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COMPARISON OF INTERCOOLER CHARACTERISTICS

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ADVANCE RESTRICTED REPORT*

COMPARISON OF INTERCOOLER CHARACTERISTICS

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SUMMARY

A method is presented of comparing the performance, weight, and general dimensional characteristics of intercoolers. The performance and dimensional characteristics covered in the comparisons are cooling effectiveness, pressure drops and weight flows of the charge and cooling air, power losses, volume, frontal area, and width.

A method of presenting intercooler data is described in which two types of charts are plotted: (1) A performance chart setting forth all the important characteristics of a given intercooler and (2) a replot of these characteristics for a number of intercoolers intended to assist in making a selection to satisfy a given set of installation conditions. The characteristics of commercial intercoolers obtained from manufacturers' data and of some computed designs are presented on this basis.

A standard test procedure and instrumentation are suggested whereby comparable data may be obtained by different testing organizations.

INTRODUCTION

The temperature drop in the charge provided by any given intercooler depends on the temperatures, the densities, and the pressure drops of the charge and cooling air. Because of the large number of possible combinations of intercooler sizes and operating conditions, a concise method of presenting intercooler data is required. A method of presenting intercooler data is suggested in this report in which by the use of the proper factors

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the same performance charts apply regardless of operating conditions or intercooler width (that is, the dimension at right angles to the directions of charge and cooling-air flows), provided the other intercooler dimensions are held constant. Intercooler manufacturers usually select for production a few of an unlimited number of combinations of intercooler dimensions and internal arrangements, leaving the intercooler width to the choice of the customer. Thus, a few charts will present the performance of the entire stock of intercoolers listed by any manufacturer. The performance charts for all the commercial intercoolers on which data could be obtained are given in this report.

The characteristics of importance in selecting an intercooler to provide a given cooling effectiveness (drop in charge temperature per degree difference between inlet temperatures) for a given engine and flight condition are: the pressure drop of the charge, the weight flow of the cooling air, the intercooler weight, the volume and core dimensions, and the associated power losses. Changes in certain intercooler dimensions may improve some of these characteristics to the detriment of others. All factors must be held within limits which vary with different installations. Thus, intercoolers cannot be compared on the basis of a single characteristic but rather on the basis of curves setting forth the various characteristics. Based on the performance charts previously mentioned, charts are presented on which the important dimensions and performance characteristics of the various commercial intercoolers are compared. These comparison charts provide a means of quickly determining the intercooler having the characteristics best suited for a given installation. A comparison between the commercial intercoolers and several theoretical intercoolers of the tubular type is also shown.

In order to insure a fair comparison of intercoolers of various manufacturers, a standard test procedure is suggested.

SYMBOLS

M rate of air flow, pounds per second

l length in the direction of air flow, inches

d tube diameter, inches

s clearance between tubes perpendicular to flow across tube, inches

s_p tube pitch in direction of flow across tube banks, inches

m number of tube banks in direction of flow across tube

W intercooler weight, pounds

v intercooler volume, cubic inches

A_1 core face area at right angles to cooling-air flow (frontal area), square inches

w intercooler width (core dimension perpendicular to both charge and cooling-air flows), inches

H heat-transfer rate, Btu per second

h heat-transfer coefficient per unit width based on the area of the intercooler section perpendicular to the intercooler width, Btu per second per square inch per °F per inch of intercooler width

Δp pressure drop of air across intercooler, inches of water

σ density of air relative to standard atmosphere

ρ_o standard atmospheric density (0.0765 lb per cu ft)

P power, horsepower

T_o cooling-air temperature at intercooler entrance, °F

$T_{1_{ex}}$ temperature rise of cooling air, °F

T_2 temperature difference between charge and cooling air at intercooler entrances, °F

$T_{2_{ex}}$ temperature difference between charge at exit and cooling air at entrance, °F

η cooling effectiveness, $\dfrac{T_2 - T_{2_{ex}}}{T_2}$

Subscripts $_1$ and $_2$ refer to cooling air and charge, respectively, and subscripts en and av, to entrance and average conditions.

ANALYSIS

A "specific" intercooler is defined as one in which the internal structure and the flow passage lengths are fixed and only the dimension w is variable. An examination of equations (16), (17), and (18) (see appendix) shows that the cooling efficiency of a given "specific" intercooler is a function only of h, M_1/M_2, and M_1/w. But h is a function of M_1/M_2 and M_1/w; therefore symbolically

$$\eta = f\left(\frac{M_1}{M_2}, \frac{M_1}{w}\right) \tag{1}$$

Furthermore,

$$\frac{M_1}{w} = K_1(\sigma_{1_{av}}\ \Delta p_1)^{n_1} \tag{2}$$

and

$$\frac{M_2}{w} = K_2(\sigma_{2_{av}}\ \Delta p_2)^{n_2} \tag{3}$$

where K_1, K_2, n_1, and n_2 are constants.

Therefore,

$$\eta = f_1(\sigma_{1_{av}}\ \Delta p_1,\ \sigma_{2_{av}}\ \Delta p_2) \tag{4}$$

Thus, for a given value of $\sigma_{1_{av}}\Delta p_1$ $\left(\text{or } \frac{M_1}{w}\right)$,

$$\eta = f_2(\sigma_{2_{av}}\ \Delta p_2) \tag{5}$$

or

$$\eta = f_3(M_1/M_2) \tag{6}$$

therefore the relations of η to $\sigma_{2_{av}} \Delta p_2$ and to M_1/M_2, for a specific intercooler, can each be expressed by a single curve for a given value of $\sigma_{1_{av}} \Delta p_1$.

The cooling performance of a specific intercooler can be completely described by curves of η plotted against $\sigma_{1_{av}} \Delta p_1$ and $\sigma_{2_{av}} \Delta p_2$ (equation (4)), M_1/w against $\sigma_{1_{av}} \Delta p_1$ (equation (2)), and M_2/w against $\sigma_{2_{av}} \Delta p_2$ (equation (3)). The performance data may be obtained by testing a single intercooler, and, when plotted in this form, the data apply for all widths of this specific intercooler. The curves are also general with regard to charge and cooling air density and temperature.

Other characteristics of the specific intercooler may be graphically expressed in terms of $\sigma_{2_{av}} \Delta p_2$ and η for a given value of $\sigma_{1_{av}} \Delta p_1$ as follows:

The intercooler volume per unit charge flow is

$$v/M_2 = l_1 \; l_2 \frac{w}{M_2} \tag{7}$$

By equation (3) w/M_2, the intercooler width per unit charge flow, may be expressed in terms of $\sigma_{2_{av}} \Delta p_2$ and by equation (5) in terms of η. Likewise, the intercooler weight per unit charge flow

$$\frac{W}{M_2} = \frac{W}{w} \frac{w}{M_2} \tag{8}$$

is a function of η. This is true also for the frontal area per unit charge flow:

$$\frac{A_1}{M_2} = \frac{l_2 w}{M_2} \tag{9}$$

The horsepower required to force the cooling air through the intercooler per unit charge flow is given by

$$\frac{\sigma_{1_{av}}^{2} P_1}{M_2} = \frac{5.2 M_1}{M_2} \frac{\sigma_{1_{av}} \Delta p_1}{550 \; \rho_0} \tag{10}$$

in which M_1/M_2 may be expressed in terms of η by means of equation (6). If the cooling air is discharged from the intercooler into a compartment in which the air velocity is practically zero, the total-head loss is

$$\sigma_{1_{av}} \Delta p_{1_T} = \sigma_{1_{av}} \Delta p_1 + \frac{1}{10.4}\left(\frac{M_1}{w}\right)^2 \left(\frac{1}{g\rho_0 l_2^{\,2}}\right)$$

and the associated power loss is given by

$$\frac{\sigma_{1_{av}}^{\,2} P_{1_T}}{M_2} = \frac{M_1}{M_2} \; \frac{5.2\sigma_{1_{av}} \Delta p_1 + \frac{1}{2}\left(\frac{M_1}{w}\right)^2 \left(\frac{1}{g\rho_0 l_2^{\,2}}\right)}{550\ \rho_0} \tag{11}$$

The total-head loss Δp_{1_T} and the associated power loss P_{1_T}, if desired, can be computed and included in the figures where these values occur. The power loss given by equation (11) does not strictly represent the airplane drag horsepower due to the intercooler since no account is taken of the Meredith effect which depends on flight conditions.

The horsepower required to force the charge through the intercooler per unit charge flow is given by

$$\frac{\sigma_{2_{av}}^{\,2} P_2}{M_2} = \frac{5.2\sigma_{2_{av}} \Delta p_2}{550\ \rho_0} \tag{12}$$

and is also a function of η by equation (5).

