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RESEARCH MEMORANDUM

STUDY OF RAM-AIR HEAT EXCHANGERS FOR REDUCING TURBINE COOLING-AIR TEMPERATURE OF A SUPERSONIC AIRCRAFT TURBOJET ENGINE

By Anthony J. Diaguila, John N. B. Livingood, and Ernst R. G. Eckert

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Cleveland, Ohio

NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS

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4073

CV-1

SUMMARY

An analytical investigation was made to determine the sizes and weights of the cores of crossflow finned-tube heat exchangers for reducing the temperature of turbine cooling air for an engine designed for operation at a flight Mach number of 2.5 and an altitude of 70,000 feet. A compressor-bleed-air weight flow of 2.70 pounds per second was assumed for the turbine coolant; ram air was considered for reducing the compressor-bleed-air temperature. The available pressure drops of the two fluids and the inlet states of each fluid were prescribed. Reductions in compressor-bleed-air temperature of 100°, 200°, 300°, and 400° F and values of the ratio of compressor-bleed-air to ram-air weight flow of 1.00, 0.75, 0.50, and 0.25 were considered. The calculations were made with the use of prepared charts.

For a compressor-bleed-air temperature reduction of 300° F, the heat-exchanger core weight was reduced from about 49 to 19 pounds as the ratio of compressor-bleed-air to ram-air weight flow was decreased from 1.00 to 0.25. For heat exchangers of moderate weight (30 to 60 pounds), the compressor-bleed-air temperature reduction can be increased by about 100° F if the weight-flow ratio is reduced from 1.00 to 0.25. Reductions in weight-flow ratio for a fixed compressor-bleed-air temperature drop result in increased frontal area in the primary-fluid direction. In all calculations, only core weight was considered; duct weight was not considered.

For the conditions of this investigation the heat-exchanger core weight was less than 60 pounds in most cases. If a sea-level static specific engine weight of 0.300 without a heat exchanger is assumed, the core weight would increase the specific engine weight to only 0.305 for the engine size considered (12,000-pound sea-level static dry thrust).

INTRODUCTION

The problems involved in cooling gas-turbine blades in aircraft flying at Mach numbers of 2.5 or higher can be simplified if the cooling air can be supplied to the turbine at a temperature considerably lower than compressor discharge temperature. A possible method of obtaining cooling air at such low temperatures is to cool compressor bleed air by use of ram-air heat exchangers. An investigation of required heat-exchanger core sizes and weights is reported herein. 4073

Several methods of reducing cooling-air temperature may be possible. Preliminary thoughts on several air-refrigeration devices are presented in reference 1. These devices were not thoroughly analyzed, however, and some of them may not be practical. In some cases, air at a lower temperature can be obtained by bleeding off an earlier stage of the compressor. This possibility is discussed in reference 2. The final choice of the type of system to be used will depend upon such things as effectiveness in reducing cooling-air temperature, weight, complexity, effect on frontal area, effect on engine performance, and so forth. No attempt to compare different methods of reducing cooling-air temperature is made herein. Rather, this report presents the sizes and weights of the cores of ram-air heat exchangers for a range of cooling-air temperature reductions. The investigation, which was made at the NACA Lewis laboratory, considered heat exchangers for an engine designed for operation at a flight Mach number of 2.5 and an altitude of 70,000 feet. A typical compressor-bleed-air weight flow was prescribed for purposes of cooling the turbine.

Weight is of primary importance in such an application. Therefore, a heat-exchanger core of minimum weight is desired. Information presented in reference 3 shows that for minimum weight a finned-tube heat exchanger is a desirable core configuration. Reference 3 considers only the case where one heat resistance is negligible: for example, an air-to-liquid heat exchanger. It is expected, however, that conditions will be similar in an air-to-air heat exchanger; and a finned-tube arrangement, as shown in reference 3, was chosen for the following calculations. For the application considered herein, the initial conditions for each fluid are known. In addition, the allowable pressure drop for both the compressor bleed air and the ram air is assumed as prescribed.

Reference 4 presents a method for calculating the core dimensions of a crossflow gas-to-gas heat exchanger with prescribed initial conditions and pressure drop. This method is applied herein in the determination of the sizes and weights of a number of finned-tube heat exchangers for the previously mentioned flight conditions. A range of temperature reductions in the compressor bleed air from 100° to 400° F and a range of ratios of bleed-air to ram-air weight flows from 1.00 to 0.25 are considered. The smallest ratio of compressor-bleed-air to ram-bleed-air weight flow was limited to 0.25 so that the required increase in the engine inlet diameter which is necessary to supply the additional ram air would be reasonably small. In an engine for supersonic flight the inlet diameter may be the limiting radial dimension.

The results presented herein are for a single flight condition and a fixed compressor-bleed-air flow. No attempts to generalize for other conditions should be made. Furthermore, a final decision as to what type and size heat exchanger should be installed involves many design considerations as well as more intensive calculations.

This report does, however, give an indication of trends that might be expected in heat-exchanger size and weight as cooling-air temperature reduction and ratio of cooling-air flow to ram-air flow are varied.

SYMBOLS

The following symbols, with consistent units, are used:

A	heat-transfer area
A'	free-flow area
c_p	specific heat at constant pressure
d	hydraulic diameter
f	friction factor
G	mass velocity, w/A'
h	heat-transfer coefficient
K_c	pressure-loss coefficient for abrupt contraction
K_e	pressure-loss coefficient for abrupt expansion
k	thermal conductivity
l	length
$l_{f/2}$	one-half fin length (see fig. 5)
Pr	Prandtl number, $c_p \mu / k$
p	pressure

R gas constant

Re Reynolds number, Gd/μ

St Stanton number, h/Gc_p

T temperature

Tu heat-transfer parameter (number of transfer units) 4073

t_f fin thickness

w weight-flow rate

α heat-transfer surface area per unit volume

η_f fin effectiveness

η_T thermal effectiveness

η_O surface effectiveness

μ viscosity

σ ratio of free-flow to frontal area

Subscripts:

F frontal

f fin

i heat-exchanger inlet

m mean

max maximum

n no-flow direction

1 fluid on compressor-bleed-air side of heat exchanger

2 fluid on ram-air side of heat exchanger

STATEMENT OF PROBLEM

The current investigation, as stated previously, is aimed at determining the sizes and weights of ram-air heat-exchanger cores for use in a hypothetical engine flying at a Mach number of 2.5 and an altitude of 70,000 feet having a sea-level static dry thrust of 12,000 pounds with a turbine-inlet temperature of 2500° R. For such an application of a heat exchanger, attention must be given to the space available for installation in the engine. Figure 1 shows one proposed type of installation. Compressor air, to be used for cooling the turbine, is bled from the compressor discharge for this investigation and is denoted as fluid 1. Ram air, used to cool the compressor bleed air, is bled from the engine inlet and, after passage through the heat exchanger, may be discharged into an engine ejector; the ram bleed air is denoted as fluid 2. Figure 2 presents a view of the finned-tube geometry which was selected (from ref. 3) for the heat-exchanger core. The compressor bleed air is passed through the tubes. The ram air passes in crossflow over the fins and the tubes.

