1.10	66.3
7310	21.25	109.4	.0089	53.20	227.10	217.79	342.5	224.72	261.95	227.4	5.96	1.833	.794	180.6	2.05	86.0
10965	31.10	112.0	.0080	53.20	136.10	195.78	342.5	202.71	239.94	228.3	5.48	1.833	.794	181.3	3.07	79.1
14620	39.20	117.6	.0074	53.20	94.40	178.90	342.5	185.83	223.06	236.5	5.28	1.833	.794	187.8	4.27	76.2

d)By inspection, the minimum value of P/h_t is selected for each combination of $(\rho Vg)_h$ and $(\rho Vg)_c$. Columns 27 to 31 need be calculated only for these minimum values.

e)Figure 7 shows $h_t A_t$ as a function of M_c for counterflow.

TABLE III

OUTLINE OF CALCULATIONS FOR THE DESIGN OF A TUBE-TYPE CROSS-FLOW INTERCOOLER

(L = 2 ft , D = 1/48 ft. Subscript 2 refers to charge-air side.)

1 n (a)	2 s (tube diam.)	3 m (a)	4 mn	5 $A_2 = nsL$ (sq. ft.)	6 $A_1 = \frac{mn \pi D^2}{4}$ (sq. ft.)	7 $(\rho V)_h = \frac{M_h}{g A_2}$ $\left(\frac{\text{slugs}}{\text{sq.ft.x sec.}}\right)$	8 $R_h = \frac{(\rho V)_h D}{\mu}$	9 $h_h D/k$ (b)	10 $1/h_h$	11 $(\rho V)_c$ (c) $\left(\frac{\text{slugs}}{\text{sq.ft.x sec.}}\right)$	12 f_2 (d)	13 $q_h = \frac{(\rho V)_h^2}{2\rho_H}$ $\left(\frac{\text{lb.}}{\text{sq.ft}}\right)$	14 $\Delta p_h = 4 m f_2 q_h$ $\left(\frac{\text{lb.}}{\text{sq.ft}}\right)$	15 $\frac{P_h}{550} = \frac{M_h \Delta p_h}{g \eta_p \rho_h 550}$ (hp.)	16 R_c	17 f_1 (e)	18 $\left(\frac{\Delta p}{q}\right)_c = \frac{4 f_1 L}{D}$	19 q_c $\left(\frac{\text{lb.}}{\text{sq.ft}}\right)$	20 Δp_c $\left(\frac{\text{lb.}}{\text{sq.ft}}\right)$	21 $\frac{P_c}{550} = \frac{(\rho V)_c A_2 \Delta p_c}{550 \eta_p \rho_c}$ (hp.)	22 Weight of tubes $= \frac{0.925 A_2 L}{D}$ (f) (lb.)	23 Weight of casing $= 2.89[L(s+D)(m+n)+mn(s+D)^2]$ (g) (lb.)	24 $W = (22) + (23)$ (lb.)	25 $\frac{P_W}{550} = \frac{W V_0 \epsilon (C_D/C_L)}{550}$ (hp.)	26 $\frac{P_T}{550} = \frac{P_h+P_c+P_W}{550}$ (hp.)
20	$\frac{D}{4}$	20	400	0.208	.137	0.2735	1.285×10^4	95.0	52.5	1.115	.1745	19.03	265.7	18.65	6.534×10^4	.0048	1.815	685.0	1243.28	507.08	12.15	6.80	18.95	1.14	526.87
		40	800		.274					.375			531.3	37.30	2.198	.0059	2.300	77.5	178.20	48.89	24.30	10.58	34.88	2.09	88.28
		60	1200		.410					.215			797.0	55.95	1.260	.0067	2.590	25.5	65.97	15.53	36.37	14.37	50.74	3.04	74.52
		100	2000		.684					.115			1328.0	93.23	.674	.0076	2.950	7.3	21.50	4.52	60.67	21.95	82.62	4.96	102.71
20	$\frac{D}{2}$	30	600	.416	.205	.1369	.643	61.8	80.2	.780	.1248	4.76	71.3	5.01	4.571	.0051	1.960	335.2	656.99	280.49	18.18	10.71	28.89	1.73	287.23
		50	1000		.342					.313			118.8	8.34	1.834	.0061	2.390	54.0	129.01	36.87	30.34	15.44	45.78	2.75	47.96
		80	1600		.547					.161			190.1	13.35	.943	.0071	2.755	14.3	39.34	9.25	48.52	22.54	71.06	4.26	26.86
		100	2000		.684					.121			237.6	16.68	.709	.0075	2.925	8.1	23.60	5.22	60.67	27.28	87.95	5.28	27.18
20	3/4 D	30	600	.624	.205	.0912	.429	48.0	104.0	1.085	.1145	2.12	29.1	2.04	6.358	.0048	1.825	648.7	1183.88	703.08	18.18	12.82	31.00	1.86	706.98
		60	1200		.410					.268			58.1	4.08	1.570	.0062	2.475	39.6	97.96	28.74	36.37	21.43	57.80	3.47	36.29
		80	1600		.547					.179			77.5	5.44	1,049	.0070	2.700	17.6	47.65	12.46	48.52	27.46	75.98	4.56	22.46
		100	2000		.684					.128			96.9	6.80	.750	.0074	2.895	9.0	26.14	6.11	60.67	32.92	93.59	5.62	18.53
		120	2400		.816					.104			116.2	7.84	.609	.0078	3.000	6.0	17.88	4.05	72.38	38.66	111.04	6.66	18.55
30	$\frac{D}{4}$	20	600	.312	.205	.1824	.857	73.0	67.5	.675	.1880	8.47	127.4	8.94	3.956	.0053	2.020	251.0	507.02	150.38	18.18	8.69	26.87	1.61	160.93
		30	900		.308					.341			191.1	13.42	1.998	.0061	2.350	64.1	150.56	42.15	27.28	10.78	38.06	2.28	57.85
		50	1500		.513					.170			318.5	22.36	.996	.0070	2.725	15.9	43.38	10.10	45.51	14.96	60.47	3.63	36.09
		60	1800		.616					.138			382.1	26.82	.809	.0073	2.850	10.5	29.90	6.79	54.64	17.05	71.69	4.30	37.91
30	$\frac{D}{2}$	20	600	.624	.205	.0912	.429	48.0	104.0	1.085	.1290	2.12	21.8	1.53	6.358	.0048	1.825	648.7	1183.88	703.08	18.18	10.71	28.89	1.73	706.34
		30	900		.308					.435			32.7	2.30	2.548	.0058	2.225	104.3	232:07	82.88	27.28	13.38	40.64	2.44	87.62
		40	1200		.410					.268			43.6	3.06	1.570	.0062	2.475	39.6	97.96	28.74	36.37	16.01	52.38	3.14	34.94
		60	1800		.616					.148			65.5	4.60	.867	.0072	2.895	9.0	26.14	6.36	54.64	19.50	24.14	4.45	15.41
		80	2400		.816					.104			87.3	6.13	.609	.0078	3.000	6.0	17.88	4.05	72.38	26.60	98.98	5.94	16.12
30	3/4 D	30	900	.936	.308	.0608	.286	37.5	134.0	.550	.1165	.94	13.2	.92	3.223	.0055	2.115	166.7	352.57	159.20	27.28	16.08	43.36	2.60	162.72
		40	1200		.410					.301			17.6	1.23	1.764	.0062	2.402	49.9	119.91	39.51	36.37	19.33	55.70	3.34	44.08
		50	1500		.513					.210			22.0	1.54	1.231	.0067	2.600	24.3	63.18	18.17	45.51	22.58	68.09	4.09	23.80
		60	1800		.616					.159			26.3	1.85	.938	.0071	2.770	13.9	38.59	10.09	54.64	25.84	80.48	4.83	16.77
		80	2400		.816					.110			35.1	2.47	.645	.0077	2.975	6.6	19.66	4.69	72.38	32.34	104.72	6.28	13.44
50	$\frac{D}{4}$	30	1500	.520	.510	.1094	.514	53.7	92.8	.182	.2050	3.04	74.8	5.25	1.067	.0069	1.625	18.4	29.82	7.41	45.24	18.72	63.96	3.84	16.50
		40	2000		.680					.124			99.7	7.00	.727	.0075	1.765	8.5	14.95	3.37	60.32	21.20	81.52	4.89	15.26
		48	2400		.816					.102			119.7	8.40	.598	.0078	1.850	5.7	10.60	2.36	72.38	23.18	95.56	5.23	16.49
50	$\frac{D}{2}$	30	1500	1.040	.510	.0547	.257	35.0	142.5	.215	.1825	.76	16.6	1.17	1.260	.0067	1.560	25.5	39.73	11.63	45.24	23.16	68.40	4.10	16.90
		40	2000		.680					.138			22.2	1.56	.809	.0073	1.725	10.5	18.10	4.54	60.32	26.38	86.70	5.20	11.30
		48	2400		.816					.110			26.6	1.87	.645	.0077	1.820	6.7	12.14	2.91	72.38	28.94	101.32	6.08	10.86
50	3/4 D	30	1500	1.560	.510	.0365	.172	27.5	182.5	.250	.1180	.34	4.8	.34	1.465	.0065	1.500	34.4	51.66	17.59	45.24	27.84	73.08	4.39	22.32
		40	2000		.680					.152			6.4	.45	.891	.0072	1.685	12.8	21.58	5.98	60.32	31.86	92.18	5.53	11.96
		48	2400		.816					.118			7.7	.54	.691	.0076	1.740	7.7	13.35	3.43	72.38	35.08	107.46	6.41	10.38