The relative densities (σ) in the foregoing equations are average densities. The average densities can be calculated from the entrance densities by means of the following relations:

$$\sigma_{1_{av}} = \sigma_{1_{en}}\left(\frac{2 + \beta_1}{2 + 2\beta_1}\right) \tag{13}$$

where $$\beta_1 = \frac{T_{1_{cx}}}{T_0 + 460} = \frac{(M_2/M_1)\ \eta\ T_2}{T_0 + 460}$$

$$\sigma_{2_{av}} = \sigma_{2_{en}} \left(\frac{2 - \beta_2}{2 - 2\beta_2} \right) \tag{14}$$

where

$$\beta_2 = \frac{\eta T_2}{T_2 + T_0 + 460}$$

Equations (13) and (14) are shown graphically in figure 1. The effect of pressure change through the intercooler has been neglected in these equations because this effect is usually small. Where large pressure drops occur, the density may be corrected for change in pressure by means of the relation that density varies directly with absolute pressure.

TEST ASSEMBLY AND METHOD

For the purpose of standardizing intercooler testing, the following test assembly and procedure are suggested:

The equipment necessary for testing intercoolers should consist essentially of a variable-speed motor-driven blower, an air heater, two air-flow meters, manometers, and thermocouples with auxiliary instruments. Air-flow measurements may be made by means of orifice plates placed in the air streams in accordance with the procedure outlined by the American Society of Mechanical Engineers (reference 1). The charge should be supplied to the intercooler at a temperature considerably higher (say 200° F) than that of the cooling air in order to promote accuracy in the measurement of cooling effectiveness. The width of the unit tested will be governed by the capacity of the blower and the heating capacity of the air heater. Static-pressure tubes should be installed near exits and entrances of the intercooler block for the measurement of pressure drop in the charge and cooling air across the intercooler. The entrance and exit ducts should be of uniform cross section between the intercooler and the static-tube positions. The static tubes at the exits should be placed at a sufficient distance downstream from the block in order to permit the flow to adjust itself to the change in flow area. A distance of 8 or 10 diameters (tubular type) or plate spaces (plate type) is probably safe. The static pressure taps should be located in positions where they will not be affected by velocity head and should have entrance sections free from surface irregularities. Tubes for measuring total head should be placed at the same distances from the intercooler block as the static pressure taps.

Thermocouples placed at various positions across each entrance and exit duct provide a means of determining inlet and outlet temperatures. There should be probably one thermocouple for each 4 or 5 square inches of flow section across which the thermocouples are distributed. The air stream should be thermally insulated from the surrounding atmosphere between the thermocouple positions and the test unit. The thermocouples across any section may be connected in series in the cold-junction box; the total voltage of the group, when divided by the number of thermocouples in the group should yield a satisfactory mean-temperature indication. Baffles should be placed upstream from all the thermocouples in order to insure uniform temperature distribution. Greater accuracy will be assured if the thermocouples are provided with shields to eliminate radiation effects.

A number of series of tests should be made; for a given series one weight flow should be held constant while the other weight flow is varied. The values of M_1/w and M_2/w should be chosen to cover the useful range of the intercooler. The necessary data to be used in determining the performance of an intercooler are:

1. Weight flow of cooling air per unit width
2. Weight flow of charge per unit width
3. Outlet and inlet cooling-air temperature
4. Outlet and inlet charge temperature
5. Outlet and inlet charge static pressure
6. Outlet and inlet cooling-air static pressure
7. Outlet and inlet charge total head
8. Outlet and inlet cooling-air total head

The following information should be included with the performance data:

9. Core weight per unit core width
10. Complete intercooler weight per unit core width
11. A sketch showing all internal and external dimensions

DISCUSSION

Types of Flow

Figure 2 gives the cooling effectiveness of cross flow, counterflow, and parallel flow as a function of the weight flow of charge and cooling air, the over-all heat-transfer coefficient, and the intercooler surface. The curves are a plot of equations (16), (17), and (18) of the appendix. In the cases of parallel flow and counterflow l_1 is the dimension parallel to the direction of air flow and l_2 is the dimension at right angles to l_1 in the plane across which heat is transferred. Figure 2 shows that on the basis of cooling effectiveness counterflow is slightly better than cross flow and greatly superior to parallel flow. Because of practical difficulties encountered in arranging efficient counterflow passages, no commercial intercoolers of this type have yet appeared.

Figure 2 indicates that the cooling surface of an intercooler may be of such magnitude that any additional surface would increase only slightly the cooling effectiveness because of the flattening tendency of the curves. This point should be considered in design because of the increase in pressure drop, weight, and volume with cooling surface. It may be noted further that the flattening tendency is more abrupt and occurs at lower values of the abscissa $\frac{wl_1l_2h}{M_1c_p}$ when the weight-flow ratio M_1/M_2 is large.

Performance Charts for "Specific" Intercoolers

The method of plotting performance data for specific intercoolers suggested in the analysis is followed in figures 3, 4, 5, and 6, in which $\sigma_{1_{av}} \Delta p_1$ is plotted against $\sigma_{2_{av}} \Delta p_2$ for a number of values of η. The figures also include a plot of $\sigma_{1_{av}} \Delta p_1$ against M_1/w and of $\sigma_{2_{av}} \Delta p_2$ against M_2/w. The external dimensions and the weight per unit width of the intercooler core are given in each of the figures. As previously pointed out in the analysis, the data in each figure apply to an intercooler of any width w, provided all other dimensions remain at fixed values. It is evident that the curves for each specific intercooler may be obtained by testing only one intercooler unit of any convenient width.

Figure 3 gives the data on all the Harrison intercoolers and figure 4 gives the data on all the Airesearch intercoolers on which information could be found. The Harrison intercoolers are of copper construction, whereas the Airesearch intercoolers are of aluminum construction. The use of aluminum instead of copper in the construction of the Harrison intercoolers would not affect their performance and would reduce their weight by the ratio of the density of aluminum to the density of copper. Figure 5 is an illustrative performance chart for a specific charge-through-tube intercooler calculated from reference 3. Figure 6 is a similar chart for two specific charge-across-tube intercoolers calculated from reference 2. Figures 3 and 4 were plotted from experimental data furnished by the manufacturers (reference 4), and no attempt has been made at this laboratory to check the accuracy of these data.

The lines with arrows in figure 3(a) show the method by which the chart may be used. 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 $\sigma_{2_{av}}\,\Delta p_2$ a corresponding value of M_2/w may be read at the right, using the long-dash curve. The relation of $\sigma_{1_{av}}\,\Delta p_1$ to M_1/w is given by the short-dash curve, the value of M_1/w being read at the top. Performance of any altitude is obtained by assigning the proper values to $\sigma_{1_{av}}$ and $\sigma_{2_{av}}$. An example of the use of figure 3(a) follows:

Assumptions:

1. Altitude, ft 13,200
2. Brake horsepower 1,000
3. Weight flow or charge, lb per sec 1.74
4. Cooling-air pressure drop Δp_1, in. of water 9
5. Charge inlet temperature, $^{\circ}F$ 200
6. Charge inlet pressure, in. Hg 40
7. Cooling-air inlet temperature (standard altitude), $^{\circ}F$ 12
8. Required charge outlet temperature, $^{\circ}F$ 78

Let it be required to find the values of cooling effectiveness η, charge pressure drop Δp_2, intercooler width w, cooling-air weight flow M_1, and intercooler weight W.

9. From items 5, 7, and 8, $\eta = \frac{200 - 78}{200 - 12} = 0.65$

10. From a table of standard altitude and item 4,
$\sigma_{1_{en}} = 0.667$ for 13,200 feet; therefore
$\sigma_{1_{en}} \Delta p_1 = 6$ inches of water

In item 10, $\sigma_{1_{en}}$ is used as a first approximation. The value of $\sigma_{1_{av}}$ can be found after M_1/M_2 has been determined.