The pressure drop of the turbine cooling air (fluid 1) through the heat exchanger was assumed to be about 20 percent of the compressor-bleed-air pressure; for this investigation, there was sufficient pressure left to overcome the inlet, exit, and ducting losses and to force the fluid through the turbine blades. A pressure drop of about 35 percent of the ram-air pressure available at the heat-exchanger inlet was assumed for fluid 2. This value was arbitrarily chosen with the possiblity of using the ram air after passing the heat exchanger as secondary air in an engine ejector. Sufficient data are not available at this time to determine the optimum pressure drop for an arrangement of this type. It is believed, however, that 35 percent is a reasonable assumption. A weight flow (w_1) for the turbine-cooling air of 2.7 pounds per second was prescribed; this is a value to be expected at the specified speed and altitude conditions. Fluid conditions, the selected ranges of the variables, and the heat-exchanger geometry are prescribed.

CALCULATION PROCEDURE

A procedure for the determination of the core dimensions of a gas-to-gas heat exchanger with a prescribed core configuration is presented in detail in reference 4. Since the same procedure is employed in this report, only a resumé of its development is given herein. The following fluid conditions were prescribed:

	Compressor bleed air (fluid 1)	Ram air (fluid 2)
Temperature, °R	1410	880
Pressure, lb/sq ft abs	5300	1080
Pressure drop, lb/sq ft	1000	400

Temperature drops in the compressor bleed air ($-\Delta T_1$) of 100°, 200°, 300°, and 400° F and weight-flow ratios w_1/w_2 of 1.00, 0.75, 0.50, and 0.25 are considered.

The prescribed heat-exchanger geometry is as follows:

Fin area/total area	0.795
Fin pitch, 1/in.	9.681
Fin thickness, t_f, in.	0.004
Hydraulic diameter, tube side, d_1, ft	0.018
Hydraulic diameter, fin side, d_2, ft	0.0118
Free-flow area/frontal area, tube side, σ_1	0.219
Free-flow area/frontal area, fin side, σ_2	0.697
Heat-transfer area/volume, tube side, α_1, 1/ft	48.76
Heat-transfer area/volume, fin side, α_2, 1/ft	229

Assumptions

The following assumptions were made:

(1) The core configuration is fixed with the prescribed values for (a) the hydraulic diameters d_1 and d_2 of the passages of both fluids, (b) the thickness t_f of the fins, (c) the ratios of free-flow to frontal areas σ_1 and σ_2 for both flow passages, and (d) the heat-transferring areas per unit volume α_1 and α_2 of the exchanger core.

(2) The friction factors f_1 and f_2 and Stanton numbers St_1 and St_2 are known as functions of the Reynolds numbers Re_1 and Re_2 of the two fluids.

(3) Crossflow through the exchanger is assumed.

(4) The entrance pressures and temperatures $p_{1,i}$, $T_{1,i}$, $p_{2,i}$, and $T_{2,i}$ of both fluids are prescribed.

(5) The weight-flow rates w_1 and w_2 of both fluids are prescribed.

(6) The available pressure drops Δp_1 and Δp_2 of both fluids are prescribed.

(7) The temperature rise (ΔT) or drop ($-\Delta T$) of one fluid is prescribed.

(8) The material used to manufacture the heat-exchanger core is assumed to have a density of 0.300 pound per cubic inch.

It is shown in reference 4 that under the foregoing assumptions consideration of the heat-balance, weight-flow, pressure-drop, and effectiveness relations results in a system of five equations in five unknowns. These equations are (see ref. 4):

$$\eta_{O,2}(ReSt)_2 = \frac{\frac{\alpha_1}{\alpha_2}\left(\frac{c_p\mu}{d}\right)_1\left(\frac{d}{c_p\mu}\right)_2}{\left(\frac{\alpha}{\sigma Re}\right)_1 \frac{l_1}{Tu} - \frac{1}{\eta_{O,1}(ReSt)_1}} \tag{1}$$

$$\Delta T_2 = -\frac{c_{p,1}}{c_{p,2}}\frac{d_2}{d_1}\frac{\mu_1}{\mu_2}\frac{\sigma_1}{\sigma_2}\frac{Re_1}{Re_2}\frac{l_2}{l_1}\Delta T_1 \tag{2}$$

$$l = \frac{d}{2f}\left(\frac{1 - \frac{\Delta p}{p_i}}{2 - \frac{\Delta p}{p_i} + \frac{\Delta T}{T_i}}\right)\left[\frac{2p_i^2\Delta p d^2}{\mu^2 T_i p_i R(Re)^2} - (1 + \sigma^2 + K_e)\left(\frac{1 + \frac{\Delta T}{T_i}}{1 - \frac{\Delta p}{p_i}}\right) - (K_c - 1 - \sigma^2)\right] \tag{3}$$

(There is one l-equation for each side of exchanger. The proper sign must accompany the ΔT term: a negative sign indicates a temperature reduction; a positive sign indicates a temperature rise.)

$$w_1 = \sigma_1 Re_1 \frac{\mu_1}{d_1} l_2 l_n$$

or (4)

$$w_2 = \sigma_2 Re_2 \frac{\mu_2}{d_2} l_1 l_n$$

The surface heat-transfer effectiveness η_O in equation (1) may be expressed in terms of the fin effectiveness η_f by the equation

$$\eta_O = 1 - \frac{A_f}{A}(1 - \eta_f) \tag{5}$$

and the thermal effectiveness η_T necessary for evaluation of Tu in equation (1) may be expressed by the equation

$$\eta_T = \left| \frac{\Delta T_{max}}{T_{1,i} - T_{2,i}} \right| \tag{6}$$

The five unknowns in equations (1) to (4) are l_1, l_2, l_n, Re_1, and Re_2. One of these unknowns l_n is common to each of the equations (l_n is contained in the area term present in both Reynolds numbers). As a consequence, l_n can be eliminated from equations (1), (2), and both forms of (3). These four equations can then be solved for l_1, l_2, Re_1, and Re_2, and l_n can finally be determined from equation (4).

