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THE CROSS-FLOW PLATE-TYPE INTERCOOLER

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ADVANCE CONFIDENTIAL REPORT

THE CROSS-FLOW PLATE-TYPE INTERCOOLER

By Benjamin Pinkel, J. George Reuter, and
Michael F. Valerino

SUMMARY

A mathematical analysis of plate-type intercooler design has been made for both turbulent and laminar flow. Charts are presented that show how the plate-type intercooler volume, weight, plate area, frontal area, and linear dimensions can be varied without changing the intercooler operating conditions. The relations between the intercooler operating conditions and the intercooler dimensional characteristics and weight are also given.

The charts show that a large reduction in intercooler volume can be achieved with no change in intercooler operating conditions by reduction of the spacings between the plates and reduction of the dimensions in the directions of the charge-air and cooling-air flows. The dimension at right angles to the flow directions, however, is increased.

A plate-type intercooler unit having plates spaced 0.025 inch was tested to check the validity of the heat-transfer and pressure-drop theories for air flow between very closely spaced plates. The theoretical and experimental results were in close agreement. For the same operating conditions this intercooler has approximately one-third the volume of the smallest commercial intercooler for which data could be found.

INTRODUCTION

The current increases in engine power and critical altitude are accompanied by greater demands for charge-air cooling. These demands can be met either by increasing the intercooler volume or by improving the cooling accomplished per unit intercooler volume. Inasmuch as the volume and shape of an intercooler is limited by the available space, the designer, in order to meet a specified set of operating conditions and installation requirements, may find it necessary to consider revisions of the internal structure of the intercooler.

A large number of plate-type intercoolers can be designed for a given performance, such as given weight flows, pressure drops, densities, and inlet and outlet temperatures of the charge and cooling air. These intercoolers will differ in size, shape, and weight, and it is the purpose of this paper to show the relations between these factors and the plate spacings. These relations indicate how intercooler size may be either reduced or adjusted to the available space.

A plate-type intercooler having spaces of 0.025 inch between adjacent plates and having a length of 2.30 inches in the direction of cooling-air flow and 4.30 inches in the direction of charge-air flow was tested, and the results are presented herein. In order to reduce the plate deflections in actual intercooler operation due to the pressure difference between the charge air and the cooling air across each plate, the plates were curved in the direction of cooling-air flow with a radius of 3.5 inches. This intercooler test unit does not represent an optimum design; it was built and tested merely to check the theoretical heat-transfer and pressure-drop equations presented in this report.

This work was done by the NACA at the Langley Memorial Aeronautical Laboratory at Langley Field, Va., in 1941.

SYMBOLS

M weight rate of air flow, pounds per second

l length of flow passage measured in direction of flow, feet (or inches where designated)

s plate spacing (perpendicular distance between adjacent faces of adjacent plates), feet (or inches where designated)

d equivalent diameter of flow passages (four times cross section divided by wetted perimeter), feet

r ratio of equivalent diameter of passages formed by parallel flat plates of infinite width to equivalent diameter of flow passages ($2s/d$)

t plate thickness, feet (or inches where designated)

ρ_m density of plate material, pounds per cubic foot

w intercooler width measured in no-flow direction, feet (or inches where designated)

N one-half total number of intercooler plates

v intercooler volume, cubic feet (or cubic inches where designated)

A_f core face area at right angles to cooling-air flow (frontal area), square feet (or square inches where designated)

f ratio of total cross-sectional-flow area to area of core face

W intercooler weight, pounds

n number of separators between adjacent plates

E_f separator fin effectiveness

l_s length of flow passage in contact with separators of adjacent passage (measured in direction of flow), feet

A_1 cooling-air effective heat-transfer area, square feet

$$\left\{ 2Nl_1 \left[l_2 - l_{s_2} + (n_1 - 1)s_1E_f \right] \right\}$$

A_2 charge-air effective heat-transfer area, square feet

$$\left\{ 2Nl_2 \left[l_1 - l_{s_1} + (n_2 - 1)s_2E_f \right] \right\}$$

h surface heat-transfer coefficient, Btu per second per square foot per $^{\circ}F$

A_r over-all effective heat-transfer area, square feet

U over-all heat-transfer coefficient based on over-all effective heat-transfer area, Btu per second per square foot per $^{\circ}F$

S total plate area, square feet (or square inches where designated) $(2Nl_1l_2)$

V velocity of air flow, feet per second

R Reynolds number of air flow based on equivalent diameter

Δp_f skin-friction pressure drop of air in intercooler, inches of water

Δp_e pressure drop corresponding to entrance-exit losses including vena-contracta loss, inches of water

Δp_V pressure drop of air due to change in velocity distribution in intercooler channels, inches of water

Δp_H pressure change of air due to the momentum change caused by heat exchange in intercooler, inches of water

Δp total pressure drop of air across intercooler, inches of water

P power required to force air through intercooler, horsepower

ρ air density, pounds per cubic foot

ρ_o standard atmospheric density (0.0765 lb/cu ft)

σ density of air relative to standard atmosphere (ρ/ρ_o)

c_p specific heat of air at constant pressure (0.24 Btu per pound per $^{\circ}F$)

k thermal conductivity of air, Btu per second per square foot per $^{\circ}F$ gradient per foot

μ absolute viscosity of air, pounds per second per foot

β ratio of change in temperature of air in passing through intercooler to absolute temperature of air at intercooler entrance

η cooling effectiveness, ratio of temperature drop of charge air to temperature difference between charge air and cooling air at entrance

Subscripts:

1 cooling air

2 charge air

av average condition in intercooler

o reference intercooler (except in ρ_o)

Symbols with subscript o are primed to indicate laminar flow.

Throughout this paper the term "operating conditions" refers to a set of values that include η, M_1, M_2, $\sigma_{1_{av}} \Delta p_{f_1}$, and $\sigma_{2_{av}} \Delta p_{f_2}$.

The following parameters are used in this paper:

$$K_1 = A_1/S$$

$$K_2 = A_2/S$$

$$J_1 = \frac{l_2 - l_{s_2}}{l_2}$$

$$J_2 = \frac{l_1 - l_{s_1}}{l_1}$$

$$L_1 = J_1/K_1$$

$$L_2 = J_2/K_2$$

$$\theta = \frac{M_1}{M_2} \frac{\sigma_{1_{av}} \Delta p_{f_1}}{\sigma_{2_{av}} \Delta p_{f_2}}$$

ANALYSIS

Heat-Transfer and Pressure-Drop Equations

It is convenient to express the cooling provided by an intercooler in terms of a factor η, called the cooling effectiveness. The cooling effectiveness is defined as the ratio of the drop in charge-air temperature to the difference between the charge-air and cooling-air temperatures at the intercooler entrances.

In reference 1 it is shown that the cooling effectiveness may be written as a function of M_1/M_2, and UA_r/M_2c_p, that is,

$$\eta = \Phi\left(\frac{M_1}{M_2}, \frac{UA_r}{M_2c_p}\right) \tag{1}$$

Curves showing this relationship for cross-flow heat exchangers were obtained from Nusselt's analysis (reference 2) and are shown in figure 1.

The total pressure drop across an intercooler may be considered as the sum of the pressure changes arising from the following sources:

1. Surface-friction loss

2. Entrance-exit loss

3. Velocity-profile loss

4. Momentum change accompanying heat exchange

The surface-friction pressure-drop relationships for both turbulent and laminar flow are given in reference 3. These relationships may be expressed in the notation of this paper as follows:

For turbulent flow,

$$\sigma_{av}\,\Delta p_f = \left(\frac{0.0854}{10.4gp_o}\right) r^{\frac{6}{5}}\ (\rho V)^2 \left(\frac{\rho Vs}{\mu}\right)^{-\frac{1}{5}} \frac{l}{s} \tag{2}$$

For laminar flow,

$$\sigma_{av} \Delta p_f = \frac{24\mu}{10.4gp_o} \rho V r \frac{l}{s^2} \tag{3}$$

The entrance-exit loss includes (1) the conversion of static pressure into dynamic pressure at the entrance according to Bernoulli's equation, (2) the vena-contracta loss occurring immediately after the air enters the intercooler passages, and (3) the recovery in static pressure at the exit due to momentum change. The velocity-profile loss is caused by the change in velocity distribution of the air as it flows through the intercooler passages. The heating loss or gain is caused by the momentum change of the air flowing through the intercooler passages as a result of the density changes accompanying the transfer of heat. This heating loss or gain should not be confused with the effect of heating on the surface-friction loss, as this effect is accounted for by assigning the proper value of σ_{av} in the surface-friction equations. For an intercooler with blunt entrance and exit sections, the entrance-exit loss, velocity-profile loss, and heating loss or gain may be expressed in terms of the surface-friction loss:

Entrance-exit loss.-

For turbulent flow,

$$\frac{\Delta p_e}{\Delta p_f} = 5.1\left(f^2 - 2.4\ f + 1.5\right) \frac{R^{0.2}}{\frac{rl}{2s}} \tag{4}$$

For laminar flow,

$$\frac{\Delta p_e}{\Delta p_f} = \frac{(f^2 - 2.8\ f + 1.5)}{96} \frac{R}{\frac{l}{2s}} \tag{5}$$

Velocity-profile loss.-

For turbulent flow,

$$\frac{\Delta p_V}{\Delta p_f} = 0.46 \frac{R^{0.2}}{\frac{rl}{2s}} \tag{6}$$

For laminar flow

$$\frac{\Delta p_V}{\Delta p_f} = \frac{1}{175} \frac{R}{\frac{l}{2s}} \tag{7}$$

Heating loss or gain.-

For turbulent flow,

$$\frac{\Delta p_H}{\Delta p_f} = 10.2\beta \frac{R^{0.2}}{\frac{rl}{2s}} \tag{8}$$

For laminar flow,

$$\frac{\Delta p_H}{\Delta p_f} = \frac{\beta}{48} \frac{R}{\frac{l}{2s}} \tag{9}$$

Equations (8) and (9) apply when the pressure drop through the intercooler is small compared with the absolute pressure. At extreme altitudes this condition may not hold and a correction should be applied for the effect of change in pressure on the velocity of the cooling air as it passes through the intercooler.

Of these losses, only the surface-friction loss is necessary for the transfer of heat, this transfer being accomplished most efficiently (that is, with minimum surface-friction loss) for flow through a smooth channel. By means of properly shaped intercooler entrance and exit sections, the entrance-exit loss can be avoided and the velocity-profile and heating losses reduced.

In the design charts and equations, only the surface-friction pressure drop is considered. This pressure drop is very nearly equal to the total pressure drop for intercoolers with streamlined entrance and exit sections; for intercoolers with blunt entrance

and exit sections, the surface-friction pressure drop is in most cases the largest part of the total pressure drop. For an intercooler with blunt entrance and exit sections, the entrance-exit loss, velocity-profile loss, and heating loss or gain can be determined from equations (4) to (9), or from table I, and can be added to the surface-friction loss to give the total pressure drop across the intercooler.

The heat-transfer relationships for turbulent flow are given in reference 3 and for laminar flow, in reference 4. These relationships may be expressed as follows:

For turbulent flow,

$$\frac{hs}{k} = 0.0172\ r^{\frac{1}{5}} \left(\frac{\rho V s}{\mu}\right)^{\frac{4}{5}} \tag{10}$$

For laminar flow,

$$\frac{hs}{k} = 3.8r \tag{11}$$

From equation (1) and the pressure-drop and heat-transfer equations, relations are derived in appendix A between the dimensions and weight of a plate-type intercooler and the operating conditions. For any given set of operating conditions, a different intercooler is obtained for each plate spacing. These intercoolers differ in size, shape, and weight. For a given set of operating conditions, the smaller the plate spacing the smaller are the intercooler volume and the dimensions in the flow directions and the larger is the dimension in the no-flow direction.

The following plan is used in the presentation of the relations derived in appendix A.

1. Equations and charts are given for the determination of the dimensions of a reference intercooler which is defined in this paper as one in which the plate spacings are equal to a reference spacing. The values of the reference plate spacings chosen in this paper are 0.25 inch for turbulent flow and 0.10 inch for laminar flow. For convenience, the reference intercooler is further characterized by the conditions that the plate thickness is zero and that the effect of the separator strips is neglected.

2. Equations and charts are then presented that show how the intercooler may be altered in size, shape, and weight as the plate spacings are reduced and the plate thickness is increased with respect to the values of plate spacings and plate thickness chosen for the reference intercooler.

3. Corrections for the effect of plate separator strips are given in table II.

Dimensional Characteristics and Weight of Reference Intercooler as Functions of Operation Conditions

The following equations are obtained from the equations in appendix A for the conditions of the reference intercooler; namely, $s_{1_o} = s_{2_o} = 0.25$ inch, $s_{1_o}' = s_{2_o}' = 0.10$ inch, $t = 0$, and $J = K = L = 1$. The physical properties k and μ of air are evaluated at 100° F. These equations together with figure 1 give the dimensional characteristics and the weight of the reference intercooler for any given set of operating conditions. Only the cases in which the flows in each passage are either both turbulent or both laminar are considered.