[a]The values of n and m used in the calculations are shown in figs. 12 and 13.

[b]Fig. 3 shows $h_h D/k$ as a function of R_h.

[c]The derivation of fig. 11 is explained in the text. The value of $(\rho V)_c$ is selected for which $\frac{1}{h_t} - \frac{1}{h_c} = \frac{1}{h_h}$, using the proper value of mn (column 4) and $\frac{1}{h_h}$ (column 10).

[d]Fig. 3 shows f_2 as a function of R_h and s.

[e]Fig. 4 shows f_1 as a function of R_c and L/D.

[f]Using copper tubes, wall thickness 0.005 in., density of copper is 555 lb. per cu. ft.

[g]Casing of copper, 1/32 in. thickness, assumed to encase the intercooler completely.

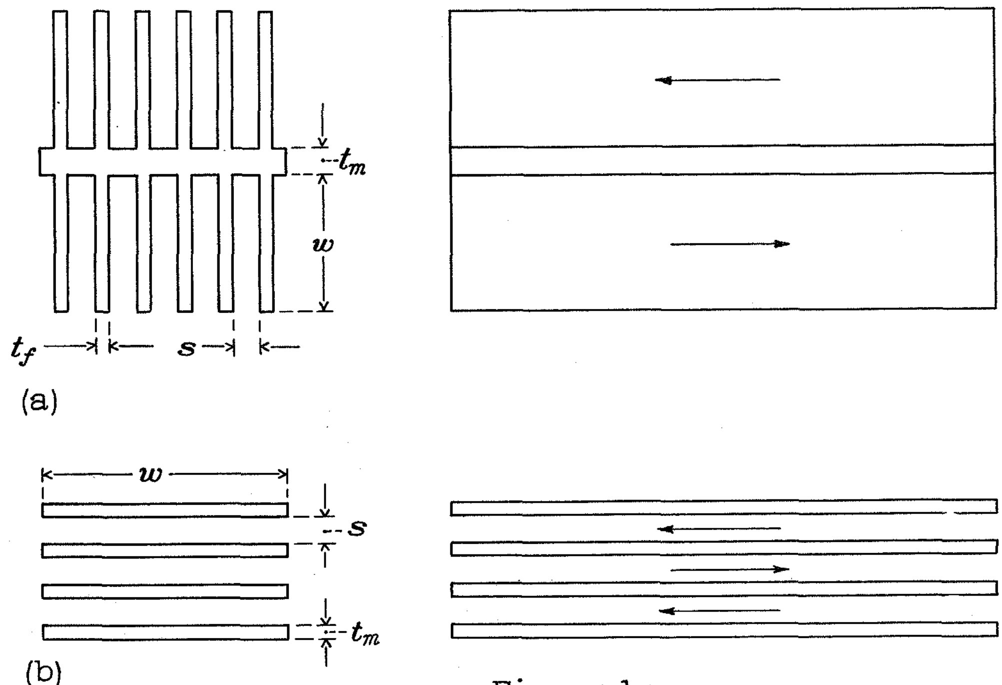


Figure 1.