11. From figure 3(a) and item 10, $M_1/w = 0.155$ pound per second per inch (first approximation)

12. From figure 3(a) and items 9 and 10, $M_2/w = 0.07$ pound per second per inch (first approximation)

13. From items 11 and 12, $M_1/M_2 = \frac{0.155}{0.07} = 2.21$ (first approximation)

14. From figure 1(a) and items 5, 7, 9, 10, and 13,
$\sigma_{1_{av}} = 0.667 \times 0.95 = 0.632$

15. From items 4 and 14, $\sigma_{1_{av}} \Delta p_1 = 9 \times 0.632 = 5.70$ inches of water

16. From figure 3(a) and item 15, $M_1/w = 0.150$ pound per second per inch

17. From figure 3(a) and items 9 and 15, $M_2/w = 0.068$ pound per second per inch; therefore, from item 16, $M_1/M_2 = \frac{0.150}{0.068} = 2.21$, which checks item 13, thus indicating that the first approximation from M_1/M_2 is correct. Because $\sigma_{1_{av}}/\sigma_{1_{en}}$ changes only slightly with change in M_1/M_2, a second approximation for $\sigma_{1_{av}}/\sigma_{1_{en}}$ is not required even if some difference between the values of M_1/M_2 in items 13 and 17 occurs

18. From figure 3(a) and items 9 and 15, $\sigma_{2_{av}} \Delta p_2 = 1.83$ inches of water

19. From items 5 and 6, $\sigma_{2_{en}} = \frac{40}{30} \times \frac{519}{660} = 1.05$; therefore from figure 1(b) and items 5, 7, and 9 $\sigma_{2_{av}} = 1.05 \times 1.11 = 1.17$

20. From items 18 and 19, $\Delta p_2 = 1.83/1.17 = 1.57$ inches of water

21. From items 3 and 17, $w = 1.74/0.068 = 25.5$ inches

22. From items 16 and 21, $M_1 = 0.150 \times 25.5 = 3.84$ pounds per second

23. From figure 3(a) and item 21, $W = 4 \times 25.5 = 102$ pounds

Comparison of Intercoolers

For the purpose of comparing a number of intercoolers on the basis of their important characteristics, a method of plotting may be used as suggested in figure 7 where the cooling effectiveness is varied by varying w/M_2. Each curve in this figure represents a specific intercooler. The data for figure 7 are calculated from the individual performance charts for intercoolers 1 to 8 (figs. 3 and 4) and by means of equations (7), (8), (9), (10), and (12). Intercoolers 25, 26, and 27, for which the performance charts are given (figs. 3(g), 3(h), and 3(i)), are not included in the comparison chart (fig. 7). All intercoolers are identified by numbers, and this identity is maintained in all figures and throughout the report.

In figure 7 a comparison is made of six Harrison and two Airesearch intercoolers of various dimensions and, where possible, at three values of $\sigma_{1_{av}} \Delta p_1$. The complete intercooler weight values are given for these intercoolers, and the dimensions are those of the core.

It may be seen in figure 7 that $\sigma_{1_{av}} \Delta p_1$ has little effect on the comparison of these intercoolers; so the comparison can be made at any value of $\sigma_{1_{av}} \Delta p_1$. Examination further reveals that for a given cooling effectiveness the Airesearch aluminum intercoolers are lighter but larger in volume and frontal area than the Harrison copper intercoolers.

An examination of the curves for the Harrison intercoolers in figure 7 shows that, for the same cooling effectiveness and cooling-air pressure drop, lengthening the intercooler in the direction of charge flow (l_2) results in a decrease in cooling-air flow, volume, weight, and frontal area at the expense of a marked increase in charge pressure drop. Increasing the cooling-air flow length (l_1), for a given cooling effectiveness and cooling-air pressure drop, decreases the cooling-air flow and charge pressure drop at the expense of increased weight and volume.

The selection of an intercooler depends on the relative importance of its various characteristics with regard to a particular installation. For example, figure 7 shows intercooler 6 to be desirable if low weight and volume are essential. This advantage is attained, however, at a sacrifice of increased pressure drop in the charge air. If a low value of $\sigma_{2_{av}} \Delta p_2$ is desired, weight and volume being of secondary importance, intercooler 1 is the logical choice. The choice between the Harrison intercooler and the Airesearch intercooler depends largely on the relative importance of weight, volume, and frontal area according to figure 7.

Figure 8 is a plot similar to figure 7 in which two of the Harrison intercoolers of figure 7 are compared with tubular intercoolers of the charge-across-tube type. The tubular intercoolers were designed from charts published in reference 2 with internal and external dimensions selected to match approximately the relations existing between $\sigma_{2_{av}} \Delta p_2$ and η for the two Harrison intercoolers. Thus, for given values of η, $\sigma_{2_{av}} \Delta p_2$, $\sigma_{1_{av}} \Delta p_1$, and M_2, the intercoolers are compared on the basis of the remaining characteristics. The tubular intercoolers of 30 banks represent the "block" type while those of five banks represent the annular-shape intercooler as described in reference 2. All internal and external dimensions of the tubular intercoolers are given in the table at the right of the figure or may be computed from it. The weight values given for these intercoolers are core weights only, and this point should be considered when making comparison with the commercial types where complete weights are given. The tube arrangement is such that the tube centers lie on the apexes of equilateral triangles. From the standpoint of volume, the equilateral tube arrangement does not give the optimum intercooler. A reduction of the intercooler dimension in the direction of charge flow and thus a reduction in volume can be made by decreasing within limits the tube spacing in this direction without material change in the other characteristics of the intercooler.

A study of intercoolers 16 to 24 reveals certain effects of tube length and spacing. The effect of increasing l_1 for a given value each of $\sigma_{1_{av}} \Delta p_1$, $\sigma_{2_{av}} \Delta p_2$, and η increases weight and volume but decreases frontal area, width, and cooling-air flow. Under given conditions of $\sigma_{1_{av}} \Delta p_1$ and η a decrease in tube spacing decreases all other variables except $\sigma_{2_{av}} \Delta p_2$, which is considerably increased. (See curves 16 and 19 of fig. 8.) Small spacing is therefore desirable if $\sigma_{2_{av}} \Delta p_2$ is not an important factor.

In figure 9 a comparison is made of the Harrison intercoolers 1 and 3 of figures 7 and 8, an Airesearch intercooler 7 of figure 7, and a number of charge-through-tube intercoolers designed from charts published in reference 3. As in figure 8, the chart designs were chosen to match approximately the relations of $\sigma_{2_{av}} \Delta p_2$ against η for the commercial intercoolers. Intercoolers 9 to 15 have their tube centers on the apexes of equilateral triangles. As previously pointed out, this arrangement is not optimum from the standpoint of volume. Intercoolers 28, 29, and 30 are identical with 9, 10, and 13, respectively, in all dimensions except that the tube spacing in the direction of cooling air flow s_p, and hence l_1, has been decreased. It is seen that this change causes a marked decrease in volume without change in the other intercooler characteristics. The effect of changes of s_p is discussed in greater detail in reference 3.

Comparison of curves 9, 10, and 11 shows the effect of varying l_2/d when other dimensions were altered to provide the same curve of $\sigma_{2_{av}} \Delta p_2$ against η for the three intercoolers. When l_2/d is increased under these conditions, there is an increase in weight and a decrease in cooling-air flow, volume, and width. To maintain the condition of constant $\sigma_{2_{av}} \Delta p_2$ as l_2/d is increased, the tube spacing s is decreased and l_1 therefore is reduced.

CONCLUDING REMARKS

The cross-flow intercooler is not the most efficient, although it is probably the most practical type.

The selection of an intercooler to satisfy a particular group of conditions is greatly simplified by the proper correlation of the test data furnished by the manufacturer, and it is believed that the methods suggested in this report should prove of material assistance in making such selection.

The final choice of any intercooler must of necessity be a compromise among conflicting factors, and the nature of the compromise will depend largely upon the relative importance assigned to these factors.

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APPENDIX

Figure 10 represents a section of an intercooler of unit width perpendicular to the dimension w through which heat is being transferred from charge to cooling air, the direction of the air streams being at right angles to each other (cross flow). The heat flowing through an elemental area dx dy of this section at distances y and x from the charge and cooling-air entrances, respectively, is

$$\frac{dH}{w} = h_2\, dx\, dy(T_{2_y} - T_w) = h_1\, dx\, dy(T_w - T_{1_x}) = \frac{dy}{l_2}\frac{M_1}{w}\, c_p\, dT_{1_x}$$

Eliminating T_w and separating variables,

$$\frac{dT_{1_x}}{T_{2_y} - T_{1_x}} = \frac{w\, l_2\, h_2\, h_1\, dx}{M_1\, c_p\, (h_1 + h_2)} = \frac{w\, l_2\, h}{M_1\, c_p}\, dx$$

where h is the over-all heat-transfer coefficient per unit intercooler width based on the area $l_1\ l_2$. Integrating between limits $T_{1_{ex}}$ to 0 and l_1 to 0 along the elemental strip in the direction of the cooling-air flow and regarding T_{2_y} as constant along the strip

$$T_{1_{ex}} = T_{2_y}\left(1 - e^{-\frac{w l_1 l_2 h}{M_1 c_p}}\right) \tag{15}$$

From figure 10 it may be seen that

$$-\frac{M_2}{w}\, c_p\, dT_{2_y} = \frac{dy}{l_2}\frac{M_1}{w}\, c_p\, T_{1_{ex}}$$

and by substituting in equation (15)

$$\frac{dT_{2_y}}{T_{2_y}} = -\frac{M_1}{l_2 M_2}\left(1 - e^{-\frac{w l_1 l_2 h}{M_1 c_p}}\right) dy$$