For turbulent flow,

$$\frac{S_o}{M_2} = \frac{W_o}{\rho_m t\, M_2} = 150.6\, s_{1_o}^{\frac{1}{5}} \left(\frac{UA_r}{M_2 c_p}\right)^{\frac{7}{5}} \frac{\left(\theta^{-\frac{2}{7}} + 1\right)^{\frac{7}{5}}}{\left(\sigma_{2_{av}}\, \Delta p_{f_2}\right)^{\frac{2}{5}}} \tag{12}$$

$$v_o = s_{1_o} S_o \tag{13}$$

$$\frac{l_{2_o}}{s_{2_o}} = 10.98\, s_{1_o}^{\frac{1}{14}} \left(\sigma_{2_{av}}\, \Delta p_{f_2}\right)^{\frac{5}{14}} \left(\frac{S_o}{M_2}\right)^{\frac{9}{14}} \tag{14}$$

$$\frac{l_{1_o}/s_{1_o}}{l_{2_o}/s_{2_o}} = \frac{M_2}{M_1}\, \theta^{\frac{5}{14}} \tag{15}$$

$$w_o = \frac{S_o}{\frac{l_{1_o}}{s_{1_o}} \frac{l_{2_o}}{s_{2_o}}} \left(\frac{1}{s_{1_o}}\right) \tag{16}$$

$$A_{f_o} = \frac{S_o}{\frac{l_{1_o}}{s_{1_o}}} \tag{17}$$

For laminar flow,

$$\frac{S_o'}{M_2} = \frac{W_o'}{\rho_m \, t \, M_2} = 31{,}800 \; s_{1_o}' \left(\frac{UA_r}{M_2 c_p}\right) \tag{18}$$

$$v_o' = s_{1_o}' \, S_o' \tag{19}$$

$$\frac{l_{2_o}'}{s_{2_o}'} = 205.5 \left(s_{1_o}'\right)^{\frac{1}{2}} \left(\sigma_{2_{av}} \, \Delta p_{f_2}\right)^{\frac{1}{2}} \left(\frac{S_o'}{M_2}\right)^{\frac{1}{2}} \tag{20}$$

$$\frac{l_{1_o}'/s_{1_o}'}{l_{2_o}'/s_{2_o}'} = \frac{M_2}{M_1} \, \theta^{\frac{1}{2}} \tag{21}$$

$$w_o' = \frac{S_o'}{\frac{l_{1_o}'}{s_{1_o}'} \frac{l_{2_o}'}{s_{2_o}'}} \left(\frac{1}{s_{1_o}'}\right) \tag{22}$$

$$A_{f_o}' = \frac{S_o'}{\frac{l_{1_o}'}{s_{1_o}'}} \tag{23}$$

Variation of Intercooler Dimensional Characteristics and Weight for Constant Operating Conditions

The variation in the dimensional characteristics and the weight of a plate-type intercooler as the plate spacings and the plate thickness vary from the reference values is given for any given set of operating conditions by the following equations:

For turbulent flow,

$$\frac{S}{S_o} = \frac{W}{W_o} = \left(\frac{s_1}{s_{1_o}}\right)^{\frac{1}{5}} \left[\frac{\left(\frac{s_2}{s_1}\right)^{\frac{1}{7}} + \theta^{-\frac{2}{7}}}{1 + \theta^{-\frac{2}{7}}}\right]^{\frac{7}{5}} \tag{24}$$

The factor $\left(\frac{s_2}{s_1}\right)^{\frac{1}{7}} + \theta^{-\frac{2}{7}}$ in equation (24) can be shown to be equal to $\left[\frac{1 + \left(\frac{s_2}{s_1}\right)^{\frac{1}{7}}}{2}\right]\left(1 + \theta^{-\frac{2}{7}}\right)$ within 1 to 2 percent for values of θ at which intercoolers operate. Thus equation (24) reduces to

$$\frac{S}{S_o} = \frac{W}{W_o} = \left(\frac{s_1}{s_{1_o}}\right)^{\frac{1}{5}} \left[\frac{1 + \left(\frac{s_2}{s_1}\right)^{\frac{1}{7}}}{2}\right]^{\frac{7}{5}} \tag{25}$$

Also,

$$\frac{v}{v_o} = \left(\frac{S}{S_o}\right)\left(\frac{s_1}{s_{1_o}}\right)\left(\frac{1}{2} + \frac{s_2}{2s_1} + \frac{t}{s_1}\right) \tag{26}$$

$$\frac{w}{w_o} = \left(\frac{S}{S_o}\right)^{-\frac{2}{7}} \left(\frac{s_1}{s_{1_o}}\right)^{-\frac{3}{7}} \left(\frac{s_2}{s_1}\right)^{-\frac{15}{14}} \left(\frac{1}{2} + \frac{s_2}{2s_1} + \frac{t}{s_1}\right) \tag{27}$$

$$\frac{l_1/s_1}{l_{1_o}/s_{1_o}} = \left(\frac{S}{S_o}\right)^{\frac{9}{14}} \left(\frac{s_1}{s_{1_o}}\right)^{\frac{1}{14}} \tag{28}$$

$$\frac{l_2/s_2}{l_{2_o}/s_{2_o}} = \left(\frac{S}{S_o}\right)^{\frac{9}{14}} \left(\frac{s_1}{s_{1_o}}\right)^{\frac{1}{14}} \left(\frac{s_2}{s_1}\right)^{\frac{1}{14}} \tag{29}$$

$$\frac{A_f}{A_{f_o}} = \left(\frac{S}{S_o}\right)^{\frac{5}{14}} \left(\frac{s_1}{s_{1_o}}\right)^{-\frac{1}{14}} \left(\frac{1}{2} + \frac{s_2}{2s_1} + \frac{t}{s_1}\right) \tag{30}$$

For laminar flow,

$$\frac{S}{S_o'} = \frac{W}{W_o'} = \left(\frac{s_1}{s_{1_o}'}\right) \left(\frac{1}{2} + \frac{s_2}{2s_1}\right) \tag{31}$$

$$\frac{v}{v_o'} = \left(\frac{s_1}{s_{1_o}'}\right)^2 \left(\frac{1}{2} + \frac{s_2}{2s_1}\right) \left(\frac{1}{2} + \frac{s_2}{2s_1} + \frac{t}{s_1}\right) \tag{32}$$

$$\frac{w}{w_o'} = \left(\frac{s_1}{s_{1_o}'}\right)^{-2} \left(\frac{s_2}{s_1}\right)^{-\frac{3}{2}} \left(\frac{1}{2} + \frac{s_2}{2s_1} + \frac{t}{s_1}\right) \tag{33}$$

$$\frac{l_1/s_1}{l_{1_o}'/s_{1_o}'} = \left(\frac{s_1}{s_{1_o}'}\right) \left(\frac{1}{2} + \frac{s_2}{2s_1}\right)^{\frac{1}{2}} \tag{34}$$

$$\frac{l_2/s_2}{l_{2_o}'/s_{2_o}'} = \left(\frac{s_1}{s_{1_o}'}\right)\left(\frac{s_2}{s_1}\right)^{\frac{1}{2}}\left(\frac{1}{2} + \frac{s_2}{2s_1}\right)^{\frac{1}{2}} \tag{35}$$

$$\frac{A_f}{A_{f_o}'} = \left(\frac{1}{2} + \frac{s_2}{2s_1}\right)^{\frac{1}{2}}\left(\frac{1}{2} + \frac{s_2}{2s_1} + \frac{t}{s_1}\right) \tag{36}$$

Corrections for Separator Strips

In table II factors are tabulated that, when multiplied by the weight and the dimensional characteristics of a plate-type intercooler as calculated from equations (12) to (36), give the intercooler weight and the dimensional characteristics corrected for the effect of plate-separator strips. These correction factors are given in terms of r, J, K, L, ψ_t, and ψ_L, where r, J, K, and L are defined under symbols and ψ_t and ψ_L are defined as follows:

$$\psi_t = r_2^{\frac{1}{5}} \frac{L_2}{K_2}^{\frac{2}{5}} \left[\frac{\left(\frac{s_2}{s_1}\right)^{\frac{1}{7}} + \theta^{-\frac{2}{7}} \left(\frac{r_2}{r_1}\right)^{-\frac{1}{7}} \left(\frac{K_2}{K_1}\right)^{\frac{5}{7}} \left(\frac{L_1}{L_2}\right)^{\frac{2}{7}}}{\left(\frac{s_2}{s_1}\right)^{\frac{1}{7}} + \theta^{-\frac{2}{7}}}\right]^{\frac{7}{5}} \tag{37}$$

When θ, r_2/r_1, L_1/L_2, and K_2/K_1, do not differ appreciably from unity, a close approximation for ψ_t is given by

$$\psi_t = \left[\frac{L_2^{\frac{2}{7}} r_2^{\frac{1}{7}}}{2K_2^{\frac{5}{7}}} + \frac{L_1^{\frac{2}{7}} r_1^{\frac{1}{7}}}{2K_1^{\frac{5}{7}}}\right]^{\frac{7}{5}} \tag{38}$$

Also,

$$\psi_L = \frac{\frac{s_2}{s_1} + \frac{r_2}{r_1}\frac{K_2}{K_1}}{r_2 K_2\left(1 + \frac{s_2}{s_1}\right)} \tag{39}$$

When r_2/r_1 and K_2/K_1 do not differ appreciably from unity, a close approximation for ψ_L is given by

$$\psi_L = \frac{1}{2}\left(\frac{1}{r_1K_1} + \frac{1}{r_2K_2}\right) \tag{40}$$

Intercooler Charts

Equations (25) to (30) for turbulent flow are plotted in figure 2 and equations (31) to (36), for laminar flow, are plotted in figure 3. The reference-plate spacings chosen for these plots are $s_{1_o} = s_{2_o} = 0.25$ inch for the case of turbulent flow and $s_{1_o}' = s_{2_o}' = 0.10$ inch for the case of laminar flow. These plots were calculated for $s_2/s_1 = 1.00$ and 0.50 and for $t = 0$, 0.01, and 0.02 inch. The curves in these figures give directly, for any given set of operating conditions, the change in intercooler dimensional characteristics and in weight when the plate spacings are changed from 0.25 inch for turbulent flow and 0.10 inch for laminar flow. The effect of variation in plate thickness may also be obtained from these figures.

Figure 4(a), plotted from equations (1) and (12) for $s_{1_o} = 0.25$ inch, and figure 4(b), plotted from equations (1) and (18) for $s_{1_o}' = 0.10$ inch, give the relation between reference-intercooler plate area and operating conditions. Figure 5(a), plotted from equation (14) for $s_{1_o} = 0.25$ inch, and figure 5(b), plotted from equation (20) for $s_{1_o}' = 0.10$ inch, present for a reference intercooler the relation between charge-air length-spacing ratio and plate area for various values of charge-air pressure drop. From the values of plate area and charge-air length-spacing ratio obtained from figures 4 and 5 the other dimensional characteristics and the weight of a reference intercooler can be readily calculated by the use of the equations previously given. A sample computation in which a plate-type intercooler is designed from the information presented in this report is given in appendix B. Figure 6 gives, for a given set of operating conditions, the variation in plate-type intercooler dimensions and weight with plate spacing.

APPARATUS AND METHODS

An intercooler unit having closely spaced plates was tested for the purpose of providing some experimental data with which to check the results of the analysis. Figure 7 is a photograph of the test assembly used in this work. The test equipment consisted essentially of a centrifugal blower driven by a Ford V-8 engine, an intercooler test unit, an electric air heater, thermocouples, manometers, air-flow measuring orifices, a potentiometer, and a multiple-switch box. The charge air and the cooling air were drawn through the intercooler by the blower, the relative quantities of the two air flows being controlled by valves as shown in the figure.

The charge air was heated electrically by units encased in a thermally insulated drum situated above the intercooler test unit. Entrance and exit temperatures were measured with iron-constantan thermocouples. At each entrance and exit four thermocouples were distributed diagonally across the rectangular flow section to obtain average temperatures of the two air streams. In each exit channel, an extra set of four thermocouples was extended across the channel in a row parallel to and downstream from a slot between two baffle plates, as shown in figure 8. These extra sets were installed to determine the advisability of mixing the exit air before measuring its temperature. It was found, however, that a better heat balance between the warmed and cooled air was obtained with exit temperatures measured upstream from the baffles. Those upstream temperature values were consequently used to determine the performance of the intercooler unit. The use of such baffles in intercooler test work is not condemned because of these observations, but it is believed rather that this subject should be investigated further. As shown in figure 8, baffles were placed upstream from the charge-entrance set of thermocouples to insure a uniform temperature distribution of the charge air before it entered the intercooler. Tests made both with and without these baffles showed no noticeable change in the performance of the test unit in either pressure drop or cooling. It is not concluded, however, that the baffles would have no effect for all flow conditions.

The four thermocouples of each set were connected in series and the cold junctions were placed in the multiple-switch box, which also served as a cold-junction box. The temperature indications thus obtained were four times the average.

Air-weight-flow measurements were made by means of flange-tap orifice plates in accordance with the procedure outlined by the American Society of Mechanical Engineers (reference 5).

A static pressure tap was situated on each side of the rectangular entrance or exit section where there could be no interference by thermocouples or their supports (fig. 8). The four taps of each flow section were connected to a common tube that led to a manometer. All taps were situated about 1 inch from the intercooler test unit.