A solution of equations (1) to (4) according to reference 4 necessitates an iteration process. Such a procedure is effective if only a few calculations on a specific configuration are required. If many solutions are needed, it is advantageous to construct charts and ~~to~~ employ the charts rather than the equations in the determination of the core dimensions. For this purpose it was necessary to specify the heat-exchanger core configuration. Figure 3 presents data for the friction factor f_2 and Stanton number St_2 for the ram air (fluid 2). These data are taken from reference 5. For the compressor air (fluid 1), the friction factor f_1 and the Stanton number St_1 are obtained from the following equations given in reference 6:

$$f_1 = 0.050 \, Re_1^{-0.2} \tag{7}$$

$$St_1 = \frac{0.019 \, Re_1^{-0.2}}{Pr^{2/3}} \tag{8}$$

Additional data obtained from reference 5, which are necessary for a graphical solution, are presented in figures 4 and 5. From figure 4 and the thermal effectiveness η_T (eq. (6)), the required number of transfer units Tu can be obtained. Figure 5 permits determination of the fin effectiveness η_f, which is necessary for calculating the surface effectiveness η_O. In the present report, however, $\eta_{O,1}$ is 1 because there are no fins on the compressor bleed side of the heat exchanger. The information in figure 5 is useful for checking the validity of the selected value of $\eta_{O,2}$, as is demonstrated in reference 4.

Charts

The charts presented in figures 6 to 9 were constructed for the particular finned-tube heat-exchanger configuration considered. The variation of surface effectiveness with Reynolds number over a considerable range was comparatively small. Therefore, an average value of the surface effectiveness $\eta_{0,2} = 0.80$ was prescribed. In addition, end losses are neglected; that is, $K_c = K_e = 0$.

4073

The graphical representation of the first equation (3) is presented in figure 6, which gives the relation between Re_1 and l_1. The connection between the two Reynolds numbers Re_1 and Re_2 is obtained from equation (1) and figure 6 and is presented in figure 7. A more convenient representation of the results shown in figure 3 is given in figure 8. The second form of equation (3) is shown in figure 9. The method of construction of these charts is given in detail in reference 4. An illustration of their use follows.

CV-2

NUMERICAL EXAMPLE

The particular example chosen to illustrate the graphical solution has a value of $\Delta T_1 = -400^\circ$ F and a ratio of $w_1/w_2 = 0.25$.

Within the temperature range under investigation, the specific heat of air is assumed constant. From a heat-balance equation, with c_p constant, $\Delta T_1 = -400^\circ$ F, and $w_1/w_2 = 0.25$, the value of ΔT_2 is found to be 100° F. The effectiveness of the exchanger, given by equation (6), is

$$\eta_T = \left|\frac{\Delta T_1}{T_{1,i} - T_{2,i}}\right| = \left|\frac{-400}{1410 - 880}\right| = 0.755$$

The required number of transfer units Tu is found from figure 4 to be 1.72. Property values are based on the mean temperatures of the two fluids, which are

$$T_{1,m} = 1410 - 400/2 = 1210^\circ \text{ R}$$

$$T_{2,m} = 880 + 100/2 = 930^\circ \text{ R}$$

The friction factors and Stanton numbers for fluid 1 are given by equations (7) and (8); those for fluid 2 are taken from figure 3. The value of $Pr^{2/3}$ was assumed to be 0.75 for both fluids.

The step-by-step solution follows:

(1) A value of $Re_2 = 5000$ was assumed. From figure 7(d), for the required $Tu = 1.72$, Re_1 was found to be 24,000.

(2) For the assumed $Re_2 = 5000$ and the given $\Delta T_2 = 100^\circ$ F, the length $l_2 = 18.4$ inches was found from figure 9.

(3) The length $l_1 = 46.80$ inches was found from figure 6 for the previously found value of Re_1 (24,000) and the given $\Delta T_1 = -400^\circ$ F. 4073

(4) From equation (2), ΔT_2 was found to be 184.2° F. This is considerably larger than the prescribed ΔT_2 of 100° F, so steps (1) to (3) were repeated for a larger assumed Re_2.

(5) For an assumed $Re_2 = 5500$, steps (1) to (3) resulted in $Re_1 = 24{,}350$, $l_2 = 15.03$ inches, $l_1 = 46.0$ inches, and $\Delta T_2 = 141.0^\circ$ F. This value of ΔT_2 is still too large, so another and larger value of Re_2 was assumed.

(6) For an assumed $Re_2 = 6000$, the same procedure yielded $Re_1 = 24{,}650$, $l_2 = 12.44$ inches, $l_1 = 44.8$ inches, and $\Delta T_2 = 111.3^\circ$ R. This value of ΔT_2 is also large.

(7) A plot of these three solutions for ΔT_2 and l_2 on figure 9 intersected a horizontal line at the prescribed value of $\Delta T_2 = 100^\circ$ F. This intersection then yielded the correct values of $l_2 = 11.55$ inches and $Re_2 = 6200$.

(8) From the correct value of $Re_2 = 6200$, ($Tu = 1.72$; $w_1/w_2 = 0.25$) the correct value of $Re_1 \doteq 24{,}750$ was read from figure 7(d) and the correct value of $l_1 = 44.90$ inches from figure 6.

(9) The value of l_n was found from equation (4) to be 5.13 inches.

(10) The volume of the heat exchanger was found to be 2660 cubic inches. The heat-exchanger weight, assuming the exchanger to be made from a metal with a density of 0.300 pound per cubic inch, was calculated from the configuration and found to be 43.86 pounds.

RESULTS AND DISCUSSION

The particular example discussed in the preceding section is one of a series of calculations made for the same heat-exchanger core configuration and the same inlet conditions for each fluid. Values of the temperature change for fluid 1 and the ratio of w_1 to w_2 were varied. The cases investigated are checked below:

$-\Delta T_1$, °F	w_1/w_2			
	1.00	0.75	0.50	0.25
100	√	√	√	
200	√	√	√	√
300	√	√	√	√
400		√	√	√

4073

Blanks in the table result from the fact that the friction and Stanton number data in reference 5 are limited to a range of Reynolds number, and the Reynolds numbers for the cases represented by the blank spaces fell outside this range.