Integrating between the limits T_2 to $T_{2_{ex}}$ and 0 to l_2 gives

$$e^{-\frac{M_1}{M_2}\left(1 - e^{-\frac{wl_1 l_2 h}{M_1 c_p}}\right)} = \frac{T_{2_{ex}}}{T_2} = 1 - \eta$$

or

$$\eta = 1 - e^{-\frac{M_1}{M_2}\left(1 - e^{-\frac{wl_1 l_2 h}{M_1 c_p}}\right)} \tag{16}$$

By a similar procedure the cooling effectiveness for counterflow is

$$\eta_c = \frac{1 - \dfrac{e^{-\frac{M_1}{M_2}\frac{wl_1 l_2 h}{M_1 c_p}}}{e^{-\frac{wl_1 l_2 h}{M_1 c_p}}}}{1 - \dfrac{M_2}{M_1}\dfrac{e^{-\frac{M_1}{M_2}\frac{wl_1 l_2 h}{M_1 c_p}}}{e^{-\frac{wl_1 l_2 h}{M_1 c_p}}}} \tag{17}$$

and for parallel flow,

$$\eta_p = \frac{1 - e^{-\frac{wl_1 l_2 h}{M_1 c_p}} e^{-\frac{M_1}{M_2}\frac{wl_1 l_2 h}{M_1 c_p}}}{M_2/M_1 + 1} \tag{18}$$

Equations (16), (17), and (18) show that the cooling effectiveness for the three types of flow depends on M_1/w, M_1/M_2, h, l_1, and l_2. These equations are based on the assumption that T_{2_y} is constant along a line at right angles to the direction of the charge flow. This assumption is not strictly valid for cross flow (equation (16)); it is shown, however, in references 2 and 3 to introduce small error in the range of practical intercooler operation. The assumption is valid for counterflow and parallel flow; so equations (17) and (18) are rigorous.

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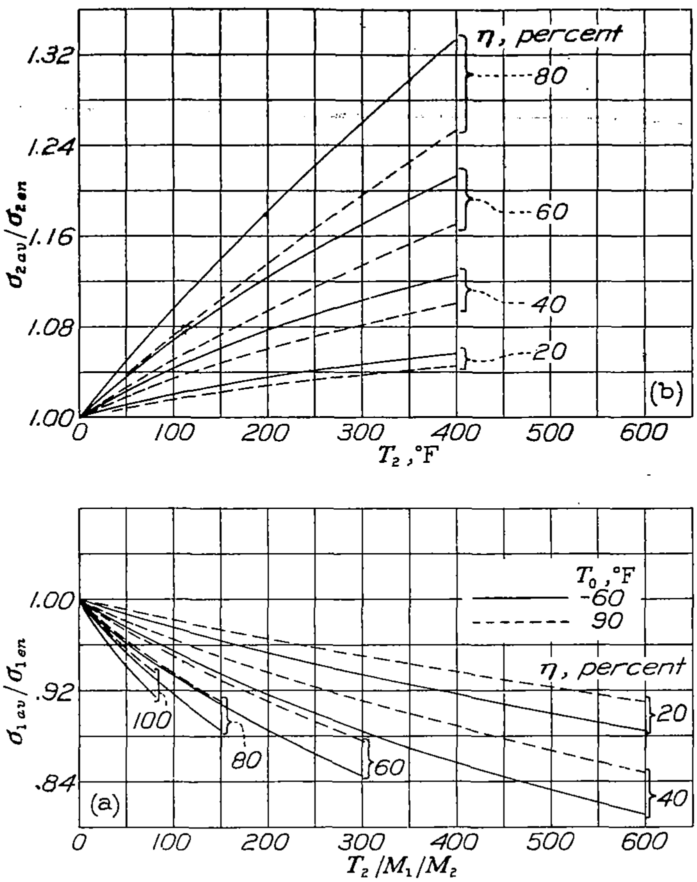


Figure 1.- Conversion factors for σ_{1av} and σ_{2av}.

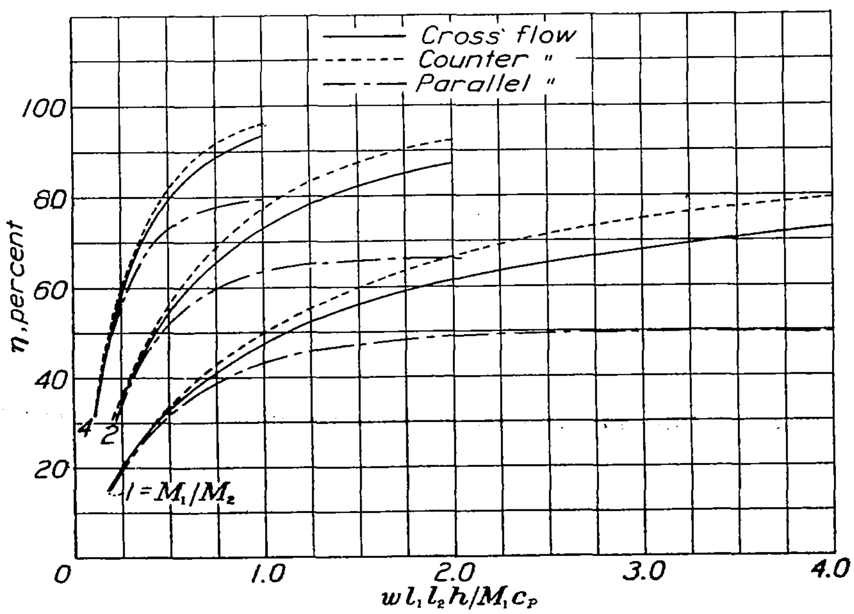
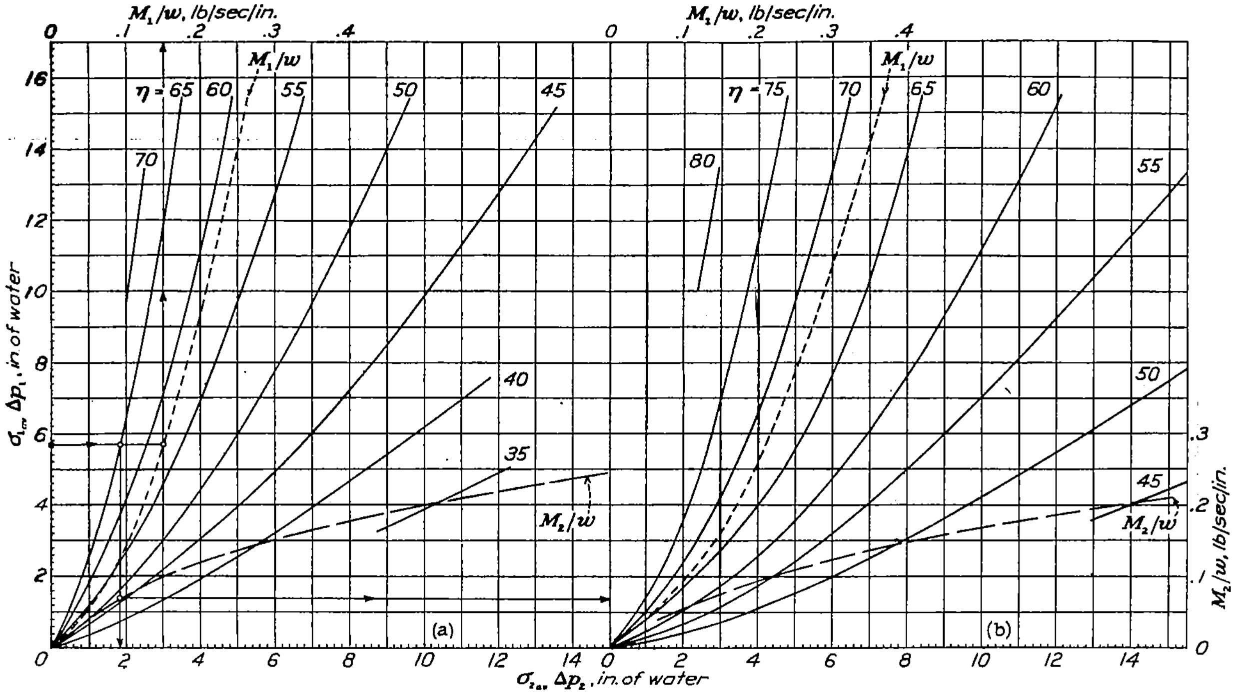
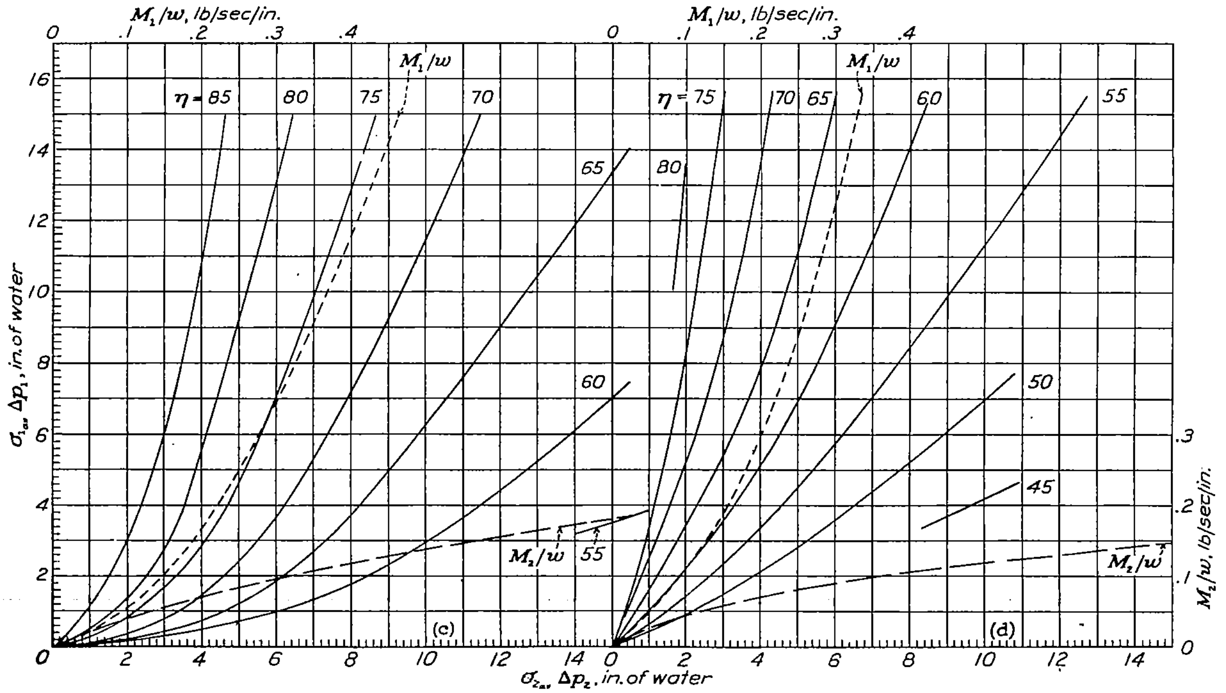