The intercooler test unit consisted of 35 brass heat-transfer plates - each 4.30 inches long, 2.30 inches wide, and 0.008 inch thick - and two end plates of the same length and width but of 0.038 inch thickness. (See figs. 9 and 10.) These plates were separated by suitably arranged separators that provided alternate passages for the cross flow of the charge air and the cooling air. In the tests the charge air flowed along the length of the plates and the cooling air flowed along the width of the plates. The intercooler end separators were 0.25 inch wide and 0.024 inch thick and the intermediate separators were of the same thickness and of 0.125 inch width. The intermediate separators were arranged as follows: one row through the center of the charge-air passages and three rows equally spaced through the cooling-air passages, as shown in figures 9 and 10. Each plate was curved to a radius of curvature of 3.5 inches about an axis parallel to the plate length. The intermediate separators and the curvature were used to reduce the bending of the plates under the pressure that would be encountered in actual intercooler operation.

Before assembly, all separators and plate-contact surfaces were tinned with soft solder. The plates and the separators were then assembled in a jig together with aluminum strips 0.022 inch thick. These strips were placed in the air spaces between the separators in order that pressure could be applied equally on all contact surfaces. This assembly, under pressure in the jig, was subjected to a temperature of about 460° F in an oven for about 20 minutes. The unit was then allowed to cool in the oven for about one hour before removal. After the unit had cooled, the aluminum strips were withdrawn from the air passages. The average thickness of the air spaces was then determined as 0.0248 inch for the charge-air passages and 0.0243 inch for the cooling-air passages.

Previously, tests had been made with a similar intercooler unit having 0.006 inch plates to determine the amount of pressure required to bend the plates appreciably. Pressure was applied to one set of air passages and the order of magnitude of the plate deflection was estimated in the other set of air passages by means of a feeler strip 0.020 inch thick. Under a gage pressure of 30 inches of mercury no noticeable resistance to the motion of the feeler in the passages was observed. When the gage pressure was increased to 40 inches of mercury, an appreciable resistance to the feeler-strip motion was detected. It was therefore decided to increase the plate thickness of the intercooler unit to be tested to 0.008 inch. No recommendations can be made in this paper with regard to the proper distance between plate separators, the plate thickness, and the plate curvature necessary for adequate plate strength. This information should be obtained from endurance tests.

The intercooler unit was placed in a cross-flow duct as shown in figures 7 and 8. The entire duct and thermocouple assembly was thermally insulated from the surrounding atmosphere. This insulation is not shown in the figures.

The intercooler test conditions covered are given in table III. For each test condition the charge-air and the cooling-air weight flows and the charge-air and cooling-air temperatures and pressures at the intercooler entrance and exit sections were measured.

DISCUSSION AND RESULTS

Charts of Intercooler Characteristics

Figures 2 and 3 show how the plate area, the weight, the volume, the frontal area, and the linear dimensions of a plate-type intercooler vary with plate spacings and plate thickness when the intercooler operating conditions remain constant. Although the effect of plate separators is not accounted for in figures 2 and 3, the trends indicated should be substantially correct because, for a given separator width and arrangement, the effect of plate separators does not appreciably vary with change in plate spacings or plate thickness. It is seen from figures 2 and 3 that, for both turbulent and laminar flow, a reduction in plate spacing results in a decrease in both intercooler volume and weight when the intercooler operating conditions

remain constant. This decrease in intercooler volume and weight is attained, however, at the expense of a large increase in intercooler width and a slight increase in intercooler frontal area.

It can be shown by means of equations (26), (27), (32), and (33) that the minimum width per unit intercooler volume is obtained when $s_2/s_1 = 1.00$. This fact is illustrated in figures 2 and 3 where it is seen that, for a given volume, the case in which $s_2/s_1 = 1.00$ provides a smaller intercooler width than does the case in which $s_2/s_1 = 0.50$. The difference in width per unit intercooler volume is small, however, and little sacrifice in width will be made by use of $s_2/s_1 = 0.50$ if this ratio gives an intercooler, proportions of which better fit the space requirements. For equal volumes the case for $s_2/s_1 = 0.50$ provides a smaller A_f, s_2, and l_2 and a larger s_1 and l_1 than does the case for $s_2/s_1 = 1.00$. For example, for $v/v_o = 0.50$ and $t = 0.01$ inch, figure 2(a), which applies for the case of turbulent flow gives

Values for \ s_2/s_1	1.00	0.50
s_1, inch	0.13	0.18
s_2, inch	.13	.09
w/w_o	2.35	2.60
l_1/l_{1_o}	.47	.65
l_2/l_{2_o}	.47	.31
A_f/A_{f_o}	1.08	.79
S/S_o	.88	.87
W/W_o	.88	.87

Figures 2 and 3, although plotted for reference-plate spacings equal to 0.25 and 0.10 inch, respectively, and for zero reference-plate thickness, can also be used for other values of reference-plate spacings and plate thickness. For example, let it be required to find the change in intercooler volume caused by the change in plate spacings s_1 and s_2 from 0.15 inch to 0.05 inch if the intercooler operating conditions remain constant. The intercooler plates are 0.01 inch thick and the air flows are assumed to be turbulent. From figure 2(a),

$$\frac{v_{0.05}}{v_o} = 0.17$$

$$\frac{v_{0.15}}{v_o} = 0.57$$

$$\frac{v_{0.05}}{v_{0.15}} = \frac{0.17}{0.57} = 0.30$$

The volume is then reduced to 30 percent of its initial value.

It is seen in figure 4 that, for laminar flow, the reference-intercooler plate area is a function of η, M_1, and M_2 and that, for turbulent flow, it is, in addition, a function of $\sigma_{1_{av}} \Delta p_{f_1}$ and $\sigma_{2_{av}} \Delta p_{f_2}$. Equations (13), (25), and (26) for turbulent flow and equations (19) and (32) for laminar flow show that, for given plate spacings and plate thickness, the intercooler volume is proportional to the plate area. Figure 4, together with figures 2 and 3, can then be used to compare turbulent and laminar flow on the basis of intercooler volume. For example, a comparison at $s_1 = s_2 = 0.10$ inch and $t = 0.01$ inch is made as follows:

From figure 2(a) for $s_1 = 0.10$ inch and $t = 0.01$ inch, $v/v_o = 0.36$ for turbulent flow. The ordinates in figure 4(a) are multiplied by this ratio. From figure 3(a) for $s_1 = 0.10$ inch and $t = 0.01$ inch, $v/v_o = 1.10$ for laminar flow. The ordinates in figure 4(b) are multiplied by this ratio.

The results of this procedure are now directly comparable on the basis of intercooler volume. These results show that, for the range of operating conditions covered in figure 4, the intercooler volume for the same operating conditions is much smaller for turbulent flow than for laminar flow when $s_1 = 0.10$ inch and $s_2/s_1 = 1.00$ or 0.50. Table IV shows a comparison of the intercooler volumes for turbulent and for laminar flow at $s_1 = 0.05$ inch and $s_2/s_1 = 1.00$ and 0.50 and for $\sigma_{2_{av}} \Delta p_{f_2} = 6$ inches of water and $\theta = 0.25$. For these conditions it is noted that the intercooler volume for laminar flow may be greater or less than that for turbulent flow, depending on the operating conditions. The intercooler volume for laminar flow tends to approach that for turbulent flow as s_1 is reduced. For any given set of operating conditions a value of s_1 may be reached at which the two volumes are the same and below which the volume for laminar flow is less than the volume for turbulent flow.

Figure 5 gives the relation between plate area and charge-air length-spacing ratio for a reference intercooler whose reference-plate spacings are 0.25 inch for turbulent flow and 0.10 inch for laminar flow.

As a further illustration of the effect of type of flow and of plate spacing on the physical properties of an intercooler when the operating conditions are held constant, the weight and dimensional characteristics of a plate-type intercooler are plotted in figure 6 against plate spacings for $s_2/s_1 = 1.00$ and 0.50 for the case of turbulent and laminar flow. The plate thickness is taken as 0.01 inch, and the operating conditions are

η, percent 65

M_1/M_2 2

$\sigma_{1_{av}} \Delta p_{f_1}$, inches of water 3

$\sigma_{2_{av}} \Delta p_{f_2}$, inches of water 6

The weight and the dimensional characteristics were calculated first for a reference intercooler, which has the reference-plate spacings s_{1_o} (for turbulent flow) and s_{1_o}' (for laminar flow) equal to 0.25 inch and 0.10 inch, respectively. (See figs. 1, 4, and 5 and equations (13), (15), (16), (17), (19), (21), (22), and (23).) Figures 2 and 3 were used to determine the variation of the physical properties of the intercooler with spacing for a plate thickness of 0.01 inch. No corrections for separators were applied. The Reynolds number of flow and the ratio of the total pressure drop to the friction pressure drop for an intercooler with blunt entrance and exit sections are also plotted in figure 6. In the calculations for the total pressure drop the heating pressure loss, or gain, which is a function of the air temperatures in the intercooler, was not included.

Table I gives representative values of the entrance-exit loss, the velocity-profile loss, and the heating pressure loss or gain for an intercooler with blunt entrance and exit sections. The velocity-profile pressure loss is given in terms of the friction pressure drop. The entrance-exit pressure loss and the heating pressure loss or gain are given as functions of the velocity-profile loss because of the reduction in the number of variables involved. The heating pressure change (due to the momentum change of the air flow) is a pressure drop when the air is being heated and a pressure gain when the air is being cooled. When the intercooler entrance and exit sections are streamlined, the entrance-exit pressure loss is eliminated and the velocity-profile pressure loss and heating pressure loss or gain are reduced.

During actual intercooler operation, the pressure difference between the charge air and the cooling air across each plate causes the plates to deflect. In order to reduce plate deflections, intermediate separators are used. The number of separators required for a given permissible deflection will depend on the maximum pressure difference across each plate and on the plate material and thickness. Proper allowances should be made for separators in the determination of the intercooler dimensions and weight. These allowances are given in table II in the form of corrections to be applied to the weight and to the dimensional characteristics of a plate-type intercooler that is assumed to have no plate separator strips. An example of the application of these corrections is given in appendix B.

An increase in the resistance of the plates to bending can be achieved by the utilization of curved plates. A reduction in the number of separators can thus be obtained. Curving the plates increases the heat transfer and the pressure drop of the air flowing along the plate curvature. References 3 and 6 show these increases to be small for the range of plate spacings and radii of curvature used in plate-type intercoolers.

Results of Tests of the 2.3- by 4.3-Inch Unit

The performance data of the 2.3- by 4.3-inch test unit previously described are shown in figure 11. The quantity $\sigma_{1_{av}} \Delta p_1$ is plotted against $\sigma_{2_{av}} \Delta p_2$ for various values of η. Figure 11 also includes a plot of M_1/w against $\sigma_{1_{av}} \Delta p_1$ and of M_2/w against $\sigma_{2_{av}} \Delta p_2$. The dimensions in the direction of air flow and the intercooler weight per unit width are also given. The data given in this figure apply to any width of intercooler, provided that all other dimensions remain constant.

The performance chart (figure 11) may be used to determine the performance of the intercooler in the following manner. For a given value each of $\sigma_{1_{av}} \Delta p_1$ and η a value of $\sigma_{2_{av}} \Delta p_2$ may be read at the bottom of the chart. For this value of $\sigma_{2_{av}} \Delta p_2$ the corresponding value of M_2/w may be read at the right of the chart by the use of the long-dash curve. In a similar manner the short-dash curve may be used to find the value of M_1/w corresponding to the given value of $\sigma_{1_{av}} \Delta p_1$.

In any specific application the approximate value of $\sigma_{1_{av}} \Delta p_1$ is known from the atmospheric conditions and flight speed. The required value of cooling effectiveness is determined from the known supercharger outlet temperature and required intercooler outlet temperature. The other quantities are read from the figure in the manner described. The quantity M_2 is known from the engine horsepower and therefore the intercooler width w, weight W, and cooling-air weight flow M_1, can be readily computed.

The general characteristics of the 2.3- by 4.3-inch intercooler tested are compared with those of two Airesearch and three Harrison intercoolers in figure 12 for a value of $\sigma_{1_{av}} \Delta p_1$ equal to 4 inches of water. The basis for this method of comparison is discussed in reference 1. It may be seen that cooling is accomplished by the intercooler of these tests and by the commercial types in the same general range of charge-air pressure drops and required cooling air. In volume, the intercooler tested is much lower, and in weight the frontal area, generally higher than the commercial types. The outstanding advantage of the intercooler tested is its small volume. Its disadvantage in weight can be materially reduced if narrower separator strips are used. The separator strips in this test specimen were unnecessarily wide and had a weight equal to 71 percent of the weight of the heat-transfer plates. If the plates and separators had been bound by welding instead of by soldering, further reduction in weight would have resulted.

In figure 12 it is seen that the width w of the intercooler tested is much greater than that of the commercial types but the dimensions in the direction of air flow are considerably less. The resulting general shape and the smaller volume of this intercooler make for convenient installations in wings or cowlings. The large width offers no special difficulty as the intercooler may be installed as a number of parallel segments.

The intercooler tested is merely illustrative and other proportions may allow more convenient installations.

Figure 13 gives the experimental values of the cooling effectiveness of the intercooler unit plotted as a function of the Reynolds numbers of the charge-air and cooling-air flows. From this plot can be determined the relationship governing the transfer of heat for the test-flow conditions encountered in this investigation. It has been previously mentioned that the cooling effectiveness of a cross-flow intercooler is a function of M_1/M_2 and UA_r/M_2c_p. (See fig. 1.) The charge-air and cooling-air weight flows, M_2 and M_1, can be evaluated in terms of their respective Reynolds numbers from the known air temperatures in the intercooler and the known intercooler dimensions. From the experimentally determined values of η, M_1, and M_2 given indirectly in figure 13 and the curves in figure 1, the term (UA_r) can be evaluated for each test condition.