CV-2 back

The heat-exchanger core lengths, frontal areas in the two flow directions, and weights for each of the 14 cases considered are listed in table I. Graphical presentations of the weights and frontal areas in the primary-fluid direction $A_{F,2}$ are given in figures 10 to 13. Figure 10 shows the heat-exchanger core weights plotted against the prescribed temperature drop $-\Delta T_1$ for a range of values of w_1/w_2. The figure shows that, for a fixed value of $-\Delta T_1$, the heat-exchanger core weight decreases with decreases in w_1/w_2; for example, for $\Delta T_1 = -300^\circ$ F, decreasing w_1/w_2 from 1.00 to 0.25 decreases the core weight from about 49 to 19 pounds, or approximately 2/3. Larger changes are found for $|\Delta T_1| > 300^\circ$ F; smaller changes, for $|\Delta T_1| < 300^\circ$ F. Figure 10 also shows that, for a fixed heat-exchanger core weight, an increase in $|\Delta T_1|$ can be achieved by reducing the ratio w_1/w_2. For any heat-exchanger weight between 30 and 60 pounds, for example, approximately a 100° F increase in $|\Delta T_1|$ can be obtained by a reduction in w_1/w_2 from 1.00 to 0.25. Smaller increases in $|\Delta T_1|$ can be obtained for the same change in w_1/w_2 for core weights less than 30 pounds. For large changes in $|\Delta T_1|$, the increase in heat-exchanger weight rises rapidly with increasing values of w_1/w_2. These same results are shown in figure 11, a crossplot of figure 10 (abscissa and parameter are interchanged). Figure 11 also shows that as $|\Delta T_1|$ is increased changes in w_1/w_2 are much more influential on exchanger core weight.

In order to obtain some idea of the effect of heat-exchanger weight on specific engine weight, a single case is considered here. Assume that a heat exchanger is to be added to a 12,000-pound-thrust engine (at static sea-level conditions) with a specific engine weight of 0.300 pound per pound of thrust; this engine then weighs 3600 pounds. If a 60-pound heat exchanger is added to the engine, the specific weight of the engine is increased to only 0.305 pound per pound of thrust.

The heat-exchanger core weights presented do not include the weight of the ducting that might be required. This additional ducting weight must be considered before selection of any heat exchanger can be finalized; the ducting may, in some cases, weigh as much as or more than the heat-exchanger core proper. For the heat exchangers requiring the smaller temperature change in compressor bleed air ($\Delta T_1 = -100^\circ$ F), high Reynolds numbers are required for the prescribed pressure drop. This condition results in high Mach numbers, which in some cases produce choked flow within the heat exchanger. Because this is apparent only in the lighter-weight heat exchangers, no attempt was made to correct this condition by enlarging the free-flow area downstream of the inlet. Such a correction would in turn increase the weight of the heat exchanger slightly.

4073

Figures 12 and 13 are plots of $A_{F,2}$ similar to those of figures 10 and 11 for core weight. Figure 12 shows that, for a fixed ΔT_1, a decrease in w_1/w_2 results in an increase in $A_{F,2}$. This increase is more pronounced for values of $w_1/w_2 < 0.5$ than for values of $w_1/w_2 > 0.5$. For $\Delta T_1 = -300^\circ$ F, a reduction in w_1/w_2 from 1.00 to 0.5 increases $A_{F,2}$ from about 100 to 123 square inches; a further decrease in w_1/w_2 from 0.5 to 0.25 results in a further increase in $A_{F,2}$ from 123 to 178 square inches. Since the curves for the various values of ΔT_1 are of the same general trend, about the same order of increases in $A_{F,2}$ are found for the various values of ΔT_1. Figure 12 also shows that, if the frontal area $A_{F,2}$ is fixed, decreases in w_1/w_2 result in decreases in ΔT_1.

For the particular application under discussion, the frontal area $A_{F,2}$ and the weight are of primary importance; changes in frontal area $A_{F,1}$ are opposite to those of $A_{F,2}$, but not nearly as pronounced. Space limitations for this particular area are not of primary concern herein.

If, from the information contained herein, an exchanger suitable for a particular application from the point of view of weight can be found, the space available for installation may not be large enough to accomodate the accompanying exchanger lengths. It may be possible, however to install the exchanger in segments (fig. 14(a)) or to fold the heat exchanger over itself (fig. 14(b)). The latter arrangement results in a heat exchanger made up in layers. The additional pressure drops resulting from U-turning the air must therefore be considered in the calculations.

For all the examples presented, it was assumed that the quantity of compressor bleed air remained constant (2.70 lb/sec). This quantity, however, could vary if one should want to increase or decrease the temperature of the turbine cooling air (compressor bleed air) to maintain a constant blade temperature. From reference 2, it was possible to estimate that an increase in compressor bleed air of about 20 percent is required for a 100° F increase in turbine-cooling-air temperature (this is applicable to the cooling of two rotor and one stator blade row). This increase results in a required compressor bleed weight flow of 3.24 pounds per second. New heat-exchanger weights were calculated with this weight flow and a required reduction in compressor-bleed-air temperature of 200° F. The results are presented in figure 11 as a dashed line.

A comparison of these results with the results for the comparable compressor-bleed-air quantity of 2.70 pounds per second and 300° F temperature reduction shows that the weight of the heat-exchanger core is reduced about 50 to 60 percent by taking a smaller temperature reduction in the heat exchanger and increasing the compressor-bleed-air flow enough to maintain a constant blade temperature. This results in a change in specific engine weight of about 0.301 to 0.304 (12,000 pounds thrust, initial engine specific weight of 0.30, $w_1/w_2 = 1$).

Some increase in heat-exchanger weight can be tolerated without affecting the specific engine weight if the coolant flow can be reduced. Reference 7 shows that every percent of coolant-flow ratio (ratio of cooling air flow to compressor air flow) results in approximately 1 percent reduction in thrust for nonafterburning turbojet engines. As a consequence, the specific engine weight is increased 1 percent for each percent of coolant-flow ratio. At the flight conditions considered, the change in cooling-air weight flow from 2.70 to 3.24 pounds per second corresponds to changing the coolant-flow ratio from 0.054 to 0.065 (compressor air flow, 49.8 pounds per second), which in turn results in a 1.1 percent increase in specific engine weight, neglecting the effects of heat-exchanger weight. For the previously assumed engine weight of 3600 pounds (12,000-pound sea-level static dry thrust and specific engine weight of 0.300) and a maximum possible saving in heat-exchanger core weight of 30 pounds (fig. 11), the saving in engine weight is less than 1 percent by using the lighter heat exchanger with the smaller temperature reduction and consequently a higher required cooling-air flow. This weight saving does not quite compensate for the specific engine weight increase of 1.1 percent resulting from operating with the higher coolant-flow ratio. For this case, therefore, it appears advantageous to use a larger heat exchanger to reduce the required coolant flow. The weight difference between the two cases considered is small, however.

It is also possible from figure 11 to determine how heat-exchanger weight is affected by compressor-bleed weight flow for a constant temperature reduction by comparing the curves for $\Delta T_1 = -200^\circ$ F for compressor bleed flows of 2.70 and 3.24 pounds per second. It can be seen that the heat-exchanger core weights increase almost proportionally to the compressor-bleed weight flow. A 20-percent increase in flow results in a 20-percent increase in weight.