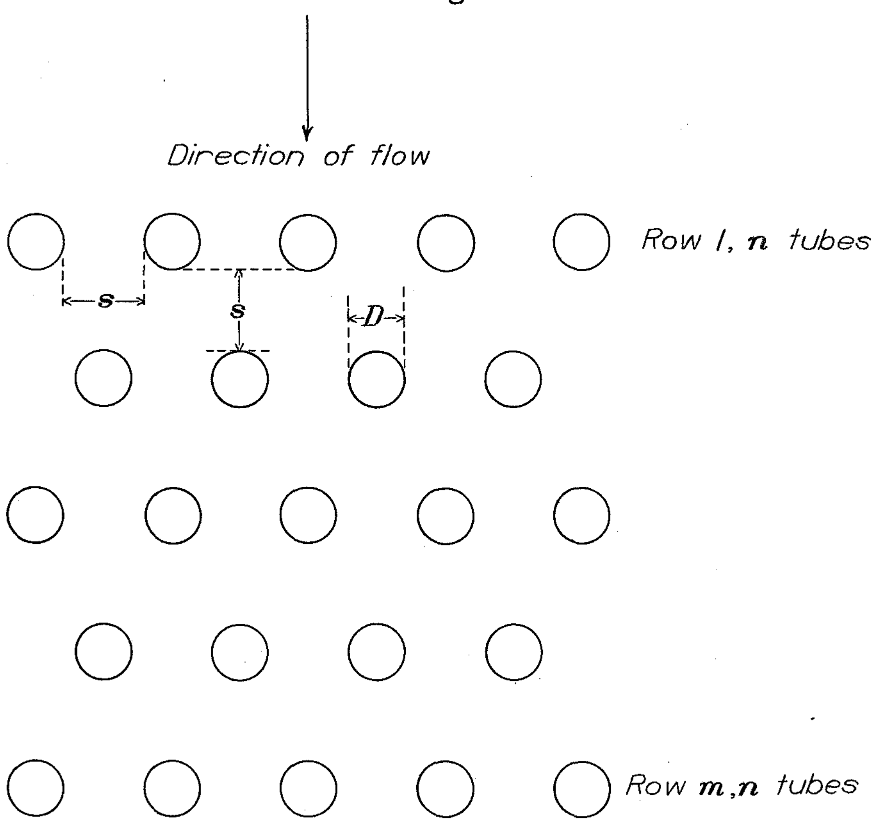


Figure 2.

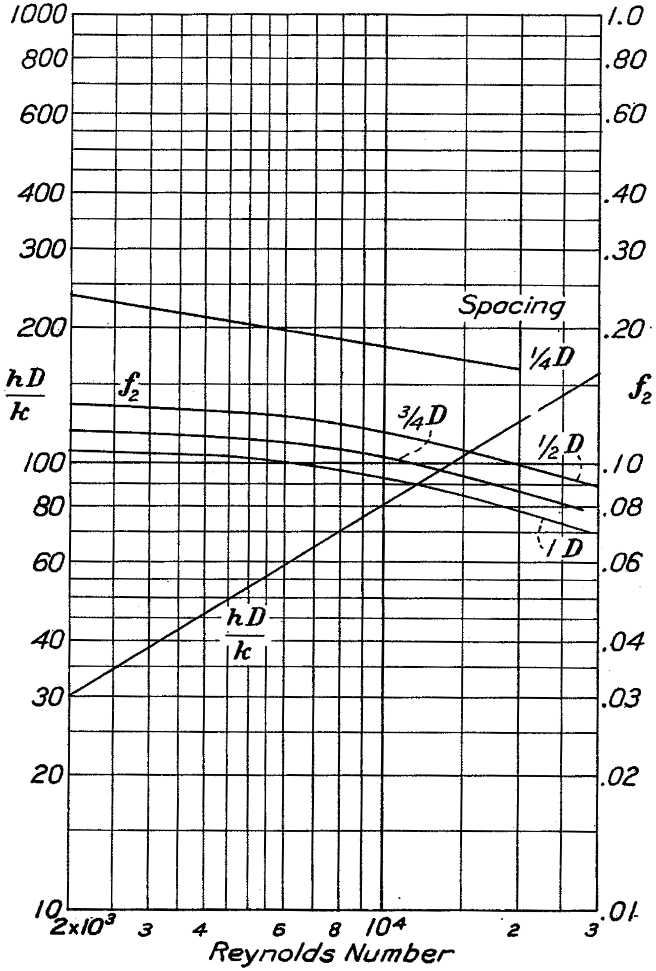


Figure 3.