If the Nusselt numbers $\frac{h_1 d_1}{k_1}$ and $\frac{h_2 d_2}{k_2}$ of the two flows are assumed to be some unknown function of Reynolds number, they can be determined only for equal Reynolds numbers of the charge-air and cooling-air flows. Under this condition

$$\frac{h_1 d_1}{k_1} = \frac{h_2 d_2}{k_2}$$

or

$$h_2 = h_1 \left(d_1 k_2 / d_2 k_1\right)$$

If this expression for h_2 is substituted in equation (41) of appendix A, the Nusselt number is given by

$$\frac{h_1 d_1}{k_1} = \frac{h_2 d_2}{k_2} = UA_r \left(\frac{d_1}{A_1 k_1} + \frac{d_2}{A_2 k_2}\right)$$

The Nusselt numbers were thus determined for the four points of equal Reynolds number of charge and cooling air shown in figure 13 and were plotted against the Reynolds number, as shown in figure 14, by the solid curve. The horizontal dashed curve in figure 14 represents the heat-transfer relationship for laminar flow given in reference 4 and the dashed curve with the slope of 0.8 represents the relationship for turbulent flow given in reference 3, page 172. The experimental curve appears to join the curve for laminar flow and to approach the curve for turbulent flow very satisfactorily in both slope and absolute value.

By means of figures 1 and 14 the cooling effectiveness of the intercooler unit was computed for each test condition. The results are shown by the dashed curves of figure 13. The agreement with experimental values at all Reynolds numbers is quite satisfactory.

In figure 15 the charge-air and the cooling-air pressure drops across the intercooler test unit are plotted against Reynolds number. The solid curve joins experimental points obtained from cold runs on the intercooler unit and checked by data from heating runs. The long dashed curves and the

short dashed curves were computed for laminar and turbulent flow, respectively, and include the surface-friction, the entrance-exit, and the velocity-profile loss. Figure 15 shows that, for intercooler operation in the transition region - for example, for Reynolds numbers between 2000 and 3000, the computed pressure-drop values for laminar flow are not very much different from the values for turbulent flow. For this reason, the region of transition is not clearly defined in the experimental curve. As a whole, there is fair agreement between the experimental and the computed pressure-drop values.

CONCLUSIONS

1. For a given set of operating conditions, which include cooling effectiveness, air-weight flows, and air-friction pressure drops, a reduction in plate-type intercooler volume can be made at the expense of increased intercooler width by decreasing the intercooler plate spacings and plate dimensions.

2. For a plate spacing of 0.10 inch or greater and for a given set of operating conditions in the range of practical interest, the intercooler volume is greater if the air flows in the intercooler passages are laminar than if the air flows are turbulent. As the plate spacing is reduced, the volume required for laminar flow approaches and eventually becomes less than that required for turbulent flow, the spacing at which the volumes are equal being a function of the operating conditions.

3. The dimensions of a plate-type intercooler that satisfies a definite set of operating conditions can be determined with fair accuracy for any plate spacing.

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APPENDIX A

DERIVATION OF EQUATIONS SHOWING THE RELATION BETWEEN THE DIMENSIONAL AND WEIGHT CHARACTERISTICS AND THE OPERATING CONDITIONS

If the thermal resistance of the intercooler metal is neglected, the over-all heat-transfer coefficient is given by

$$\frac{1}{UA_r} = \frac{1}{h_2A_2}\left(1 + \frac{h_2A_2}{h_1A_1}\right) \tag{41}$$

The charge-air weight flow is given by

$$M_2 = \rho_2V_2Nl_1s_2J_2 \tag{42}$$

and the effective heat-transfer area on the charge-air side by

$$A_2 = 2Nl_1l_2K_2 \tag{43}$$

From equations (42) and (43)

$$\frac{l_2}{s_2} = \frac{\rho_2V_2}{2}\left(\frac{A_2}{M_2}\right)L_2 \tag{44}$$

Similarly,

$$\frac{l_1}{s_1} = \frac{\rho_1V_1}{2}\left(\frac{A_1}{M_1}\right)L_1 \tag{45}$$

The intercooler volume may be expressed as

$$v = \frac{A_2}{2K_2}(s_1 + s_2 + 2t) \tag{46}$$

the intercooler width as

$$w = \frac{v}{l_1l_2} \tag{47}$$

the intercooler frontal area as

$$A_f = wl_2 \tag{48}$$

and the intercooler weight as

$$W = \rho_m t \frac{A_2}{K_2} + \rho_m \frac{A_2}{K_2}\left(1 - J_2\right)\frac{s_2}{2} + \rho_m \frac{A_2}{K_2}\left(1 - J_1\right)\frac{s_1}{2} \tag{49}$$

For turbulent flow

$$\frac{hs}{k} = 0.0172\ r^{\frac{1}{5}}\left(\frac{\rho V s}{\mu}\right)^{\frac{4}{5}} \tag{50}$$

$$\sigma_{av}\Delta p_f = \left(\frac{0.0854}{10.4 g\rho_o}\right) r^{\frac{6}{5}}\ (\rho V)^2\ \left(\frac{\rho V s}{\mu}\right)^{-\frac{1}{5}}\ \frac{l}{s} \tag{51}$$

From equations (44), (50), and (51)

$$h_2 = \frac{0.0172 k_2}{\mu_2^{\frac{6}{7}}}\left(\frac{10.4 g\rho_o}{0.0854}\right)^{\frac{2}{7}} \frac{\left(\sigma_{2_{av}}\Delta p_{f_2}\right)^{\frac{2}{7}}}{\left(s_2 r_2\right)^{\frac{1}{7}}}\left(\frac{2M_2}{L_2 A_2}\right)^{\frac{2}{7}} \tag{52}$$

similarly

$$h_1 = \frac{0.0172 k_1}{\mu_1^{\frac{6}{7}}}\left(\frac{10.4 g\rho_o}{0.0854}\right)^{\frac{2}{7}} \frac{\left(\sigma_{1_{av}}\Delta p_{f_1}\right)^{\frac{2}{7}}}{\left(s_1 r_1\right)^{\frac{1}{7}}}\left(\frac{2M_1}{L_1 A_1}\right)^{\frac{2}{7}} \tag{53}$$

By definition

$$\frac{A_1}{A_2} = \frac{K_1}{K_2} \tag{54}$$

Then, from equations (52), (53), and (54)

$$\frac{A_1 h_1}{A_2 h_2} = \theta^{\frac{2}{7}}\left(\frac{s_2 r_2}{s_1 r_1}\right)^{\frac{1}{7}}\left(\frac{K_1}{K_2}\right)^{\frac{5}{7}}\left(\frac{L_2}{L_1}\right)^{\frac{2}{7}}\left(\frac{k_1}{k_2}\right)\left(\frac{\mu_2}{\mu_1}\right)^{\frac{6}{7}} \tag{55}$$

For any practical range of intercooler operating temperatures the term $\left(\frac{k_1}{k_2}\right)\left(\frac{\mu_2}{\mu_1}\right)^{\frac{6}{7}}$ in equation (55) differs from unity by a negligible amount. This term will therefore be replaced by unity.

The substitution of equation (55) in equation (41) gives

$$\frac{1}{UA_r} = \frac{1}{h_2 A_2}\left[1 + \theta^{-\frac{2}{7}}\left(\frac{s_1 r_1}{s_2 r_2}\right)^{\frac{1}{7}}\left(\frac{K_2}{K_1}\right)^{\frac{5}{7}}\left(\frac{L_1}{L_2}\right)^{\frac{2}{7}}\right] \tag{56}$$

From equations (52) and (56) the solution for A_2/M_2 is

$$\frac{A_2}{M_2} = \left(\frac{0.0854\mu_2^{\,3}}{2 \times 10.4 g\rho_o}\right)^{\frac{2}{5}}\left(\frac{c_p}{0.0172k_2}\right)^{\frac{7}{5}}\left(r_2 s_1\right)^{\frac{1}{5}}\left(\frac{L_2}{\sigma_{2_{av}}\Delta p_{f_2}}\right)^{\frac{2}{5}}$$

$$\left(\frac{UA_r}{M_2 c_p}\right)^{\frac{7}{5}}\left[\left(\frac{s_2}{s_1}\right)^{\frac{1}{7}} + \theta^{-\frac{2}{7}}\left(\frac{r_1}{r_2}\right)^{\frac{1}{7}}\left(\frac{K_2}{K_1}\right)^{\frac{5}{7}}\left(\frac{L_1}{L_2}\right)^{\frac{2}{7}}\right]^{\frac{7}{5}} \tag{57}$$

The substitution of equation (51) in equation (44) gives

$$\frac{l_2}{s_2} = \left(\frac{10.4g\rho_o}{0.0854}\right)^{\frac{5}{14}}\frac{s_2^{\frac{1}{14}}}{\mu_2^{\frac{1}{14}} r_2^{\frac{3}{7}}}\left(\frac{L_2 A_2}{2M_2}\right)^{\frac{9}{14}}\left(\sigma_{2_{av}}\Delta p_{f_2}\right)^{\frac{5}{14}} \tag{58}$$

The ratio $\frac{l_1}{s_1}$ can be similarly found and expressed as

$$\frac{l_1/s_1}{l_2/s_2} = \theta^{\frac{5}{14}}\left(\frac{r_2}{r_1}\right)^{\frac{3}{7}}\left(\frac{L_1 K_2}{L_2 K_1}\right)^{\frac{9}{14}}\left(\frac{M_2}{M_1}\right)\left(\frac{s_1}{s_2}\right)^{\frac{1}{14}}\left(\frac{\mu_2}{\mu_1}\right)^{\frac{1}{14}} \tag{59}$$

As in the previous case, in the practical range of temperatures, the term $\left(\mu_2/\mu_1\right)^{\frac{1}{14}}$ can be replaced by unity.

For laminar flow

$$\frac{hs}{k} = 3.8r \tag{60}$$

The friction factor for flow between parallel flat plates of infinite width is $\frac{24\mu}{2\rho Vs}$ where s is the perpendicular distance between the two plates. This friction factor can be used with good accuracy for flow through rectangular channels. (See reference 3.)

Thus,

$$\sigma_{av}\Delta p_f = \frac{24\mu}{10.4g\rho_o}\,\rho Vr\,\frac{l}{s^2} \tag{61}$$

The substitution of equations (54) and (60) in equation (41) gives

$$\frac{1}{UA_r} = \frac{s_1}{3.8k_2A_2r_2}\left(\frac{s_2}{s_1} + \frac{r_2K_2k_2}{r_1K_1k_1}\right) \tag{62}$$

The solution for A_2/M_2 is, then,

$$\frac{A_2}{M_2} = \left(\frac{c_p}{3.8\ k_2}\right)\left(\frac{s_1}{r_2}\right)\left(\frac{UA_r}{M_2c_p}\right)\left(\frac{s_2}{s_1} + \frac{r_2K_2k_2}{r_1K_1k_1}\right) \tag{63}$$

The substitution of equation (44) in equation (61) results in

$$\sigma_{2_{av}}\Delta p_{f_2} = \left(\frac{24\mu_2}{10.4g\rho_o}\right)\left(\frac{2M_2}{A_2L_2}\right)\left(\frac{r_2}{s_2}\right)\left(\frac{l_2}{s_2}\right)^2 \tag{64}$$

and

$$\frac{l_2}{s_2} = \left(\frac{10.4gp_o}{24\mu_2}\right)^{\frac{1}{2}} \left(\frac{A_2L_2}{2M_2}\right)^{\frac{1}{2}} \left(\frac{s_2}{r_2}\right)^{\frac{1}{2}} \left(\sigma_{2_{av}}\Delta p_{f_2}\right)^{\frac{1}{2}} \tag{65}$$

The ratio l_1/s_1 can be similarly found and expressed as

$$\frac{l_1/s_1}{l_2/s_2} = \theta^{\frac{1}{2}} \left(\frac{s_1}{s_2}\right)^{\frac{1}{2}} \left(\frac{K_1L_1}{K_2L_2}\right)^{\frac{1}{2}} \left(\frac{r_2}{r_1}\right)^{\frac{1}{2}} \left(\frac{M_2}{M_1}\right) \left(\frac{\mu_2}{\mu_1}\right)^{\frac{1}{2}} \tag{66}$$

Equations (57), (58), and (59) for turbulent flow and equations (63), (65), and (66) for laminar flow, together with equations (46) to (49) and figure 1 give the weight and dimensional characteristics of a plate-type intercooler in terms of the plate spacings, plate thickness, and intercooler operating conditions. The effect of plate spacers on the intercooler weight and on the dimensional characteristics is accounted for in these equations by the substitution of the proper values of r, J, K, and L.

In the Analysis the foregoing relations are given for a reference intercooler (equations (12) to (17) for turbulent flow and equations (18) to (23) for laminar flow), which is defined as a plate-type intercooler having the following characteristics:

1. Charge-air and cooling-air plate spacings (s_1 and s_2) equal to a reference spacing

2. Zero plate thickness ($t = o$)

3. No plate separators ($r = J = K = L = 1$)

The variation of the intercooler weight and of the dimensional characteristics as the plate spacings and plate thickness vary from the reference case is then given for constant operating conditions (equations (25) to (30) for turbulent flow and equations (31) to (36) for laminar flow). The correction factors due to the effect of plate separators are given in table II.