SUMMARY OF RESULTS

Calculations were made of crossflow heat-exchanger core sizes and weights for possible use in an engine flying at a Mach number of 2.5 and and altitude of 70,000 feet for a compressor-bleed-air flow rate of 2.70 pounds per second. Ducting weights were not included in the calculations. A range of temperature changes in the compressor bleed air and a range of ratios of compressor-bleed to ram-air weight flows were considered. Pressure drops for the two fluids and the inlet states of each fluid were prescribed. The results of the calculations are summarized as follows:

1. For small values of compressor-bleed-air temperature changes, the required heat-exchanger core weights appear to be tolerable. For large values of the bleed-air temperature change, heat-exchanger core weights increase appreciably. A 60-pound heat exchanger increased the specific engine weight of a 12,000-pound-thrust engine (static sea-level conditions) from 0.300 to 0.305 pound per pound of thrust.

2. For a fixed value of compressor-bleed-air temperature change, the heat-exchanger core weight decreases with decreasing values of the ratio of compressor-bleed to ram-air weight flows. For example, for a compressor-bleed-air temperature change of -300° F, a reduction in the weight-flow ratio from 1.00 to 0.25 reduces the exchanger core weight from about 49 to 19 pounds.

3. For a fixed heat-exchanger core weight, an increase in compressor-bleed-air temperature change can be obtained by decreasing the compressor-bleed to ram-air weight-flow ratio. For heat exchangers of moderate weight (30 to 60 pounds), the compressor-bleed-air temperature change can be increased by about 100° F if the weight-flow ratio is reduced from 1.00 to 0.25.

4. Increases in frontal area in the primary-fluid direction result from decreases in the weight-flow ratio for a fixed compressor-bleed-air temperature change.

5. For the application under discussion, space limitations may not permit the use of exchanger core lengths found herein, so the exchangers may have to be broken into segments or folded into layers. The pressure losses connected with these types of installations are not considered herein.

6. Weights presented herein are for the exchanger core only. Ducting weights must be considered before the selection of any exchanger can be finalized.

7. Increasing the turbine cooling-air quantity to a value comparable to a decrease in temperature of an additional 100° F results in a reduction of weight of the heat-exchanger core of about 50 to 60 percent. However, this saving in weight may not compensate for increased specific engine weight because of the additional compressor-bleed-air flow requirement.

Lewis Flight Propulsion Laboratory
National Advisory Committee for Aeronautics
Cleveland, Ohio, May 23, 1956

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4073

CV-3

TABLE I. - HEAT-EXCHANGER CORE, SIZES, AND WEIGHTS

$-\Delta T_1$, °F	w_1/w_2	l_2, in.	l_n, in.	$A_{F,1}$, sq in.	l_1, in.	$A_{F,2}$, sq in.	Weight, lb
100	1.00	5.25	5.23	27.48	8.19	42.86	3.71
	.75	4.10	6.54	26.80	7.74	50.59	3.42
	.50	2.80	9.30	26.05	7.28	67.74	3.12
200	1.00	13.35	3.10	41.46	20.57	63.89	14.07
	.75	10.00	3.93	39.34	18.65	73.36	12.10
	.50	5.58	5.54	37.04	16.65	92.32	10.17
	.25	3.60	9.62	34.66	14.74	141.93	8.43
300	1.00	29.75	2.11	62.89	46.90	99.14	48.65
	.75	19.95	2.78	55.42	37.40	103.89	34.19
	.50	12.45	3.99	49.70	30.95	123.55	25.36
	.25	6.50	6.91	44.92	25.80	178.32	19.11
400	.75	44.80	1.91	85.66	86.28	161.96	121.88
	.50	24.30	2.87	69.81	60.03	172.16	69.12
	.25	11.55	5.12	59.22	44.90	230.24	43.86