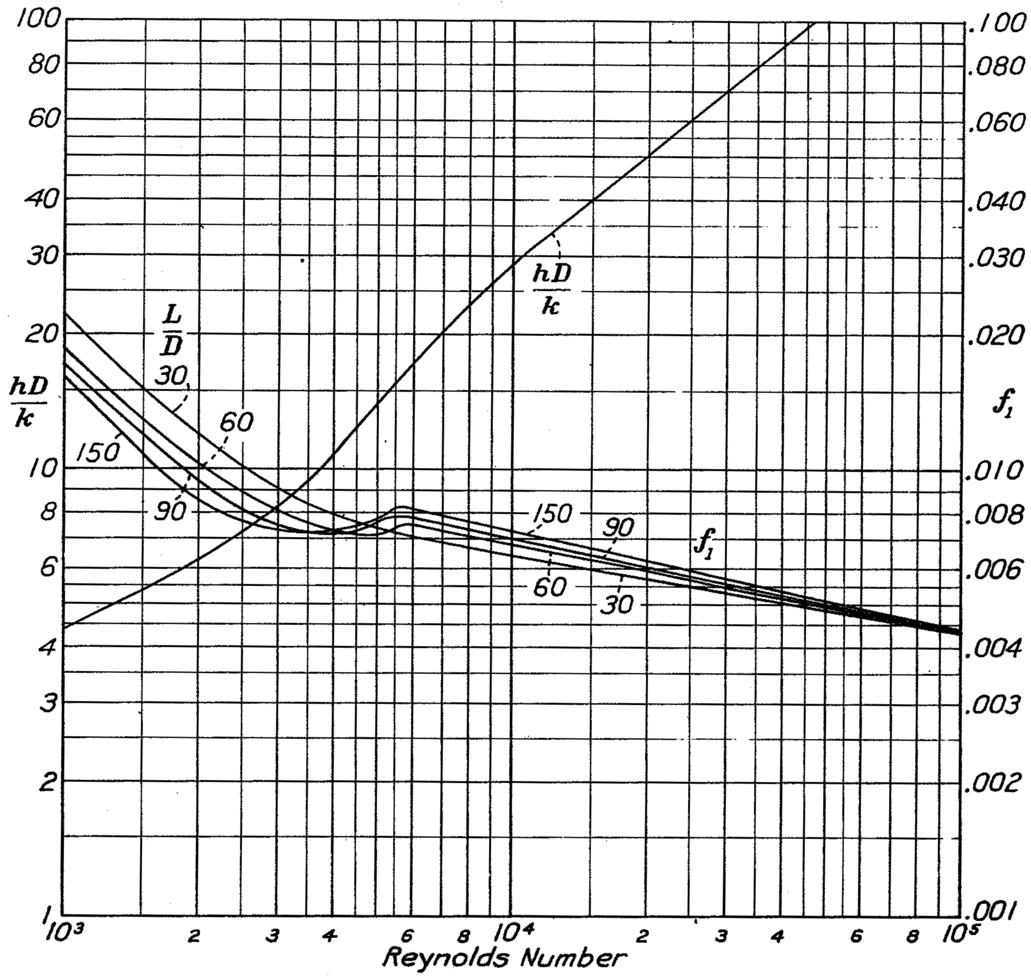
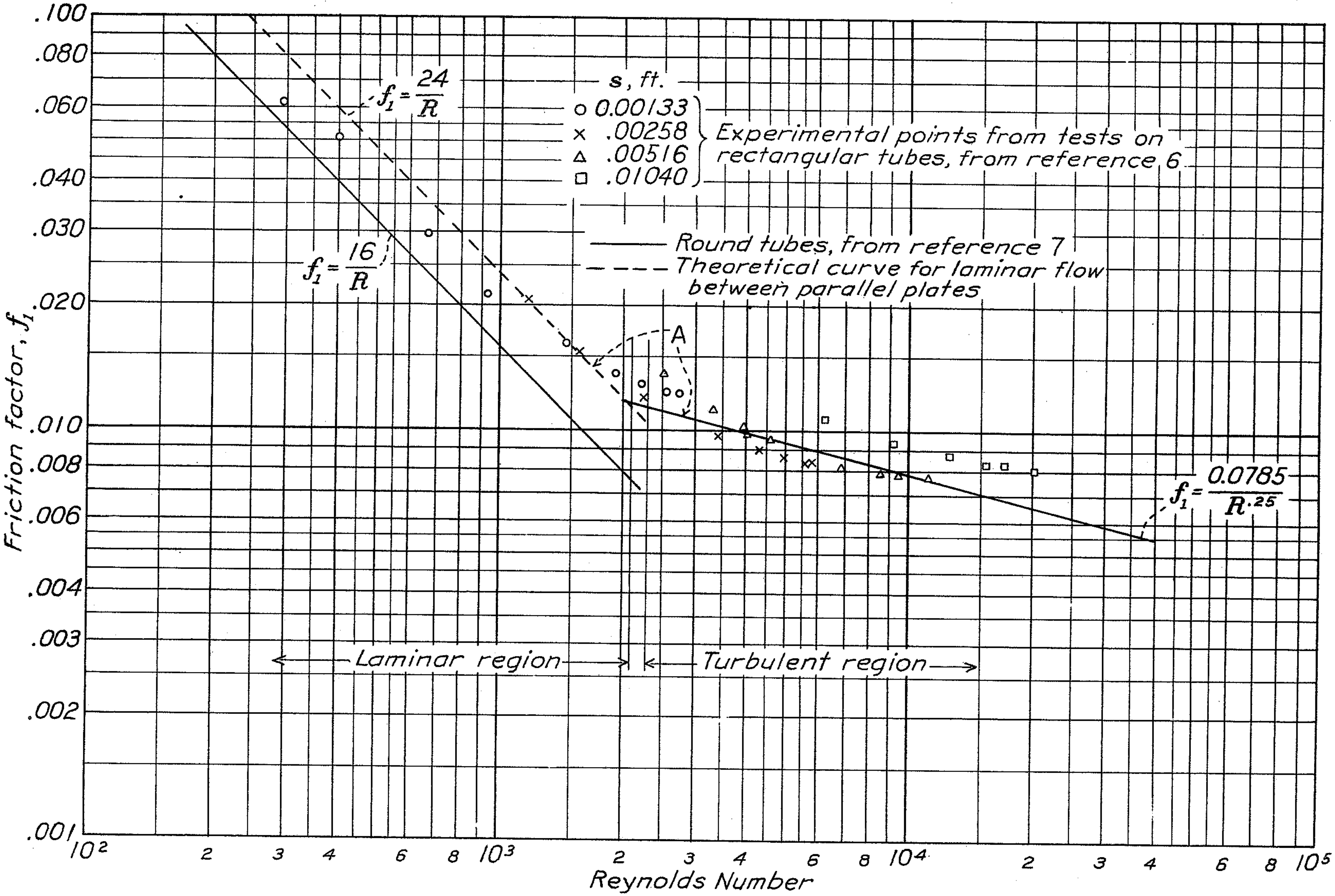


Figure 4.



s, ft.
o 0.00133
× .00258
△ .00516
□ .01040
Experimental points from tests on rectangular tubes, from reference 6
Round tubes, from reference 7
Theoretical curve for laminar flow between parallel plates
$f_1 = \frac{24}{R}$
$f_1 = \frac{16}{R}$
$f_1 = \frac{0.0785}{R^{.25}}$
A
Laminar region
Turbulent region
Friction factor, f_1.
Reynolds Number
.100
.080
.060
.040
.030
.020
.010
.008
.006
.004
.003
.002
.001
10^2
10^3
10^4
10^5