APPENDIX B

ILLUSTRATION OF PLATE-TYPE INTERCOOLER DESIGN

Operating Conditions

Let it be supposed that a plate-type intercooler is to be designed for the following set of operating conditions:

(1) η, percent 70

(2) M_2, lb/sec 4.4

(3) M_1/M_2 2

(4) $\sigma_{1_{av}} \Delta p_{f_1}$, in. water 3

(5) $\sigma_{2_{av}} \Delta p_{f_2}$, in. water 6

The intercooler width is assumed to be limited to approximately 50 inches.

Design Procedure

The charge-air and the cooling-air flows are assumed to be turbulent. This assumption can be checked after the design is completed.

(6) From items (3), (4), and (5)

$$\theta = \frac{M_1 \sigma_{1_{av}} \Delta p_{f_1}}{M_2 \sigma_{2_{av}} \Delta p_{f_2}} = 1$$

(7) From figure 4(a) and items (1), (3), (5), and (6)

$$\frac{S_o}{M_2} = 28{,}800 \text{ sq in./lb/sec}$$

(8) From equation (13) and item (7)

$$\frac{v_o}{M_2} = 7200 \text{ cu in./lb/sec}$$

(9) From figure 5 and items (5) and (7)

$$\frac{l_{2_o}}{s_{2_o}} = 477$$

(10) From equation (15) and items (3), (6), and (9)

$$\frac{l_{1_o}}{s_{1_o}} = 238$$

(11) From equation (16) and items (7), (9), and (10)

$$\frac{w_o}{M_2} = 1.015 \text{ in./lb/sec}$$

(12) If $w = 50$ in., from items (2) and (11),

$$\frac{w}{w_o} = \frac{50}{1.015 \times 4.4} = 11.2$$

Let $s_2/s_1 = 1$ and $t = 0.01$ in.

(13) Then, from figure 2(a) and item (12)

$$s_1 = s_2 = 0.04 \text{ in.}$$

As a result,

(14) $\frac{v}{v_o} = 0.14$

(15) $\frac{l_2/s_2}{l_{2_o}/s_{2_o}} = \frac{l_1/s_1}{l_{1_o}/s_{1_o}} = \frac{S}{S_o} = 0.70$

The intercooler dimensions may then be given as follows:

(16) From items (2), (7), and (15)

$$S = 88{,}700 \text{ sq in.}$$

(17) From items (9), (13), and (15)

$$l_2 = 13.35 \text{ in.}$$

(18) From items (10), (13), and (15)

$$l_1 = 8.66 \text{ in.}$$

(19) From items (2), (8), and (14)

$$v = 4435 \text{ cu in.}$$

(20) $w = 50$ in.

The charge-air and cooling-air Reynolds numbers can now be calculated and are found to be approximately 2200 and 2500, respectively. The flows are then in the transition region and are probably turbulent, as previously assumed, because of the turbulence existing in the entrance ducting and induced at the sharp-edged entrances of the intercooler.

Corrections for Plate Separators

It is further assumed that separators are placed approximately 1 inch apart in the charge-air and in the cooling-air passages and that the intermediate and end separators have a width of 0.04 inch and 0.06 inch, respectively. There will then be 14 separators for each cooling-air passage and 7 separators for each charge-air passage.

The terms r, J, K, and L are defined in the Symbols as functions of the intercooler dimensions and of the plate-separator dimensions and arrangement.

$$r_1 = r_2 = 1.04$$

$$J_1 = \frac{13.35 - (12 \times 0.04) - (2 \times 0.06)}{13.35} = 0.952$$

$$J_2 = \frac{6.66 - (5 \times 0.04) - (2 \times 0.08)}{6.66} = 0.946$$

For a fin effectiveness of 100 percent,

$$K_1 = \frac{12.71 + (13 \times 0.04)}{13.35} = 0.991$$

$$K_2 = \frac{6.30 + (6 \times 0.04)}{6.66} = 0.982$$

Then,

$$L_1 = 0.960$$

$$L_2 = 0.964$$

The substitution of r, K, and L in equation (37) gives, for $s_2/s_1 = 1$ and $\theta = 1$ (item 6),

$$\psi_t = 1.009$$

The correction factors are given in table II as functions of r, J, K, L, and ψ_t. The corrections may then be given as

S	1.01
l_2	0.96
l_1	0.96
v	1.01
w	1.10

These corrections may be applied to the dimensions of the intercooler with no plate separators (items (16) to (20)).

Then,

(21) S = 89,600 sq. in.

(22) $l_2 = 12.8$ in.

(23) $l_1 = 6.4$ in.

(24) $v = 4479$ cu in.

(25) $w = 55$ in.

If s_2/s_1 had been chosen as 0.50 instead of 1.00, the values of l_1 and l_2 would be approximately equal.

Total Pressure Drops

Table I gives the velocity profile, the momentum heating, and the entrance-exit losses in terms of l/s, f, β, and R.

(26) From item (22) for $s_2 = 0.04$ in.

$$\frac{l_2}{s_2} = 320$$

(27) From item (23) for $s_1 = 0.04$ in.

$$\frac{l_1}{s_1} = 160$$

(28) For $s_1 = s_2 = 0.04$ in. and $t = 0.01$ in.
$f_1 = f_2 = 0.4$ approximately.

The term β is a function of the entrance and exit temperatures of the air flowing through the intercooler. Let it be assumed that the entrance temperatures are 250° and 50° F for the charge and the cooling air, respectively.

Then,

(29) From item (1) the charge-air exit temperature is

$$250 - 0.70(250 - 50) = 110^\circ \text{ F}$$

(30) From items (3) and (29) the cooling-air exit temperature is

$$\frac{250 - 110}{2} + 50 = 120^{\circ}\ F$$

therefore,

(31) $\beta_2 = -0.20$

and

(32) $\beta_1 = 0.14$

From table I for the charge-air and cooling-air Reynolds numbers of about 2200 and 2500, respectively,

(33) $\Delta p_{v_2} / \Delta p_{f_2} = 0.013$

(34) $\Delta p_{v_1} / \Delta p_{f_1} = 0.026$

(35) $\Delta p_{H_2} / \Delta p_{v_2} = -4.4$

(36) $\Delta p_{H_1} / \Delta p_{v_1} = 3.1$

(37) $\Delta p_{e_2} / \Delta p_{v_2} = \Delta p_{e_1} / \Delta p_{v_1} = 7.8$

Therefore,

(38) From items (5), (33), (35), and (37)

$$\sigma_{2_{av}} \Delta p_2 = 6.3 \text{ in. water}$$

and

(39) From items (4), (34), (36), and (37)

$$\sigma_{1_{av}} \Delta p_1 = 3.9 \text{ in. water}$$

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TABLE I

VELOCITY-PROFILE, HEATING, AND ENTRANCE-EXIT PRESSURE CHANGES OF A PLATE-TYPE INTERCOOLER

[The ratio of $r = \frac{2s}{d}$ is assumed equal to unity. This assumption introduces very negligible error.]

VELOCITY-PROFILE LOSS, $\Delta p_v/\Delta p_f$						
R	Type of flow \ l/s	100	150	200	250	300
2,000	Turbulent	0.042	0.028	0.021	0.017	0.014
5,000		.031	.034	.026	.020	.017
10,000		.038	.039	.029	.023	.020
500	Laminar	.057	.038	.028	.023	.019
1,000		.114	.076	.057	.046	.038
2,000		.228	.152	.114	.091	.076

Pressure change caused by momentum change resulting from heating or cooling, $\Delta p_H/\Delta p_v$ (1)			
Type of flow \ β	0.05	0.10	0.20
Turbulent	1.11	2.22	4.44
Laminar	.18	.37	.73

Entrance-exit loss, $\Delta p_e/\Delta p_v$			
Type of flow \ f	0.2	0.4	0.6
Turbulent	11.8	7.8	4.7
Laminar	1.8	1.0	.33

[1]The expression Δp_H is a pressure drop when β is positive and a pressure gain when β is negative.

TABLE II

CORRECTION FACTORS DUE TO EFFECTS OF PLATE SEPARATORS ON WEIGHT AND DIMENSIONAL CHARACTERISTICS OF A PLATE-TYPE INTERCOOLER

Values for	Correction Factor	
	Turbulent Flow	Laminar Flow
s	ψ_t	ψ_L
l_1	$(\psi_t J_1)^{\frac{9}{14}} \quad r_1^{-\frac{3}{7}}$	$(\psi_L J_1)^{\frac{1}{2}} \quad r_1^{-\frac{1}{2}}$
l_2	$(\psi_t J_2)^{\frac{9}{14}} \quad r_2^{-\frac{3}{7}}$	$(\psi_L J_2)^{\frac{1}{2}} \quad r_2^{-\frac{1}{2}}$
w	$(r_1 r_2)^{\frac{3}{7}} \quad (J_1 J_2)^{-\frac{9}{14}} \quad \psi_t^{-\frac{2}{7}}$	$(r_1 r_2)^{\frac{1}{2}} \quad (J_1 J_2)^{-\frac{1}{2}}$
v	ψ_t	ψ_L
A_f	$\psi_t^{\frac{5}{14}} \quad r_1^{\frac{3}{7}} \quad J_1^{-\frac{9}{14}}$	$\psi_L^{\frac{1}{2}} \quad r_1^{\frac{1}{2}} \quad J_1^{-\frac{1}{2}}$
W	$\psi_t\left[1+\frac{(1-J_2)}{2}\frac{s_2}{t}+\frac{(1-J_1)}{2}\frac{s_1}{t}\right]$	$\psi_L\left[1+\frac{(1-J_2)}{2}\frac{s_2}{t}+\frac{(1-J_1)}{2}\frac{s_1}{t}\right]$

TABLE III

INTERCOOLER TEST CONDITIONS

Test series	Rate of charge-air flow (lb/sec)	Rate of cooling-air flow (lb/sec)	Temperature of charge air at entrance (°F)
1	0.0174 to 0.0179	Varied from 0.0148 to 0.1033 in 8 steps	273 to 306
2	.0254 to .0261	Varied from 0.0148 to 0.1026 in 8 steps	277 to 299
3	.0335 to .0340	Varied from 0.0147 to 0.1037 in 8 steps	275 to 295
4	.0413 to .0425	Varied from 0.0148 to 0.1056 in 8 steps	280 to 294

TABLE IV

EFFECT OF THE TYPE OF FLOW ON INTERCOOLER VOLUME FOR EQUAL OPERATING CONDITIONS

$$\left[\sigma_{2_{av}} \Delta p_{f_2} = 6 \text{ in. water}; \quad \theta = 0.25; \quad s_1 = 0.05\right]$$

$\frac{M_1}{M_2}$	η percent	Ratio of volume for laminar flow to volume for turbulent flow	
		$s_2/s_1 = 1.00$	$s_2/s_1 = 0.50$
1	50	1.44	1.15
	60	1.18	.94
	70	.93	.74
2	50	1.61	1.28
	60	1.40	1.11
	70	1.20	.95
	80	1.02	.81
4	50	1.67	1.33
	60	1.48	1.18
	70	1.31	1.04
	80	1.13	.90

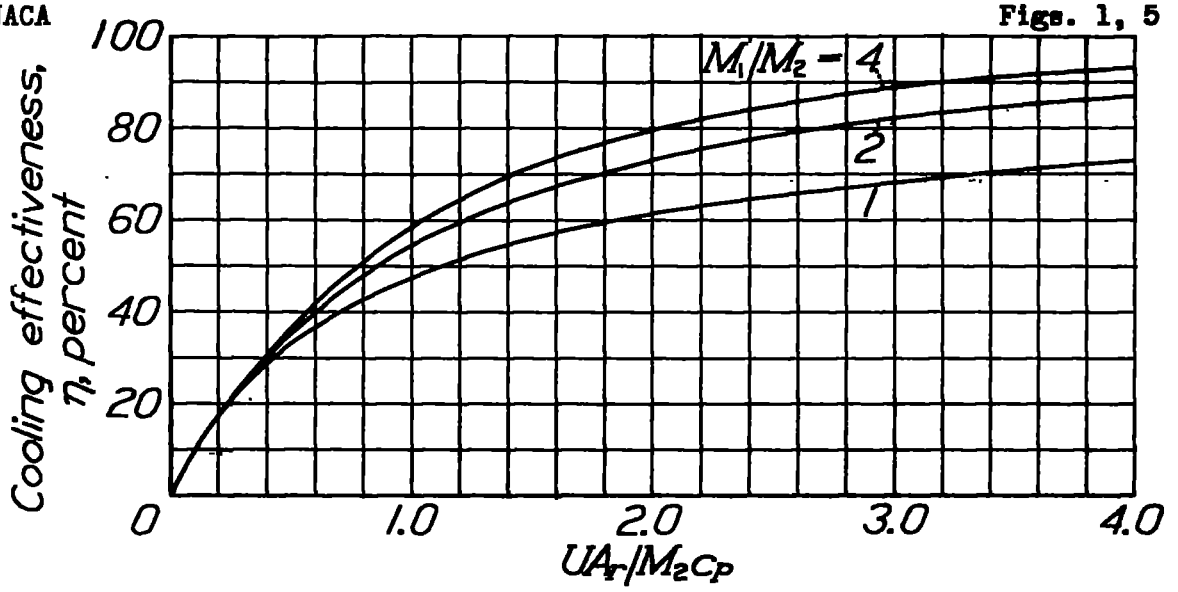


Figure 1.- Cooling effectiveness for cross-flow intercoolers. (Data from reference 2.)