Figure 2.- Comparison of cooling effectiveness of various types of flow.



(a) Intercooler 1.
l_1 = 10 in., l_2 = 10 in., W/w = 4 lb/in.

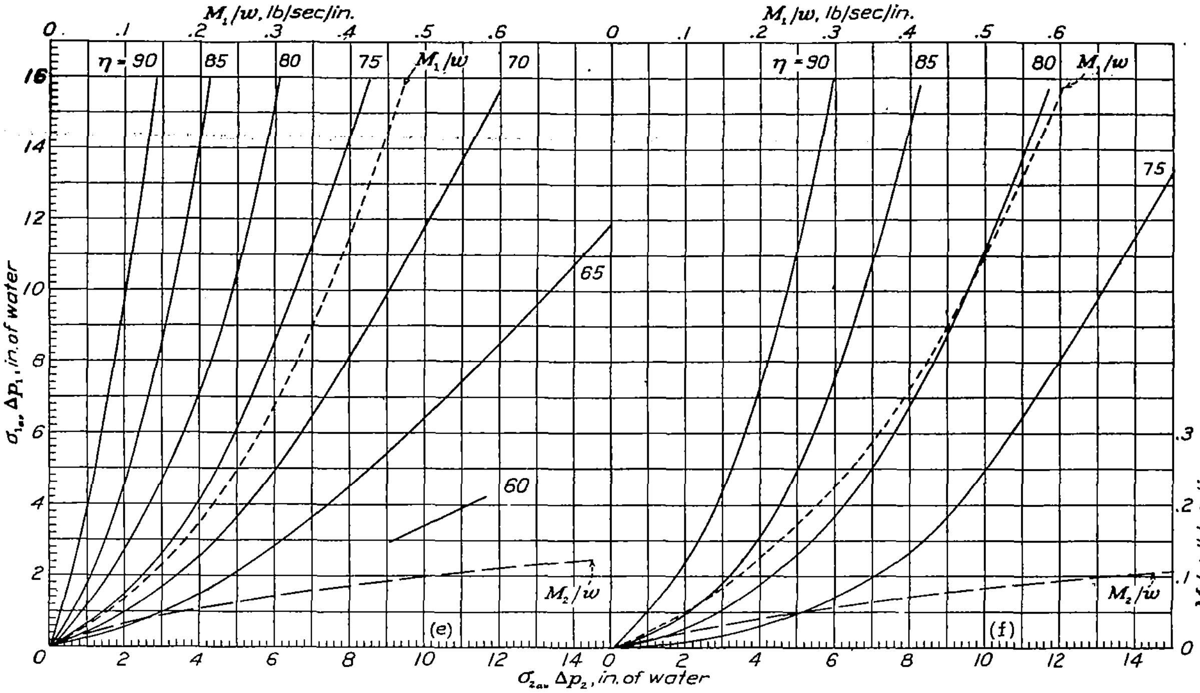
(b) Intercooler 2.
l_1 = 10 in., l_2 = 14 in., W/w = 5.2 lb/in.



(c) Intercooler 3.
l_1 = 10 in., l_2 = 18 in., W/w = 6.3 lb/in.

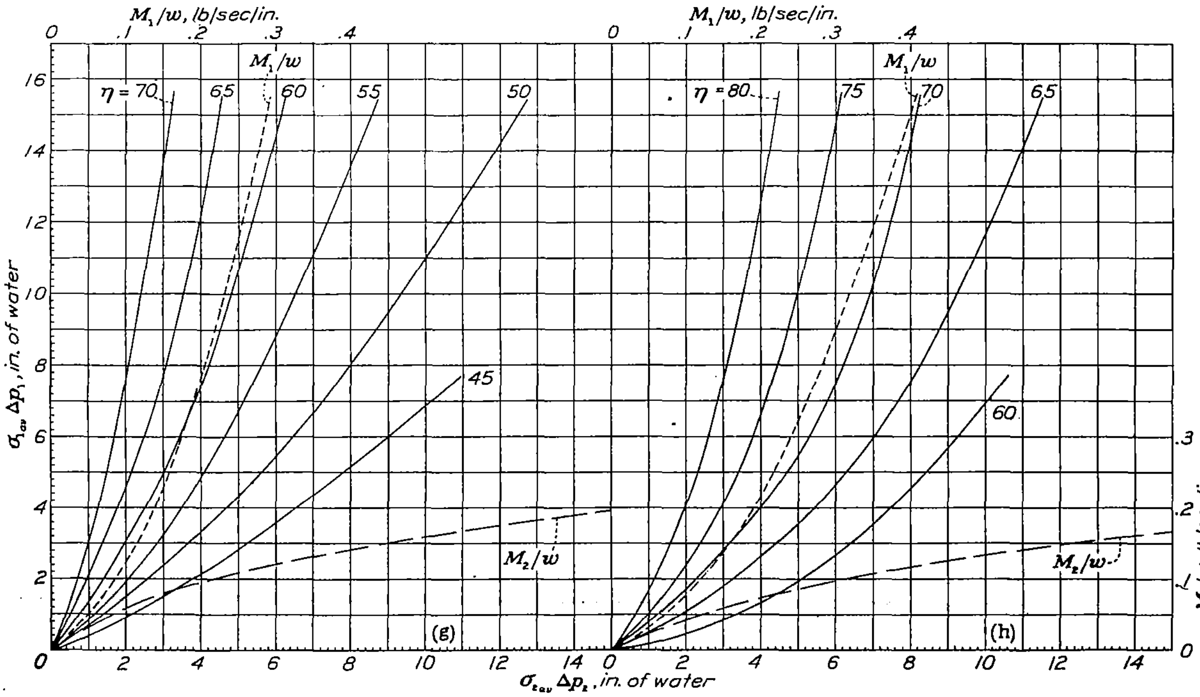
(d) Intercooler 4.
l_1 = 6 in., l_2 = 10 in., W/w = 2.4 lb/in.

Figure 3a to i.- Performance charts for Harrison intercoolers (copper).



(e) Intercooler 5.
l_1 = 6 in., l_2 = 14 in., W/w = 3.1 lb/in.