Figure 5

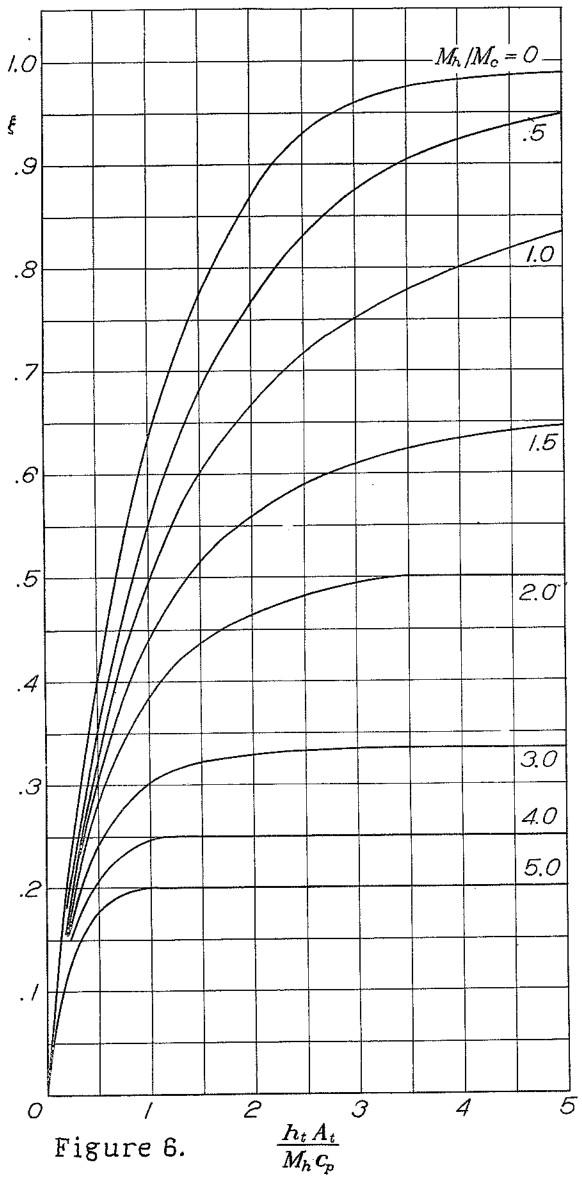


Figure 6.

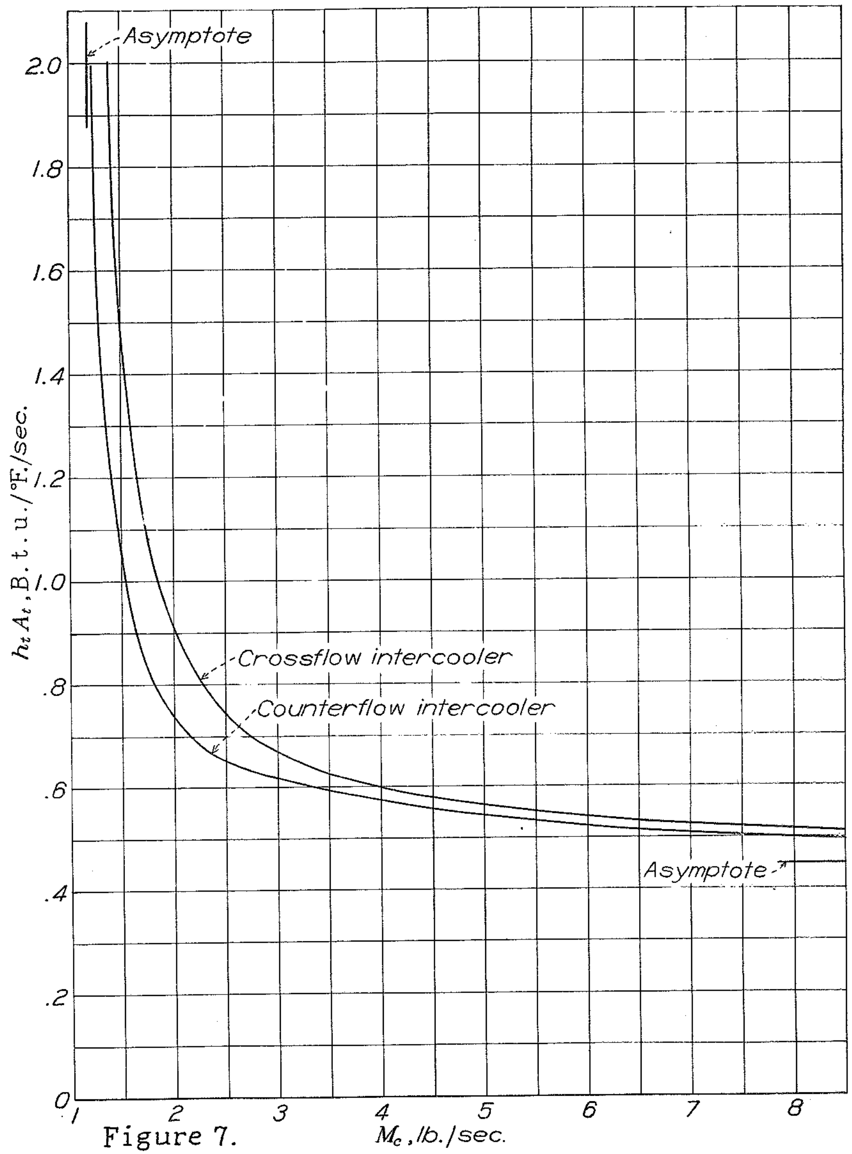


Figure 7.