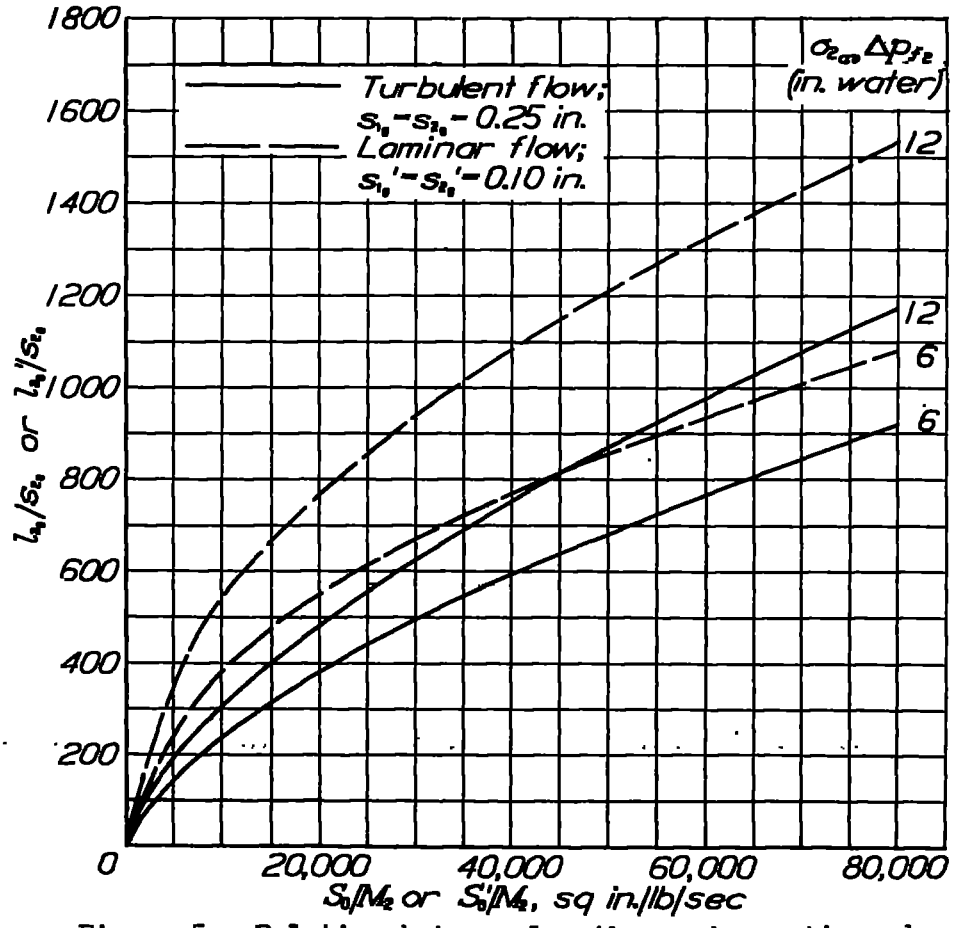
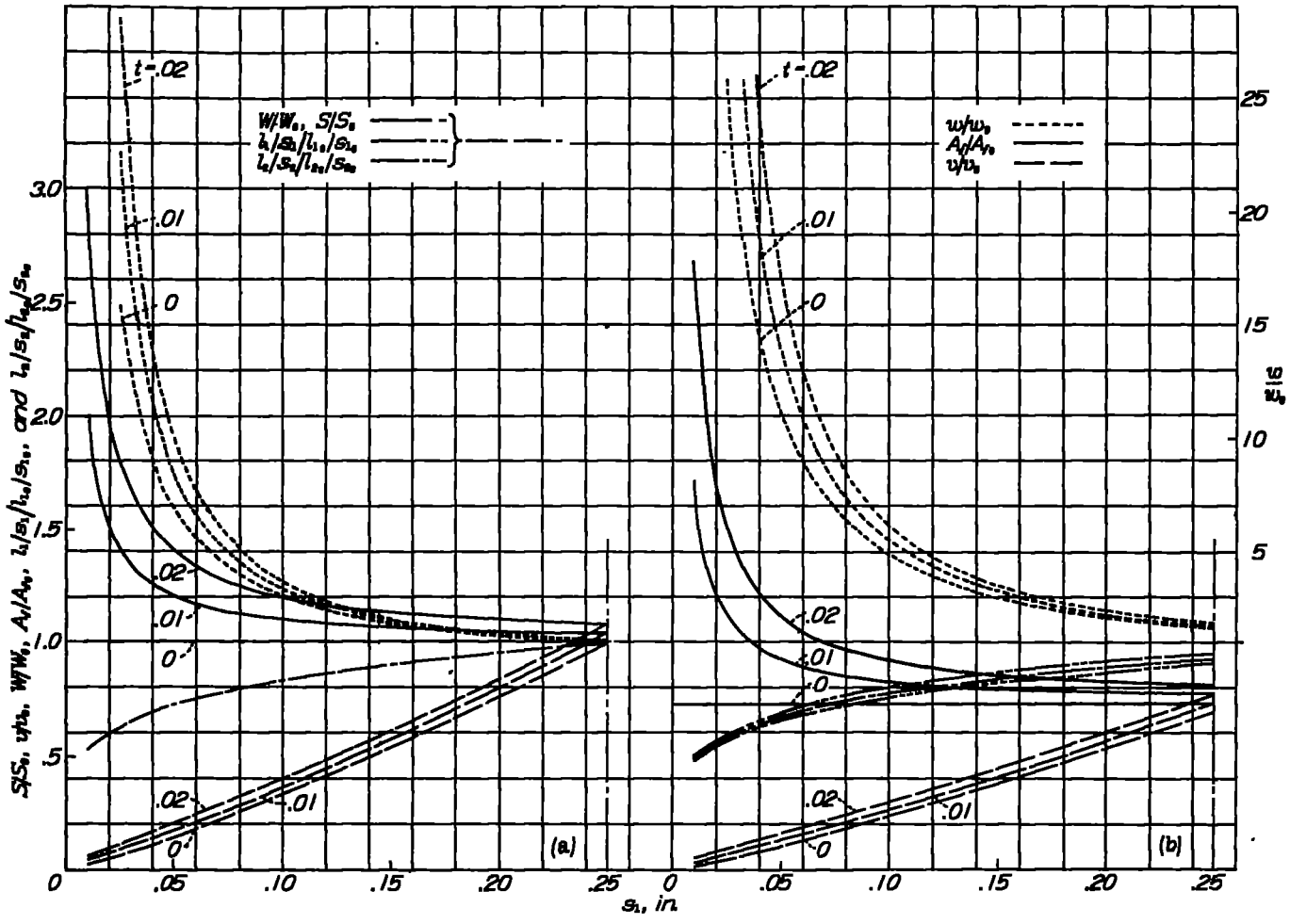


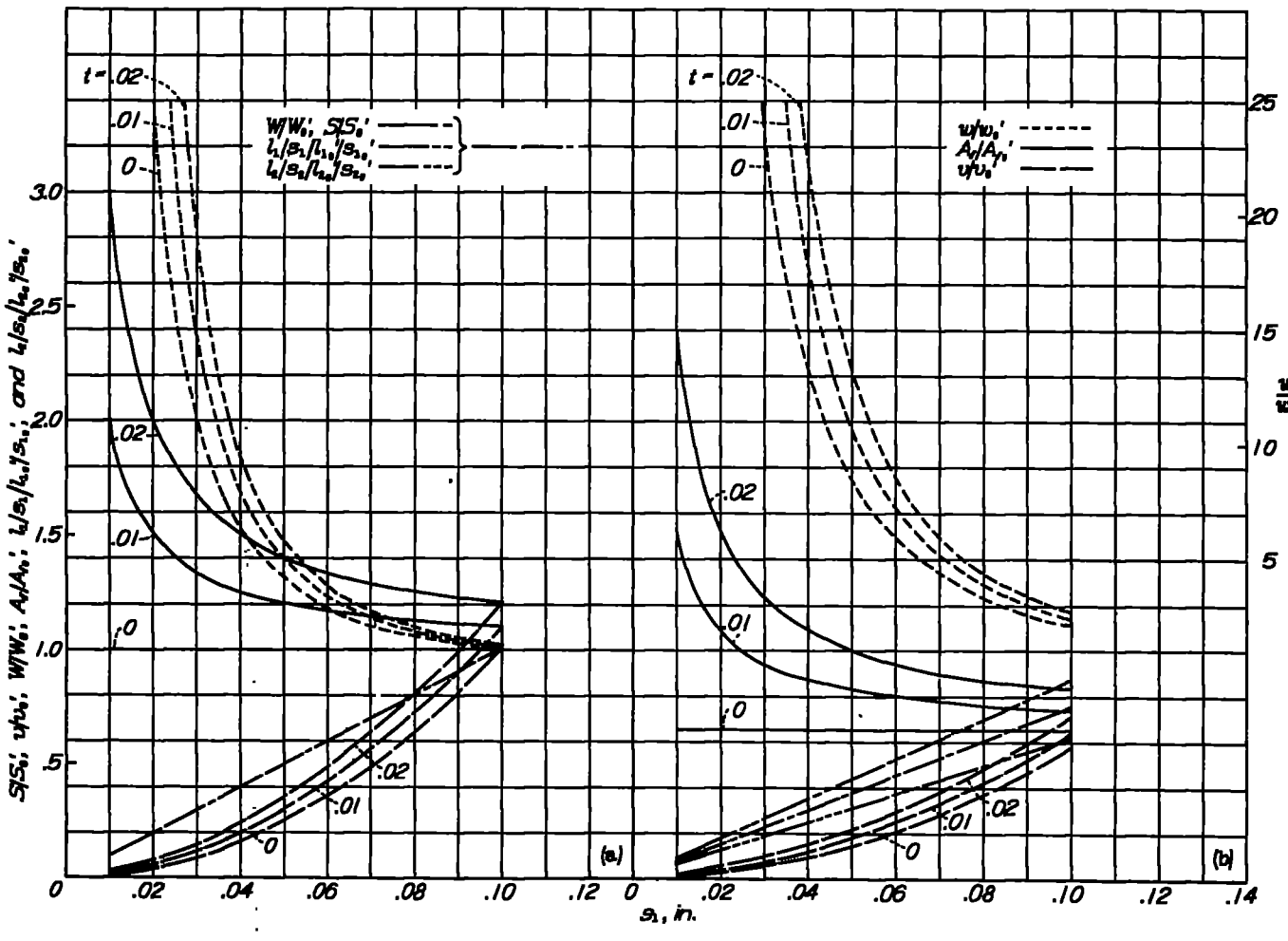
Figure 5.- Relation between length-spacing ratio and plate area for a reference intercooler.



(Subscript 0 refers to a reference intercooler which is defined as one having equal values of s_1 and s_2, $t = 0$, and no plate separators. $s_{1_0} = s_{2_0} = 0.25$ in.)

(a) $s_2/s_1 = 1.00$ (b) $s_2/s_1 = 0.50$

Figure 2.- Variation of plate-type intercooler weight and dimensional characteristics with plate spacing for constant operating conditions (η, M_1, M_2, $\sigma_{1_{av}}\Delta p_{f_1}$, $\sigma_{2_{av}}\Delta p_{f_2}$). Turbulent flow.



(a) $s_2/s_1 = 1.00$

(b) $s_2/s_1 = 0.50$

(Subscript 0 with a primed symbol refers to a reference intercooler which is defined as one having equal values of s_1 and s_2, $t = 0$, and no plate separators.

$s_{1_0}' = s_{2_0}' =$ 0.10 in.)

Figure 3.- Variation of plate-type intercooler weight and dimensional characteristics with plate spacing for constant operating conditions (η, M_1, M_2, $\sigma_{1_{av}}\Delta p_{f_1}$, $\sigma_{2_{av}}\Delta p_{f_2}$). Laminar flow.