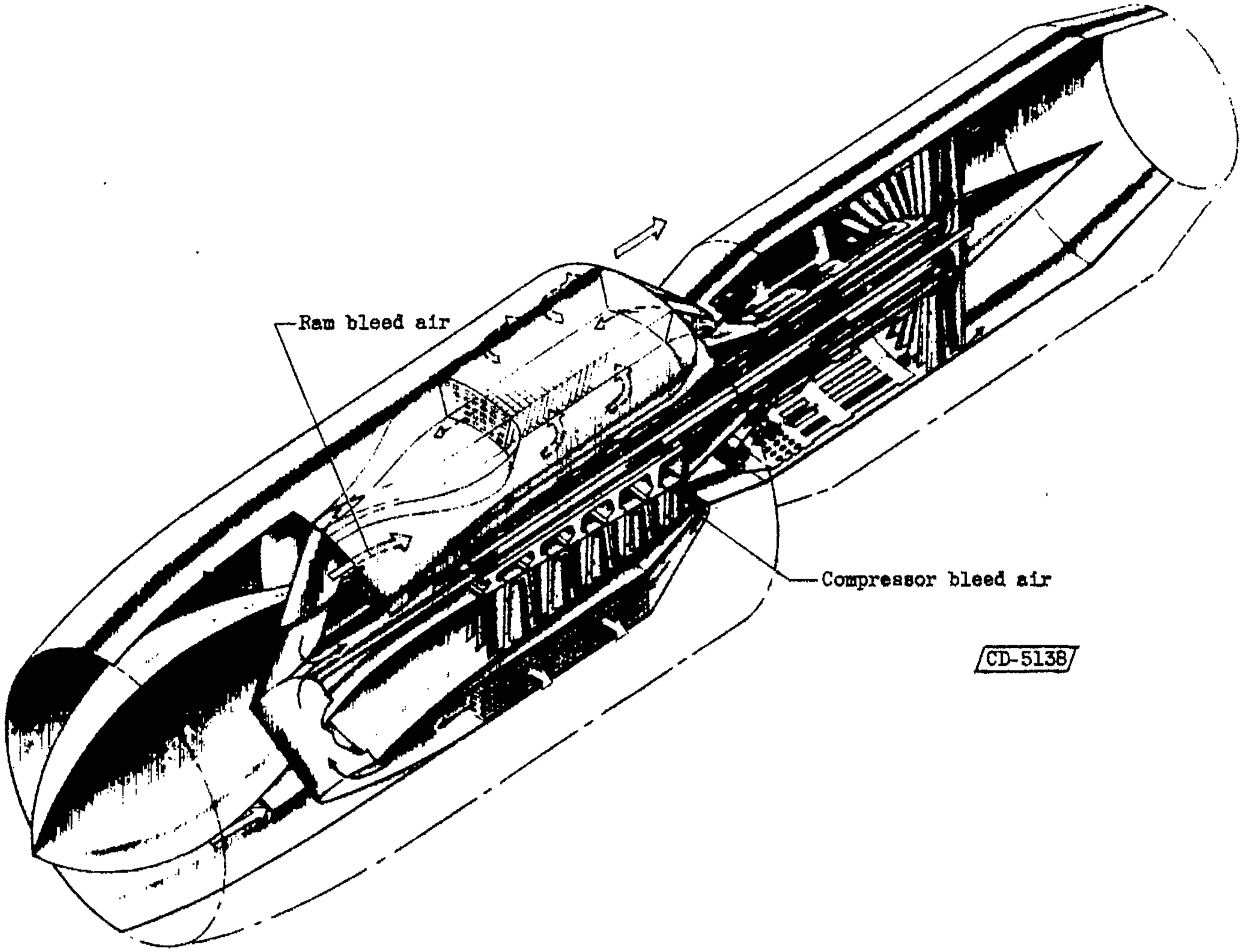

Figure 1. - Heat-exchanger installation on jet engine.

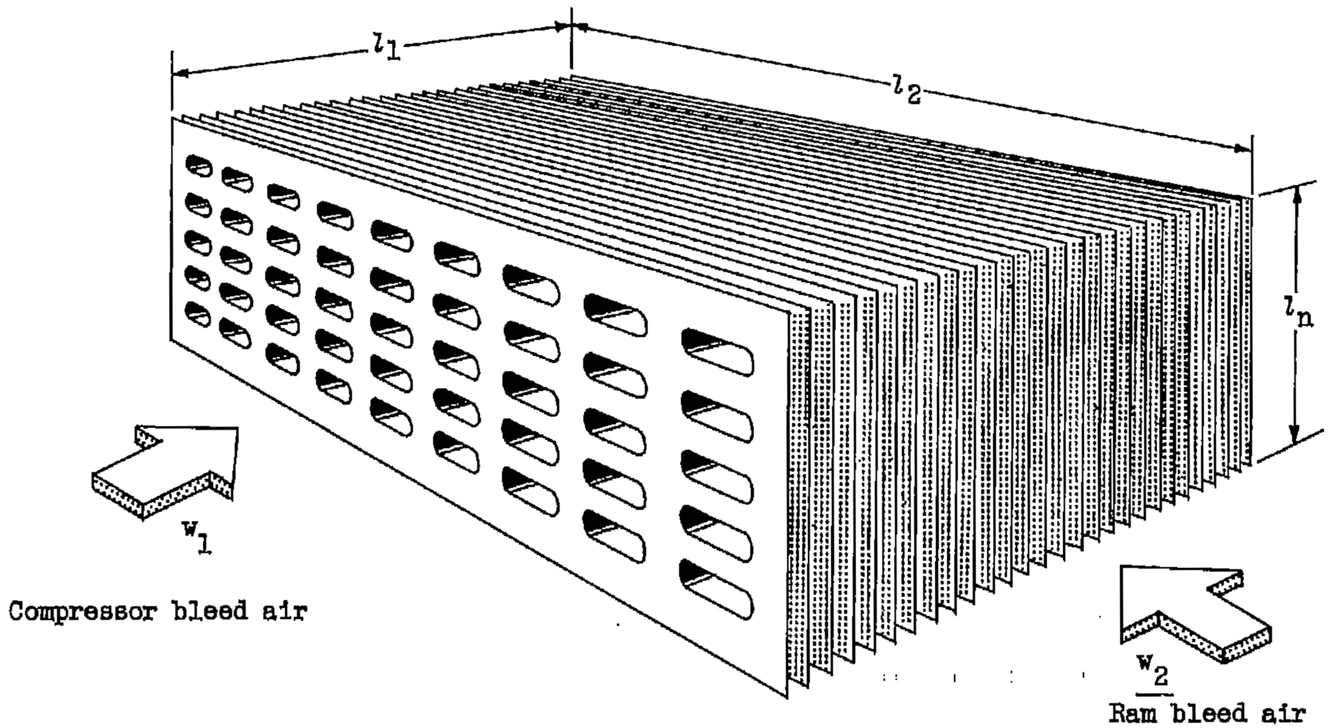

Figure 2. - Crossflow finned-tube heat-exchanger core.

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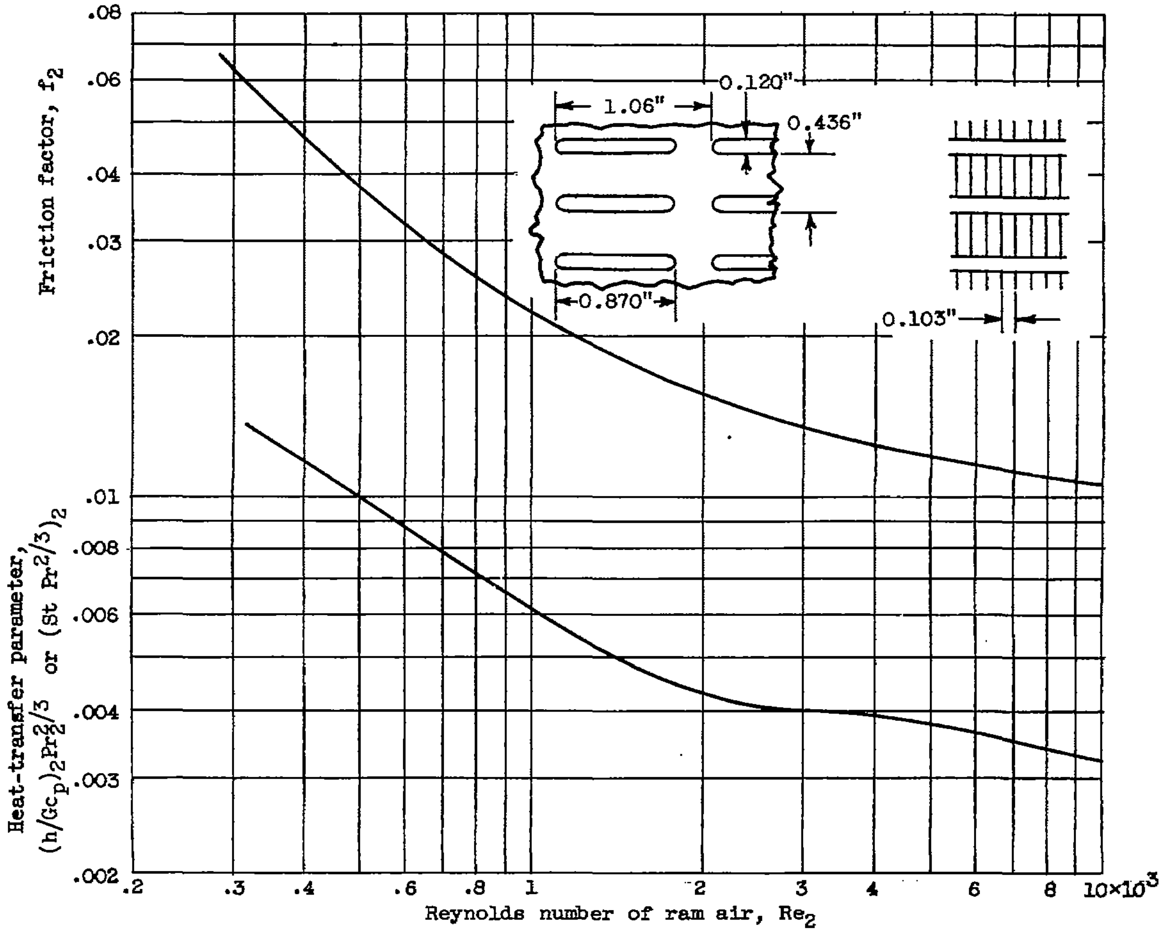