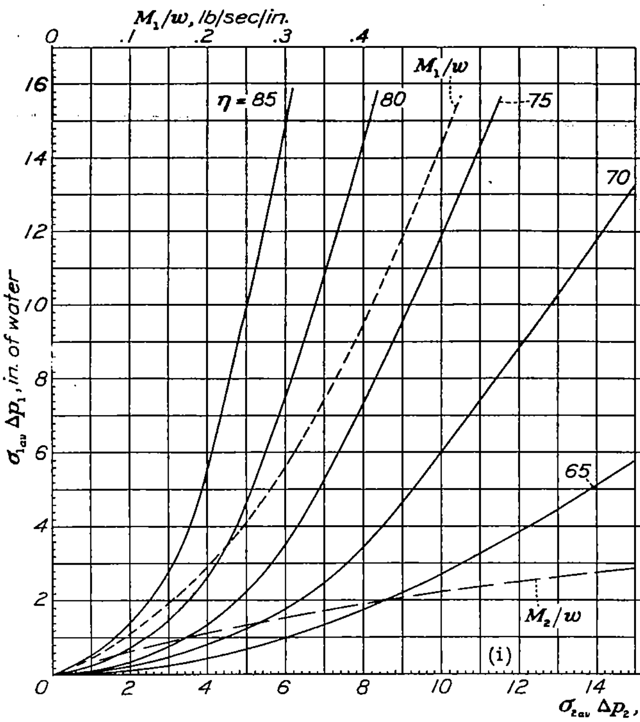
(f) Intercooler 6.
l_1 = 6 in., l_2 = 18 in., W/w = 3.8 lb/in.



(g) Intercooler 25.
l_1 = 8 in., l_2 = 10 in., W/w = 3.2 lb/in.

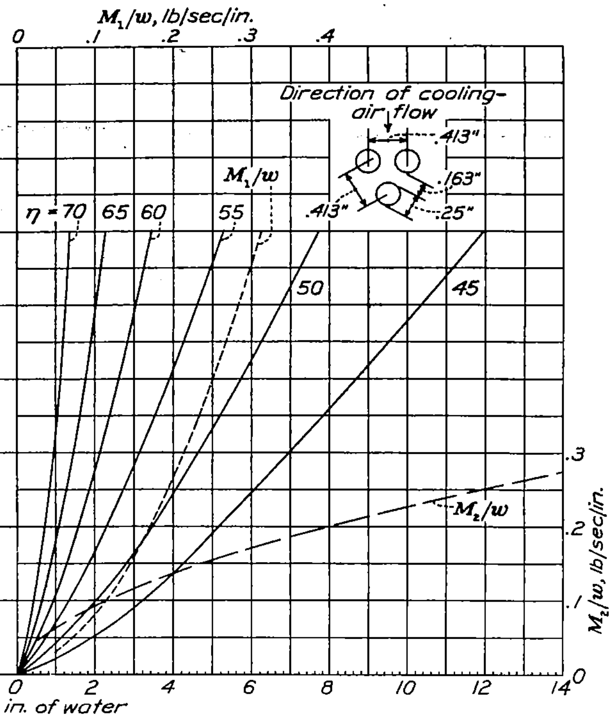
(h) Intercooler 26.
l_1 = 8 in., l_2 = 14 in., W/w = 4.1 lb/in.

Figure 3.- Continued.



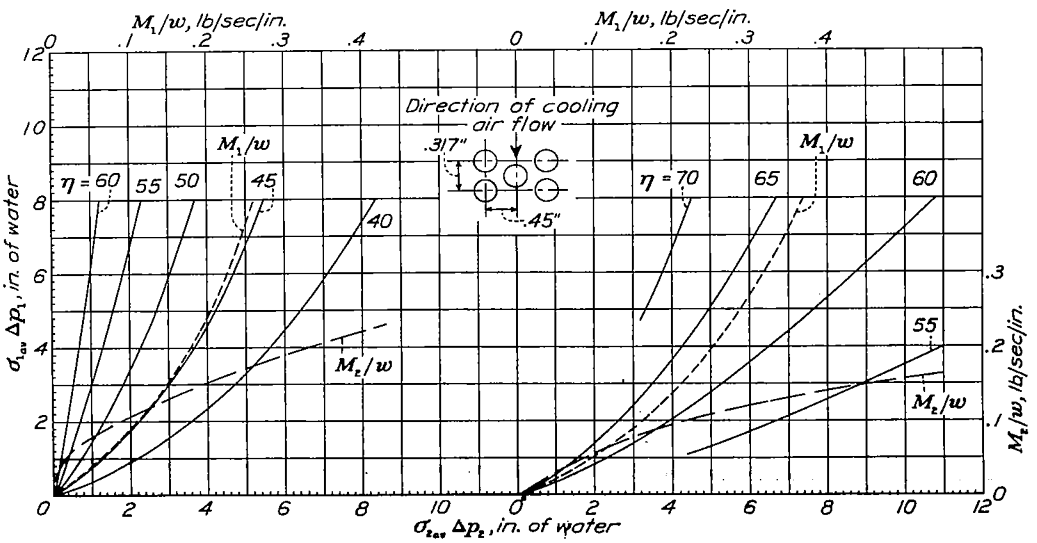
(i) Intercooler 27.
l_1 = 8 in., l_2 = 18 in., W/w = 5.1 lb/in.

Figure 3.- Concluded.



l_1 = 10.6 in., l_2 = 25 in., W/w = 2.5 lb/in.

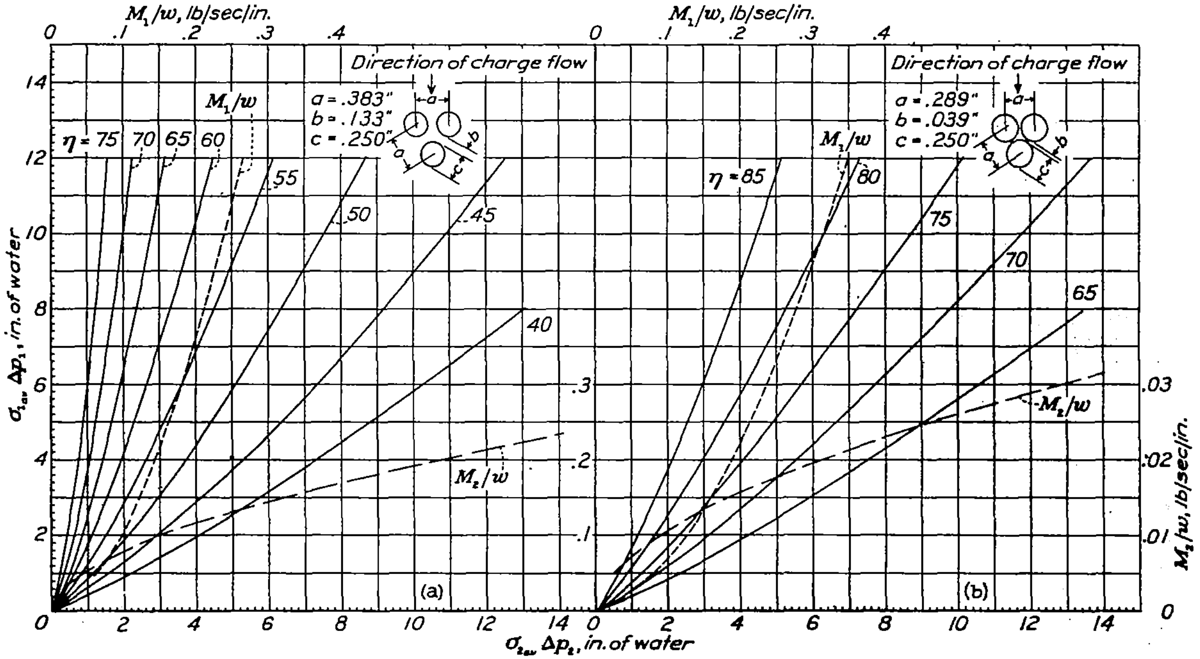
Figure 5.- Performance chart for charge-through-tube type intercooler 10.



(a) Intercooler 7.
l_1 = 11.75 in., l_2 = 14.25 in.,
W/w = 1.94 lb/in.

(b) Intercooler 8.
l_1 = 10.25 in., l_2 = 20.75 in.,
W/w = 2.38 lb/in.