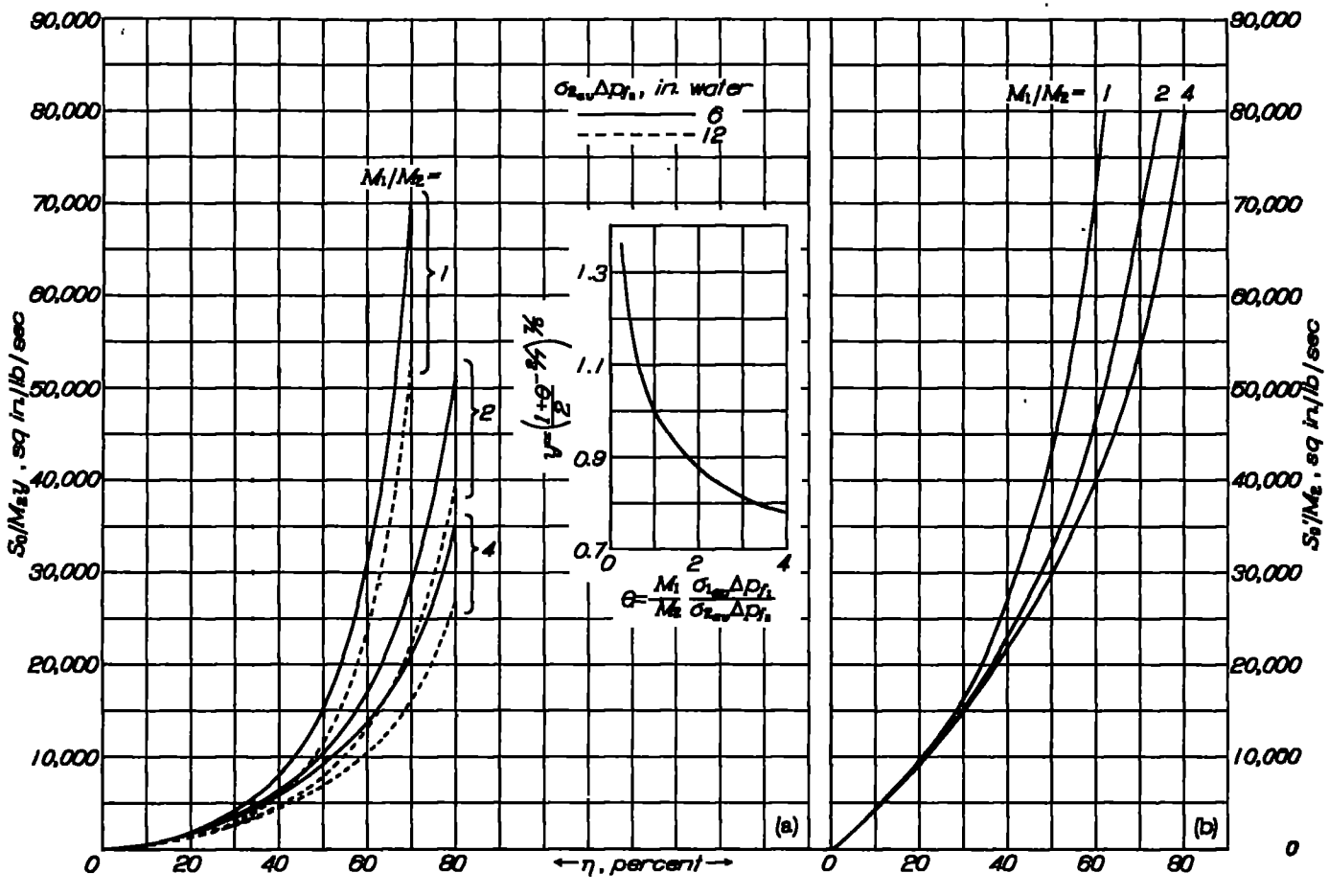


Figure 4.- Variation of reference-plate area with intercooler operating conditions.

(a) Turbulent flow; $s_{1_0} = s_{2_0} = 0.25$ in.

(b) Laminar flow; $s_{1_0} = s_{2_0}' = 0.10$ in.

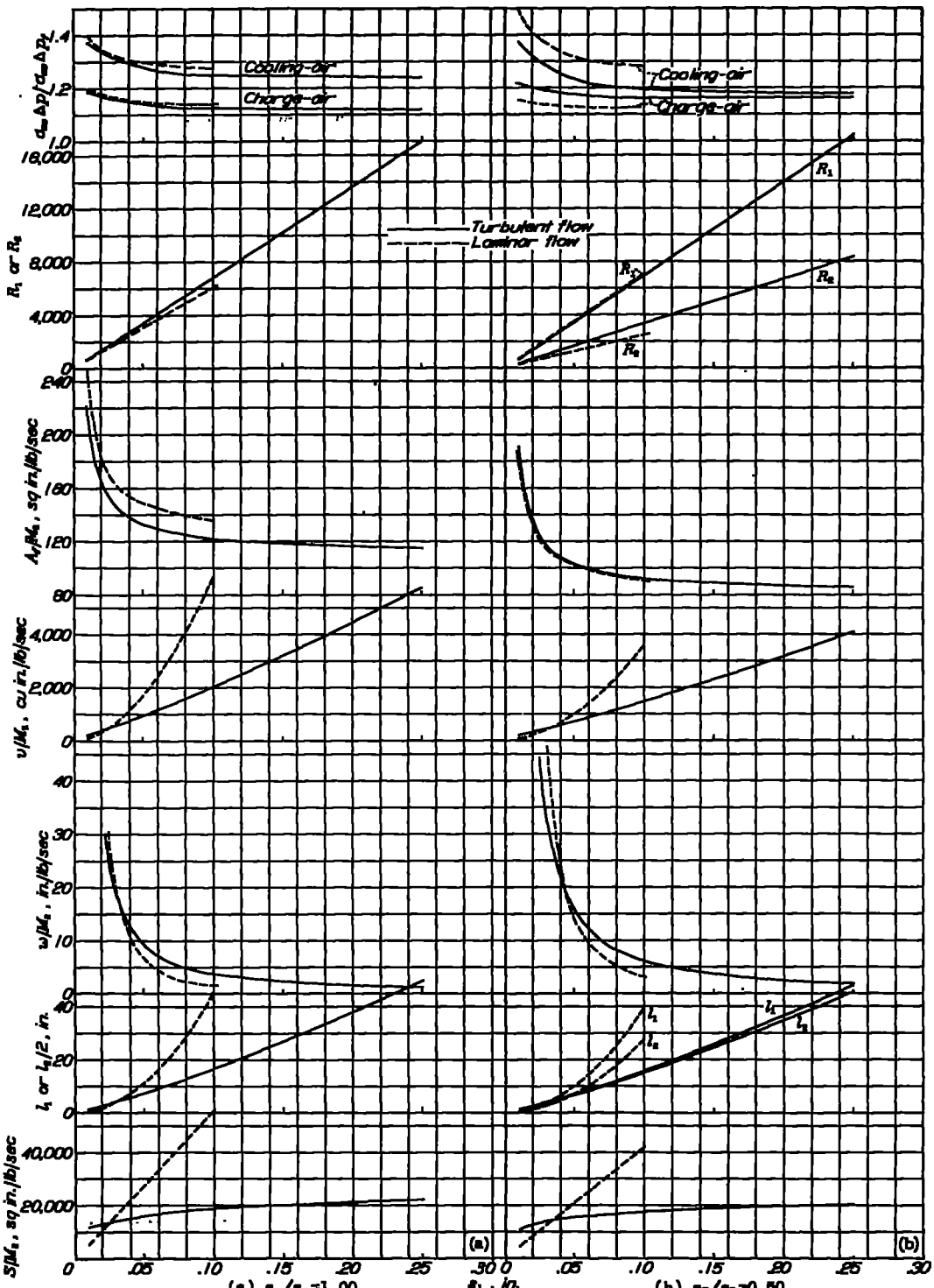


Figure 6.- Comparison of intercooler characteristics at a given operating condition for turbulent and for laminar flow, effect of plate separators not included. M_1/M_2=2; η=65 percent; $\sigma_{1_{av}}\Delta p_{f_1}$=3 inches of water; $\sigma_{2_{av}}\Delta p_{f_2}$=6 inches of water; t=0.01 inch.

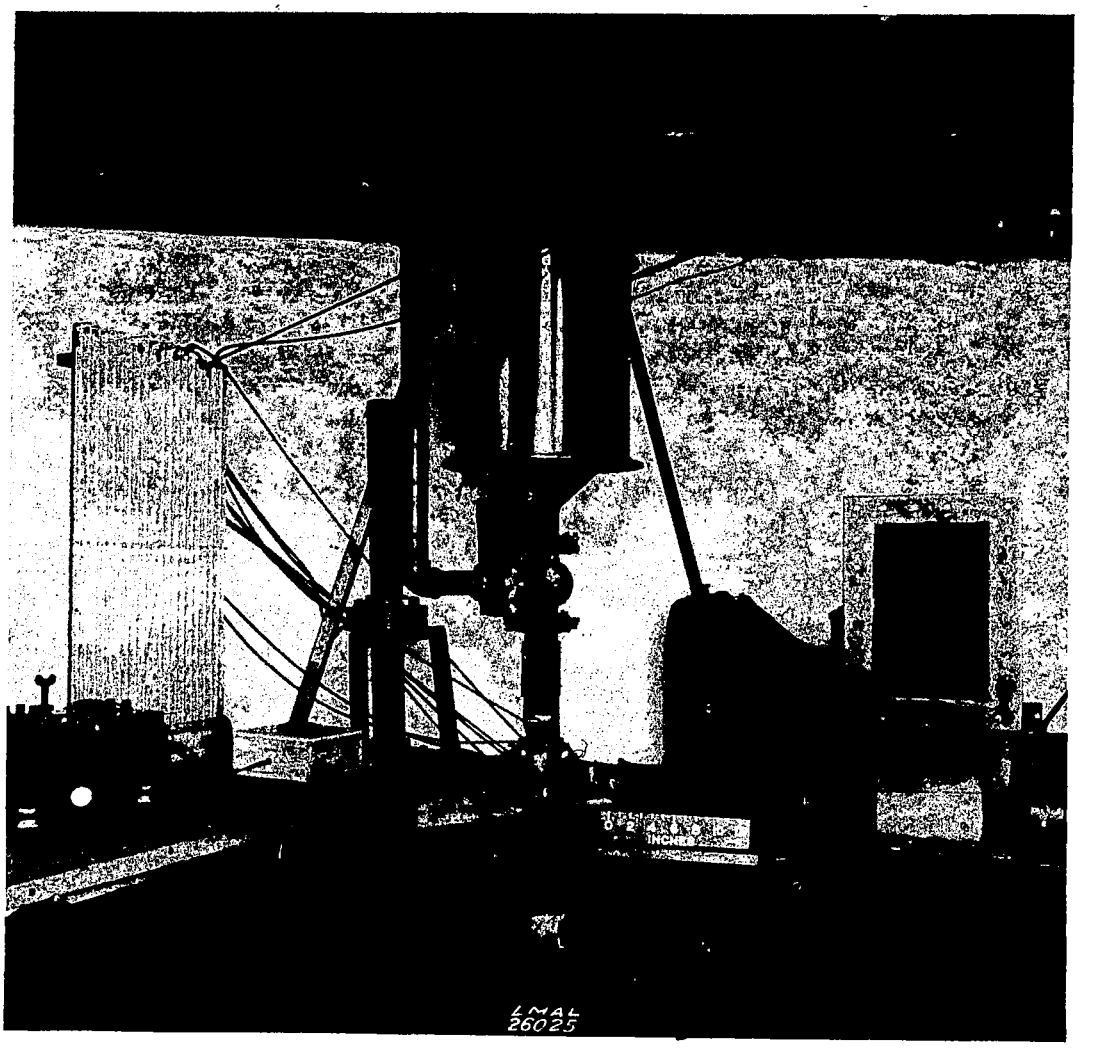


Figure 7.- Test assembly.

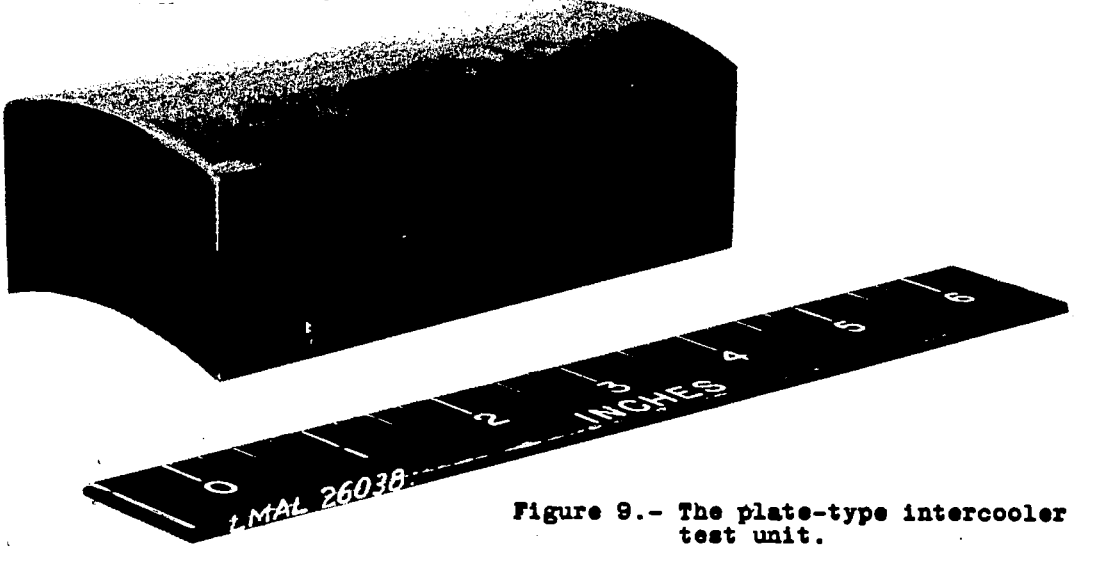


Figure 9.- The plate-type intercooler test unit.