Figure 3. - Heat-transfer and friction data for finned-tube heat-exchanger core (ref. 5).

Fin area/total area, A_f/A	0.795
Fin pitch, 1/in.	9.681
Fin thickness, t_f, in.	0.004
Hydraulic diameter, fin side, d_2, ft	0.0118
Hydraulic diameter, tube side, d_1, ft	0.018
Free-flow area/frontal area, fin side, σ_2	0.697
Free-flow area/frontal area, tube side, σ_1	0.219
Heat-transfer area/volume, fin side, α_2, sq ft/cu ft	229
Heat-transfer area/volume, tube side, α_1, sq ft/cu ft	48.76

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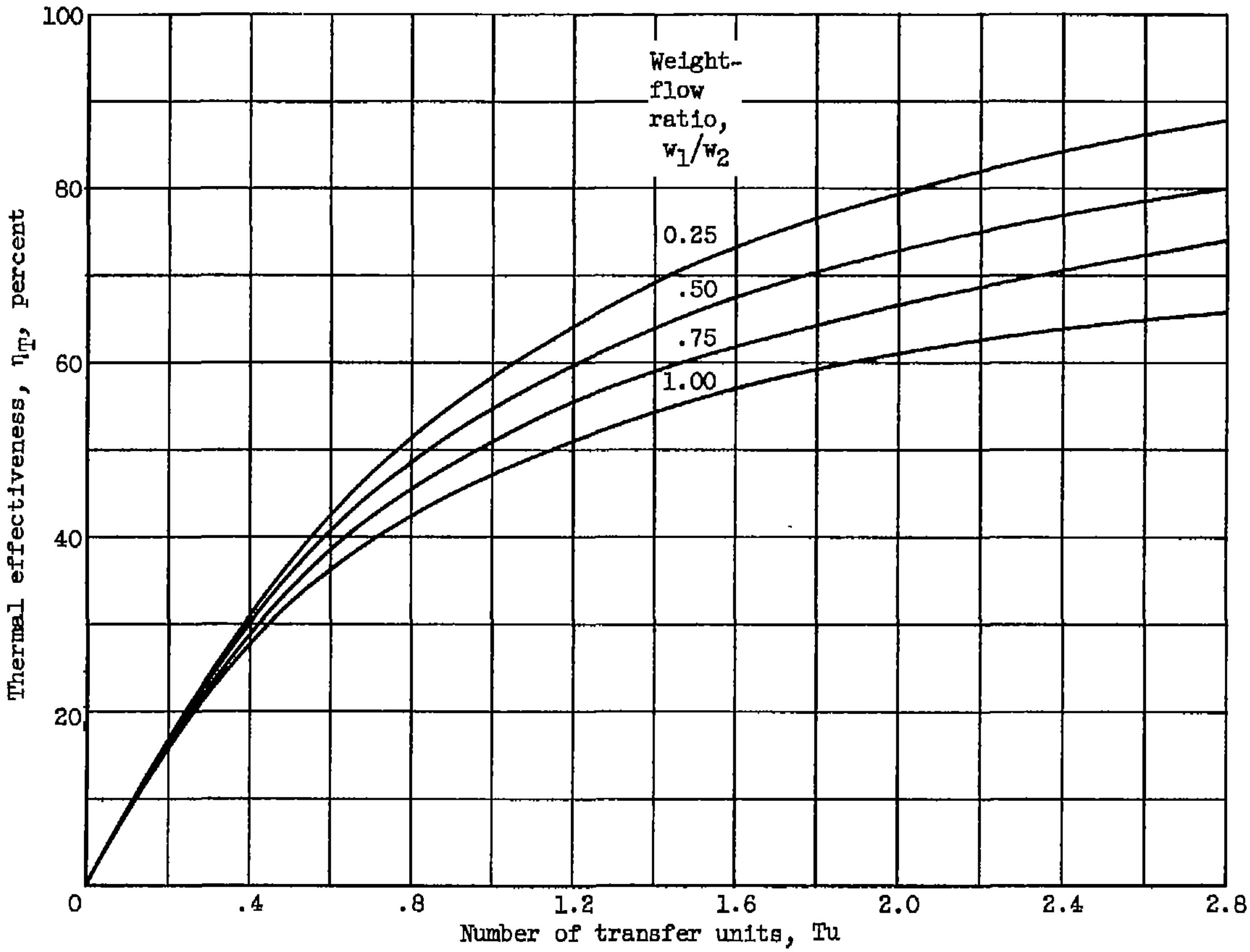


Figure 4. - Performance of crossflow heat exchanger with fluids unmixed. (Specific heat at constant pressure assumed constant.)