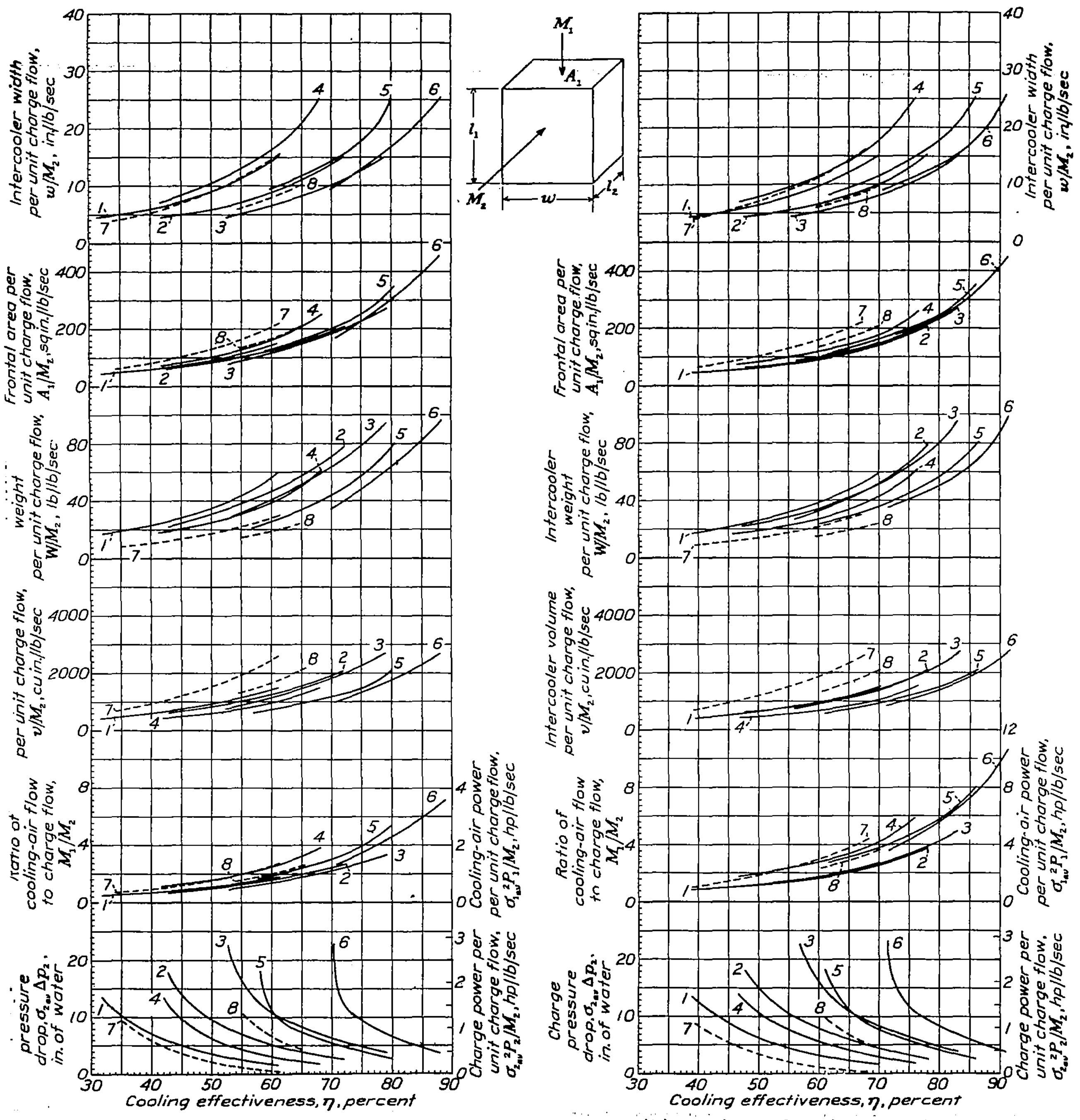
Figure 4. - Performance charts for Airesearch intercoolers (aluminum).



(a) Intercooler 18.
l_1 = 20 in., l_2 = 10 in., W/w = 2.1 lb/in.

(b) Intercooler 24.
l_1 = 20 in., l_2 = 1.2 in., W/w = 0.52 lb/in.

Figure 6.- Performance chart for charge-across-tube type intercooler.



(a) $\sigma_{1_{av}}\Delta p_1$, 4 inches of water.

(b) $\sigma_{1_{av}}\Delta p_1$. 8 inches of water

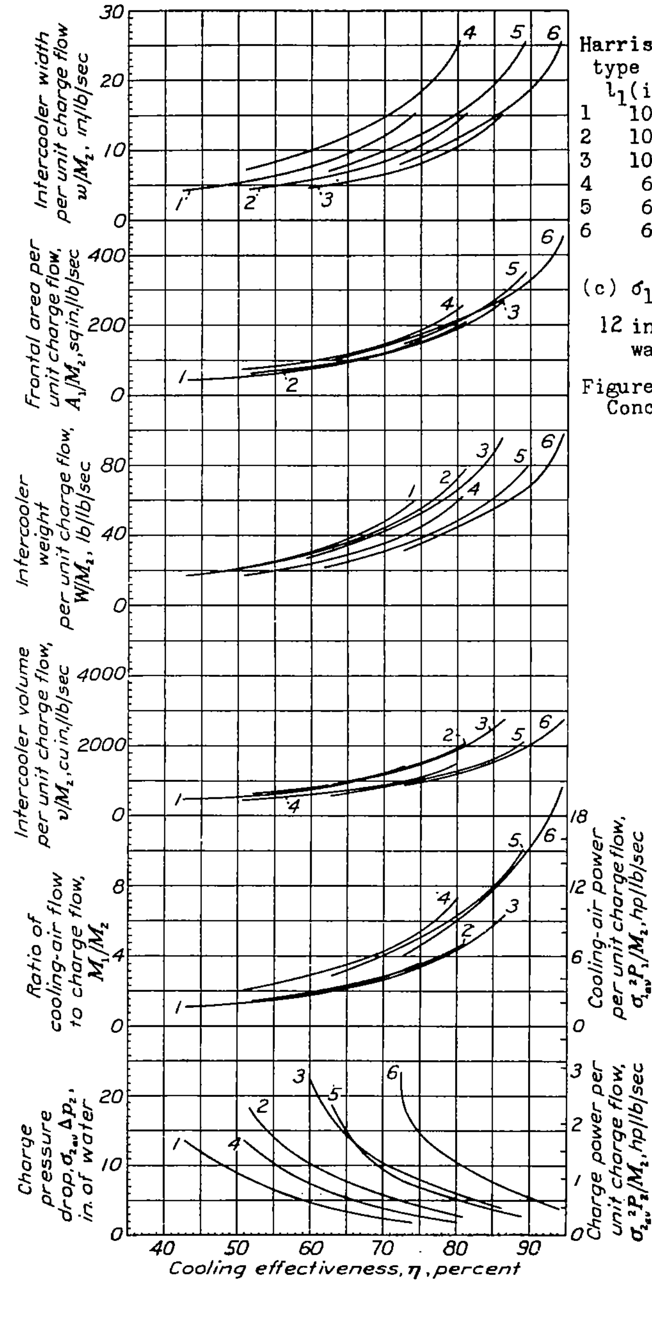
Harrison plate-type copper intercoolers

No.	1	2	3	4	5	6
l_1(in.)	10	10	10	6	6	6
l_2(in.)	10	14	18	10	14	18

Airesearch tubular-type aluminum intercoolers

No.	7	8
l_1(in.)	11.75	10.25
l_2(in.)	14.25	20.75

Figure 7a to c. - Comparison of characteristics of commercial intercoolers.