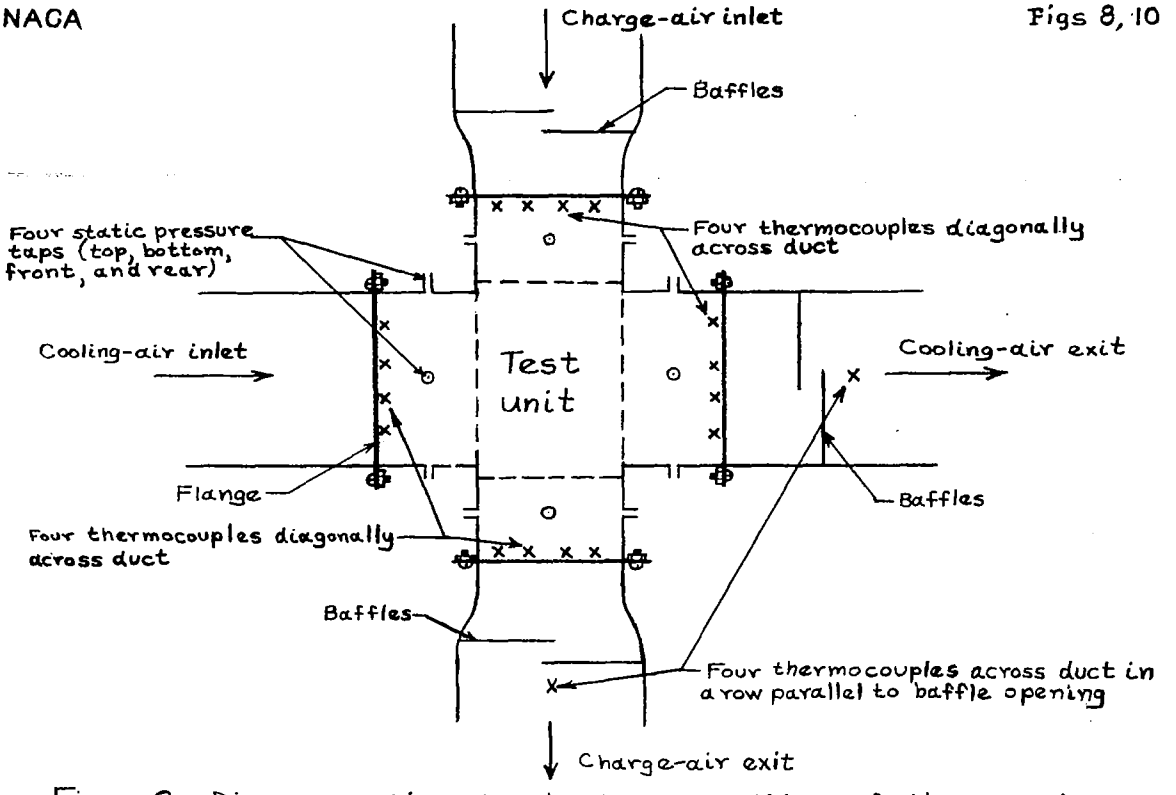


Figure 8.- Diagrammatic sketch showing positions of thermocouples, pressure taps, and baffle plates in intercooler duct.

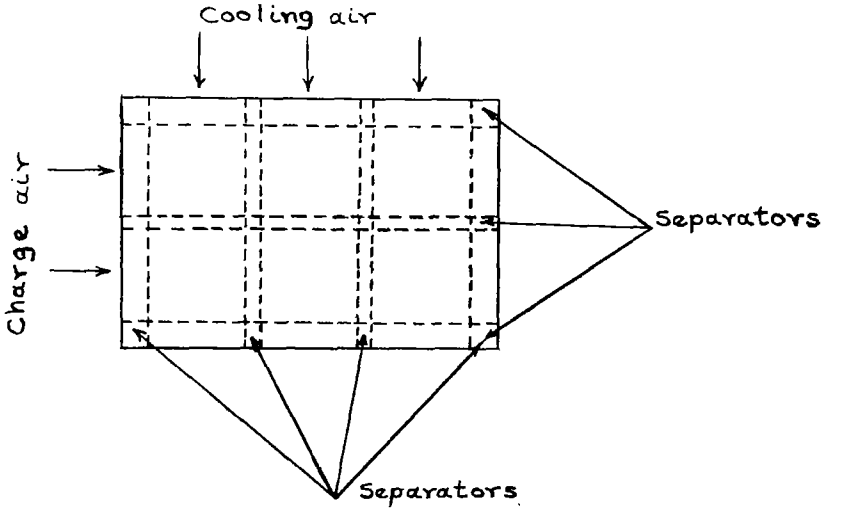
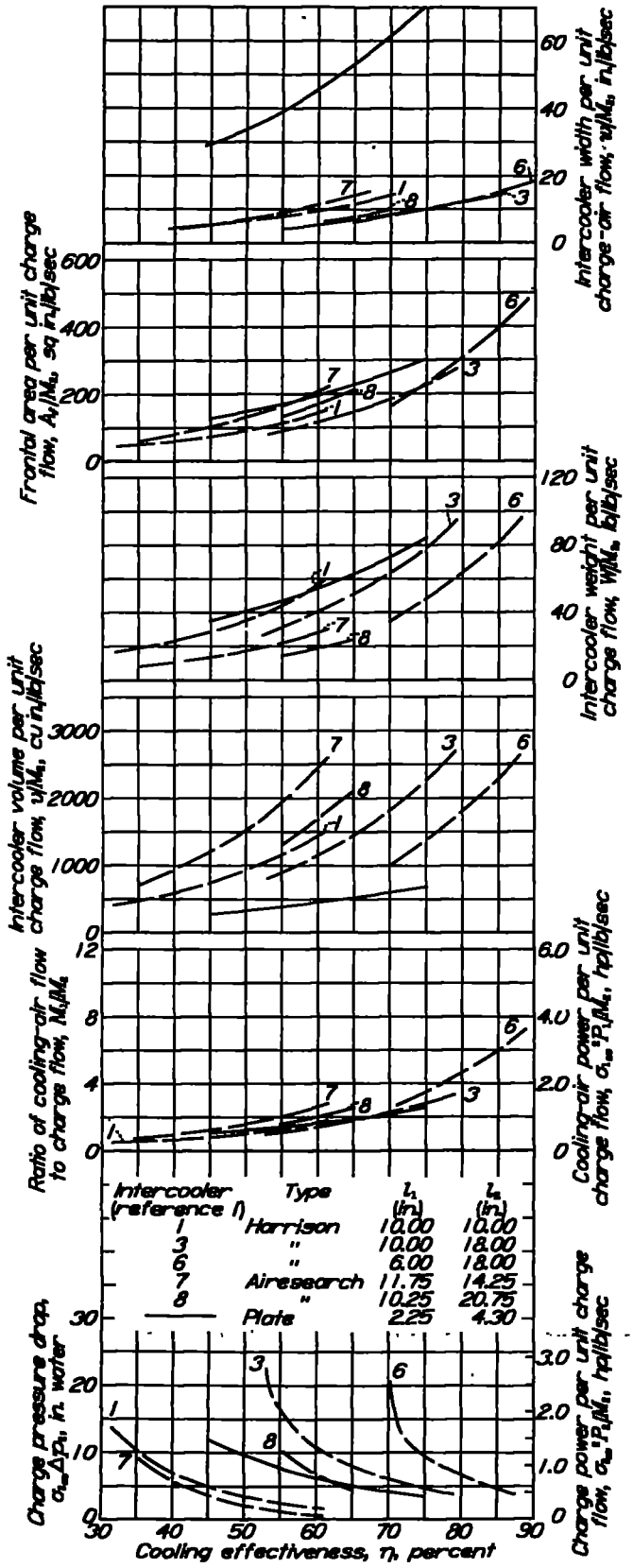


Figure 10. - Top view of intercooler test unit showing separator arrangement for charge-air and cooling-air passages.



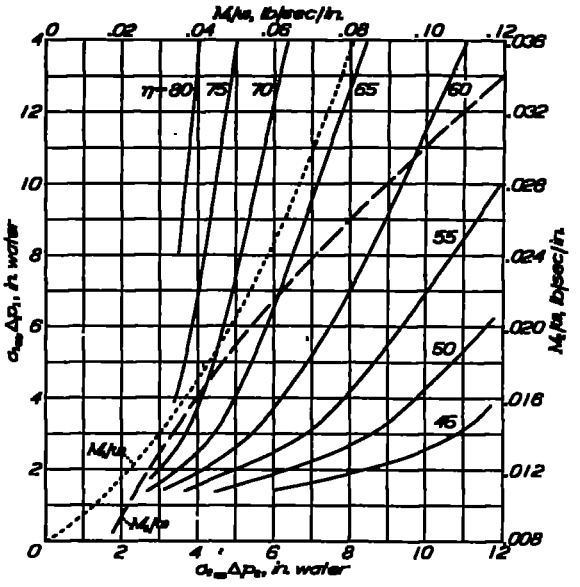


Figure 11.- Performance chart for the plate-type intercooler test unit. W/w, 1.20 pounds/inch; l_1, 2.25 inch; l_2 = 4.30 inches.

Figure 12.- Comparison between plate-type intercooler test unit and commercial intercoolers; $\sigma_{1av}\Delta p_1$, 4 inches of water.

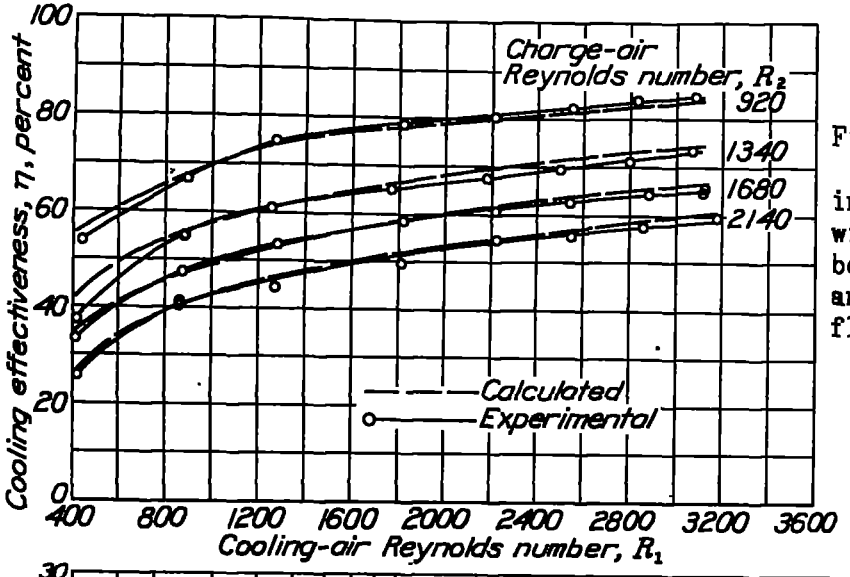


Figure 13.-Variation of cooling effectiveness with Reynolds numbers of charge-air and cooling-air flows.

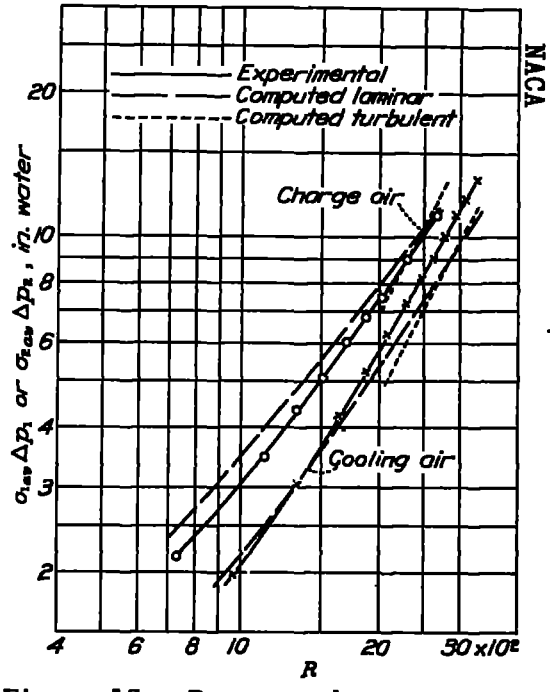


Figure 15.- Pressure drops across the plate type intercooler test unit.

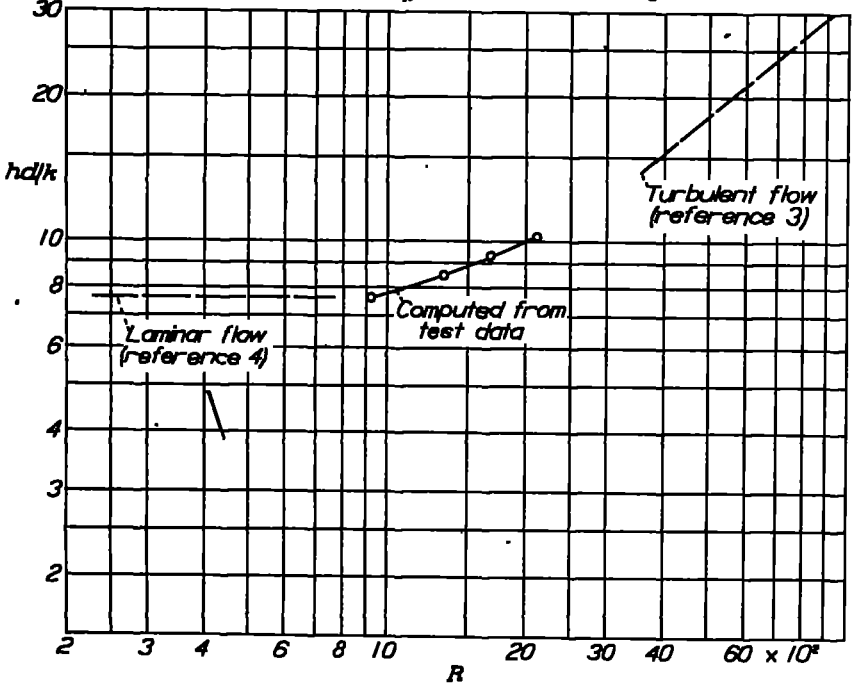


Figure 14.- Heat transfer in the plate type intercooler test unit.

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Figs. 13, 14, 15

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