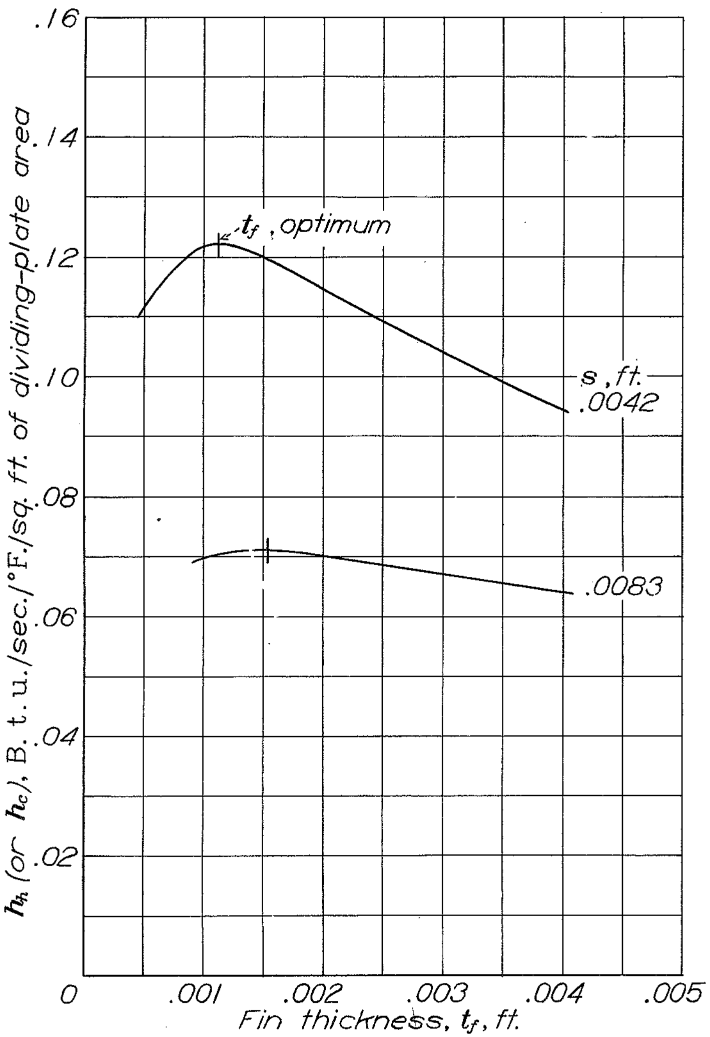


Figure 8.

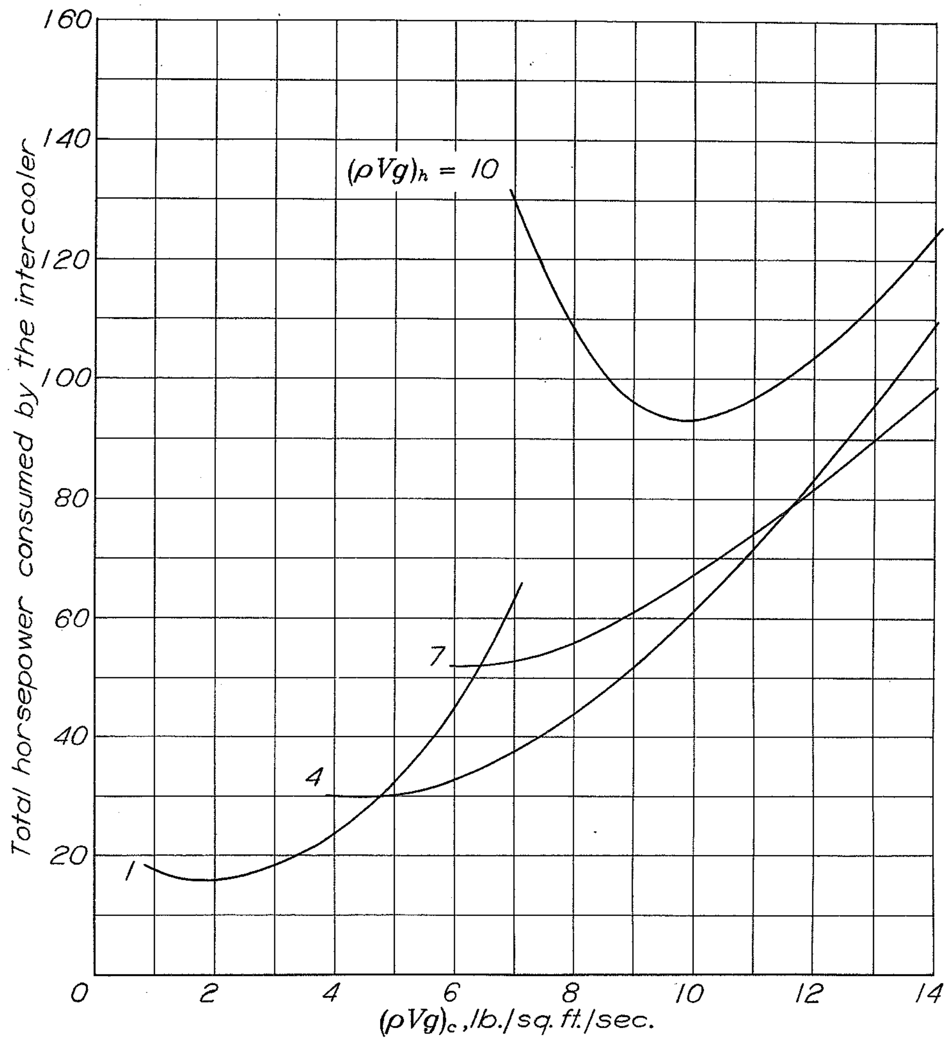


Figure 9.