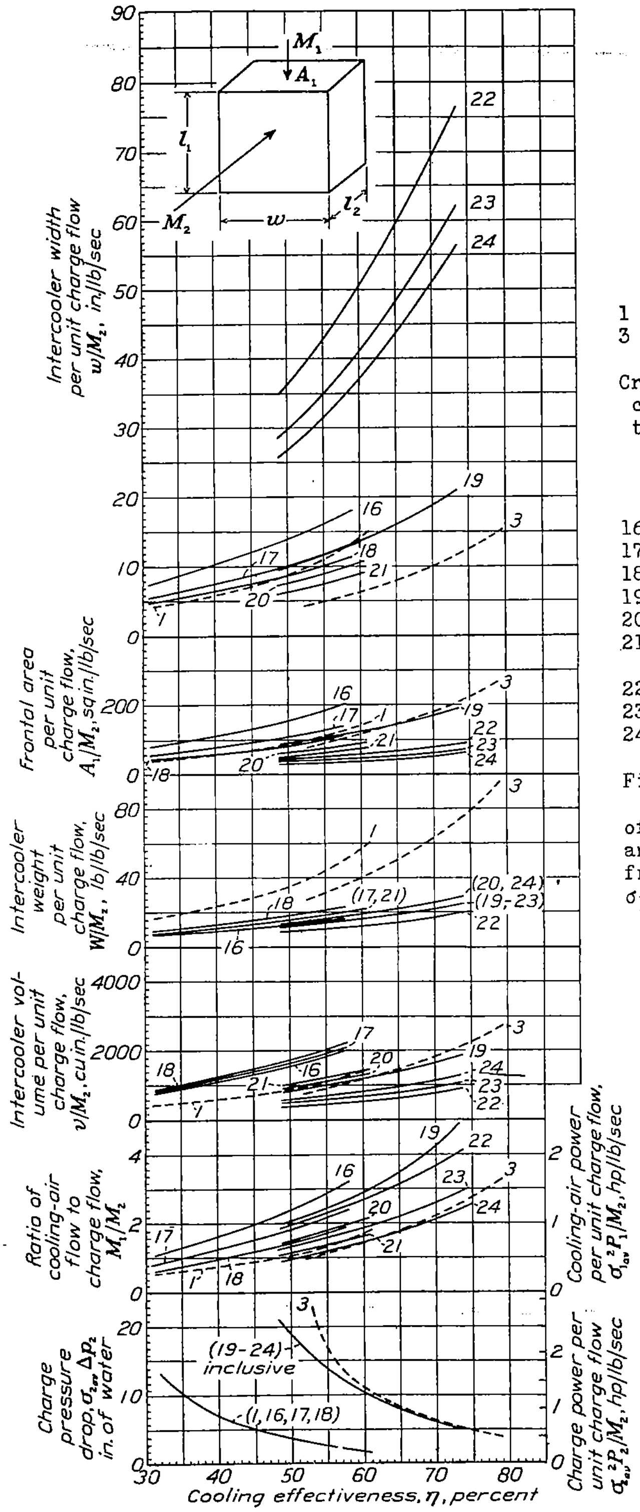


Harrison plate-type intercooler

	l_1(in.)	l_2(in.)
1	10	10
2	10	14
3	10	18
4	6	10
5	6	14
6	6	18

(c) $\sigma_{1_{av}}\Delta p_1$, 12 inches of water.

Figure 7c.- Concluded.

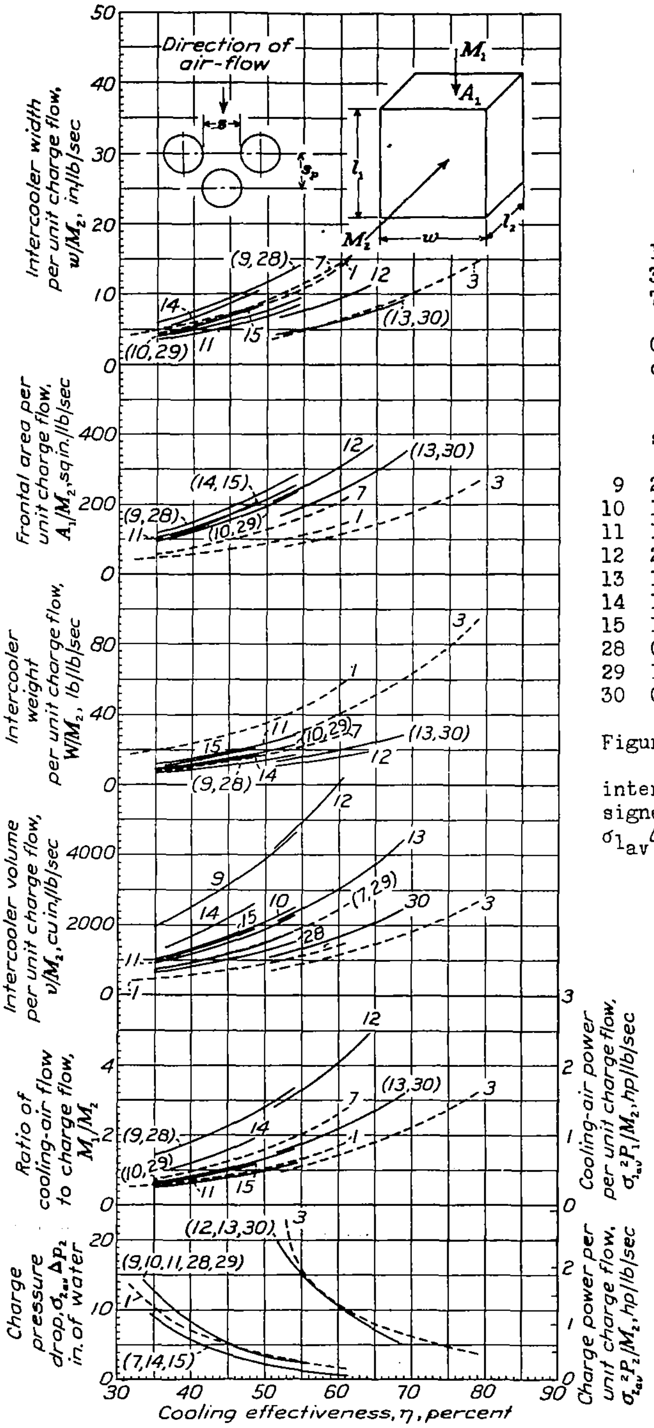


Type	l_1(in.)×l_2(in.)
1 Harrison	10 × 10
3 "	10 × 18

Cross-flow tubular intercooler; charge-across-tube type.
d = 0.25 in.
m = 30 banks

	l_1/d	l_2(in.)	md/s
16	40	11.0	42.3
17	60	10.4	48.5
18	80	9.9	56.4
19	40	8.8	83.0
20	60	8.5	95.9
21	80	8.3	108.1
	m = 5 banks		
22	40	1.2	137
23	60	1.2	164
24	80	1.2	194

Figure 8.- Comparison of characteristics of commercial intercoolers and intercoolers designed from charts of reference 2; $\sigma_{1av}\Delta p_1$, 4 inches of water.

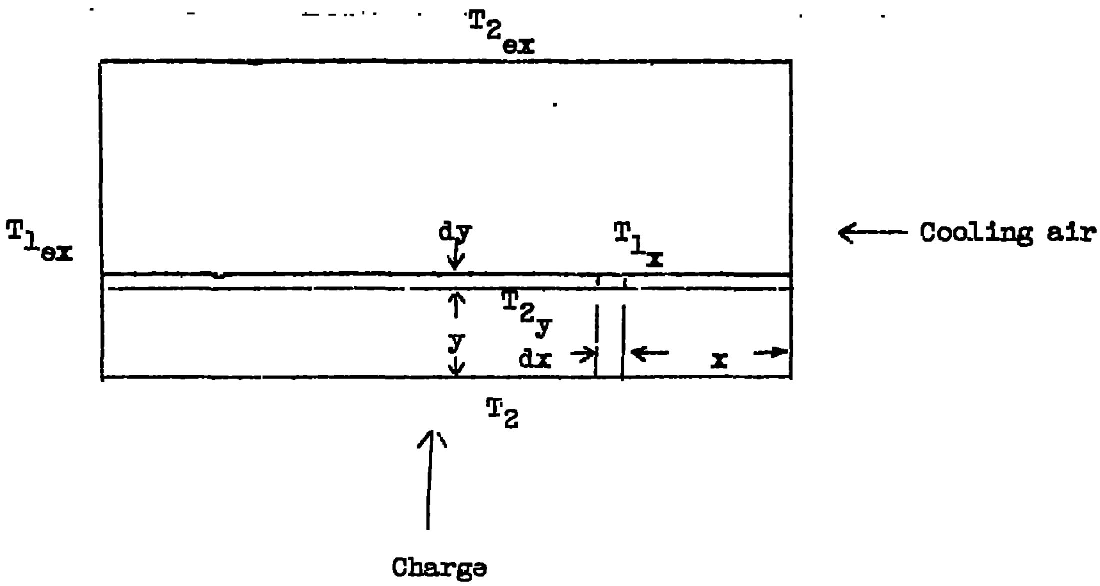


Type	l_1(in.)	×	l_2(in.)
1 Harrison	10	×	10
3 Harrison	10	×	18
7 Airesearch	11.75	×	14.25

Cross-flow tubular intercooler;
charge-through-tube type
d = 0.25 inch
m = 30 banks

	s_p/d	l_1(in.)	l_2/d	md/s
9	2.25	16.5	80	18.9
10	1.43	10.6	100	46.0
11	1.22	9.1	120	74.2
12	2.67	19.6	130	14.4
13	1.71	12.6	150	31.0
14	1.70	12.6	80	31.2
15	1.30	9.7	100	60.3
28	0.75	5.5	80	18.9
29	1.04	7.8	100	46.0
30	0.94	6.9	150	31.0

Figure 9.- Comparison of characteristics of commercial intercoolers and intercoolers designed from charts of reference 3; $\sigma_{1_{av}}\Delta p_1$, 4 inches of water.



T_w Temperature difference between wall at x and cooling air at entrance, °F.

T_{1_x} Temperature rise of cooling air in flowing x distance along elemental strip, °F.

$T_2-T_{2_y}$ Temperature drop of charge in flowing y distance over plate, °F.

$T_{1_{ex}}$ Total temperature rise of cooling air, °F.

$T_2-T_{2_{ex}}$ Total temperature drop of charge, °F.

Figure 10.- Diagram of heat exchange through a section of a cross-flow intercooler.