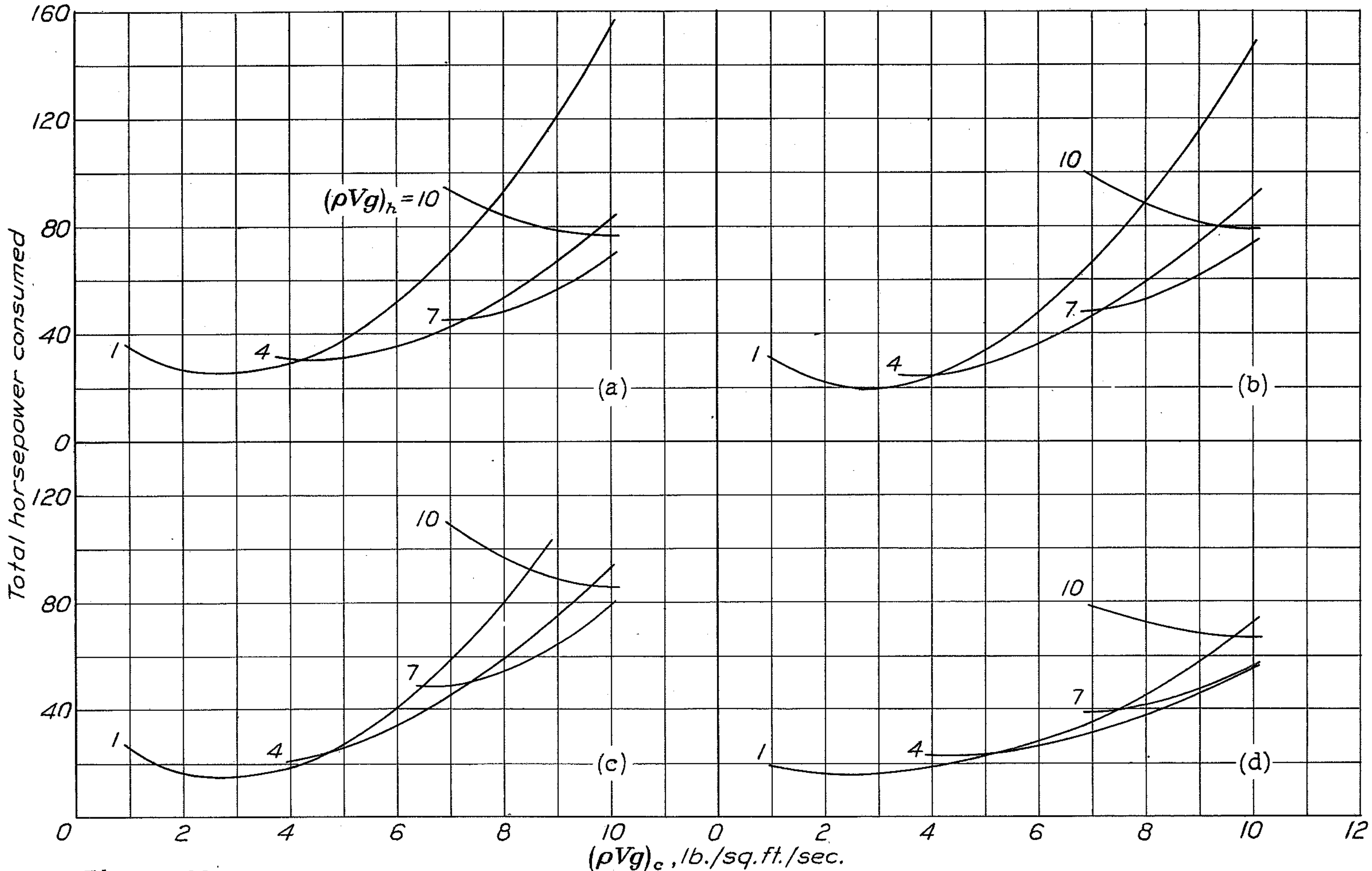

160
120
80
40
0
120
80
40
0
Total horsepower consumed
$(\rho Vg)_h = 10$
1
4
7
10
(a)
(b)
(c)
(d)
0 2 4 6 8 10
0 2 4 6 8 10 12
$(\rho Vg)_c$, lb./sq.ft./sec.

Figure 10.

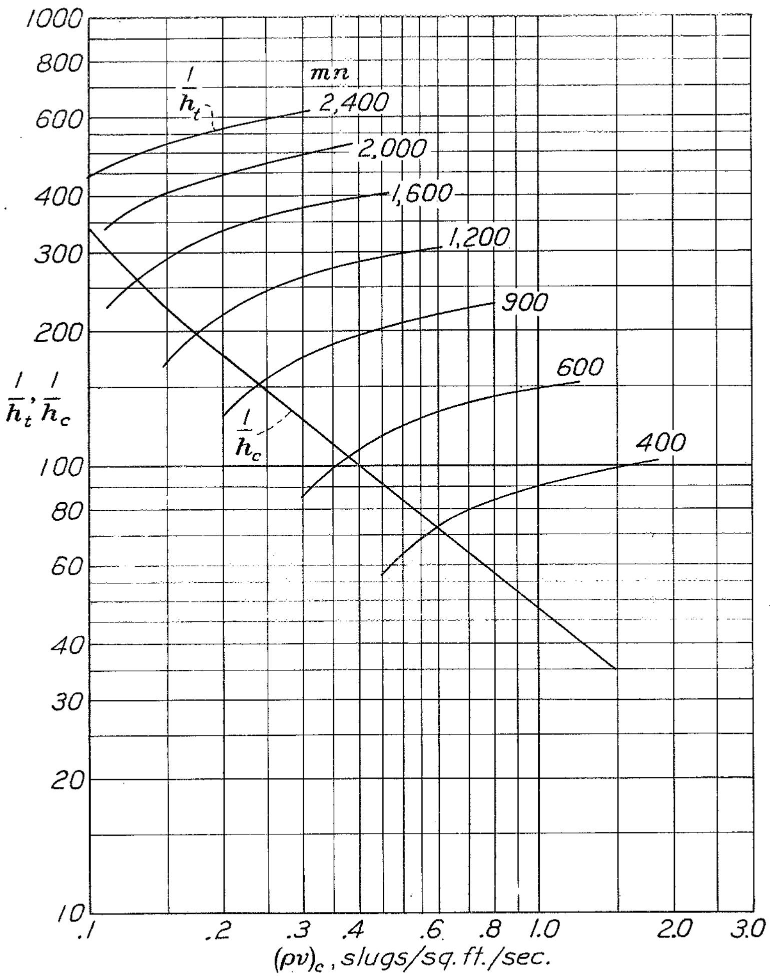


Figure 11.

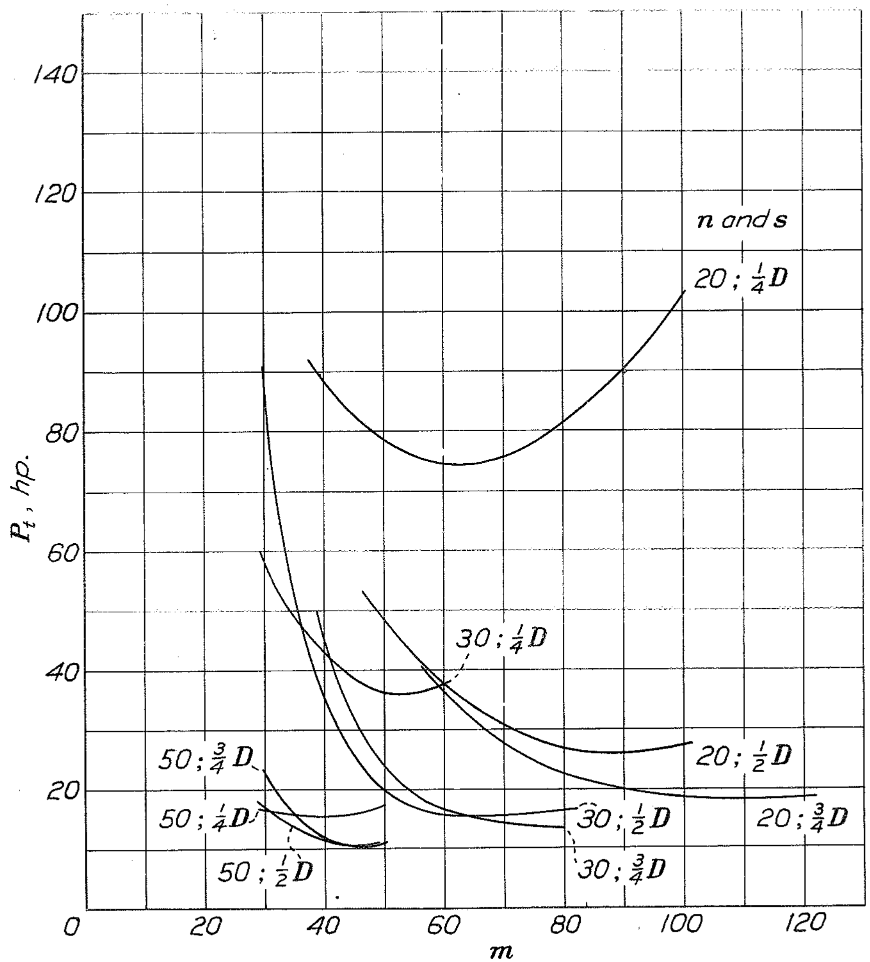


Figure 12.

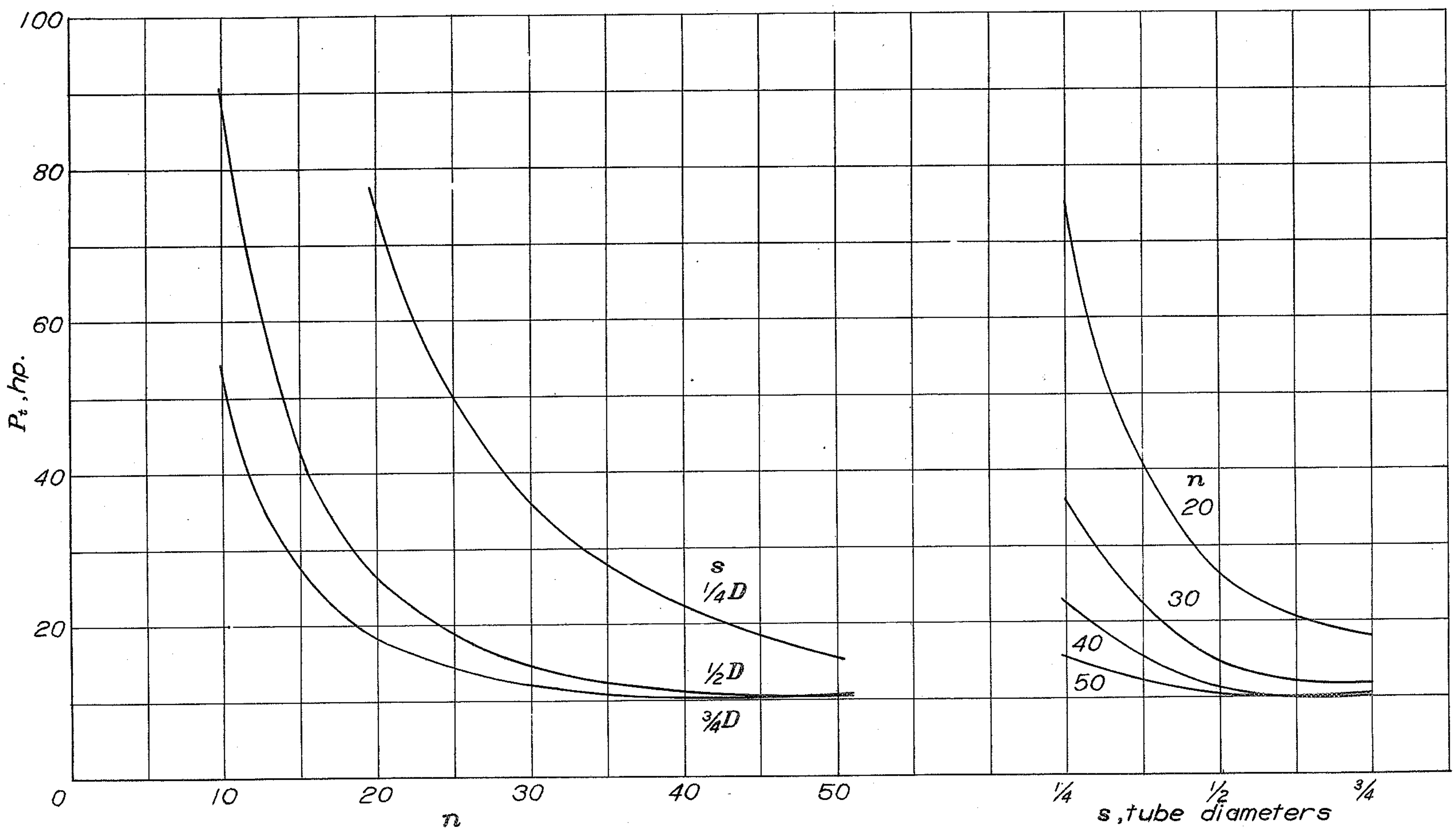

100
80
60
40
20
0
P_t, hp.
10
20
30
40
50
n
s
1/4D
1/2D
3/4D
n
20
30
40
50
1/4
1/2
3/4
s, tube diameters

Figure 13.