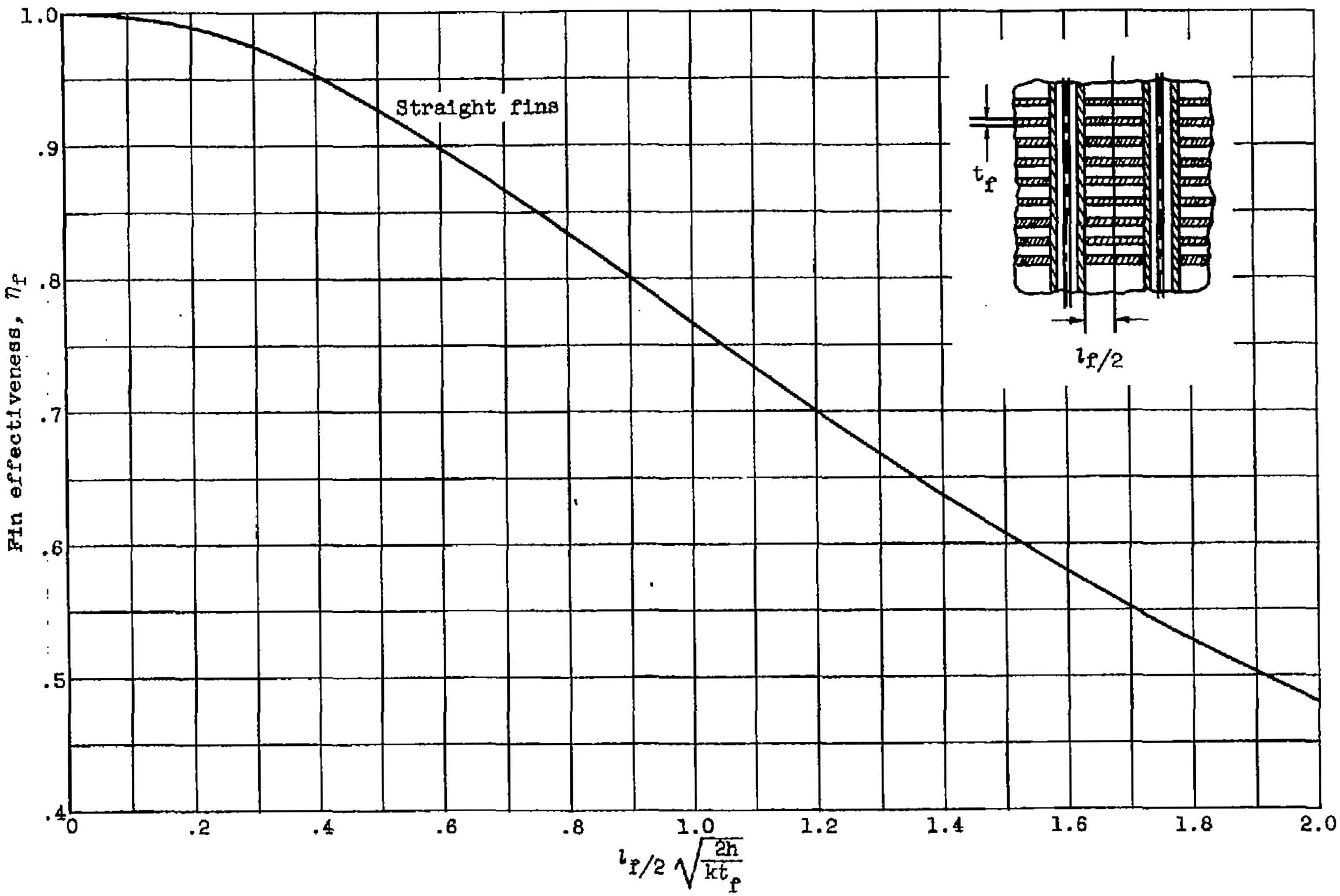


Figure 5. - Effectiveness for straight fins.

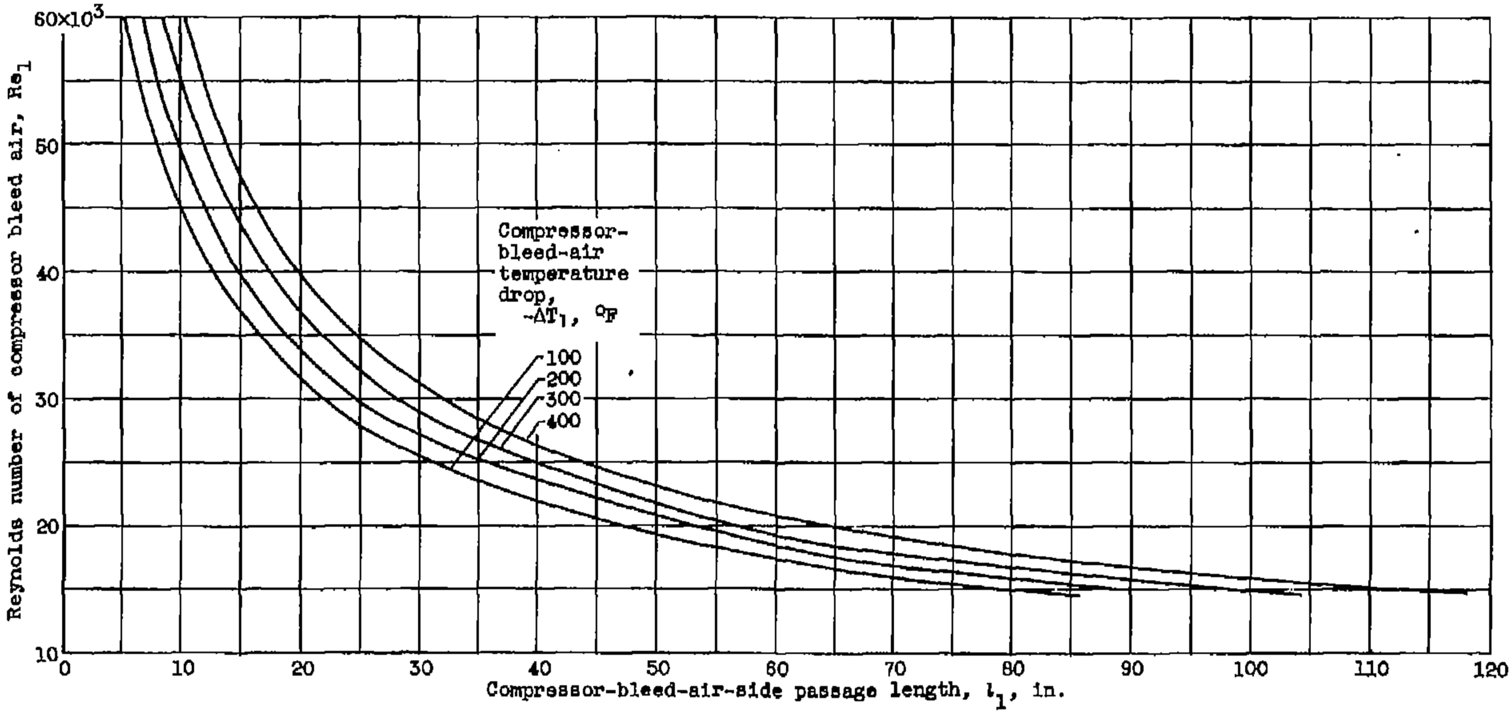
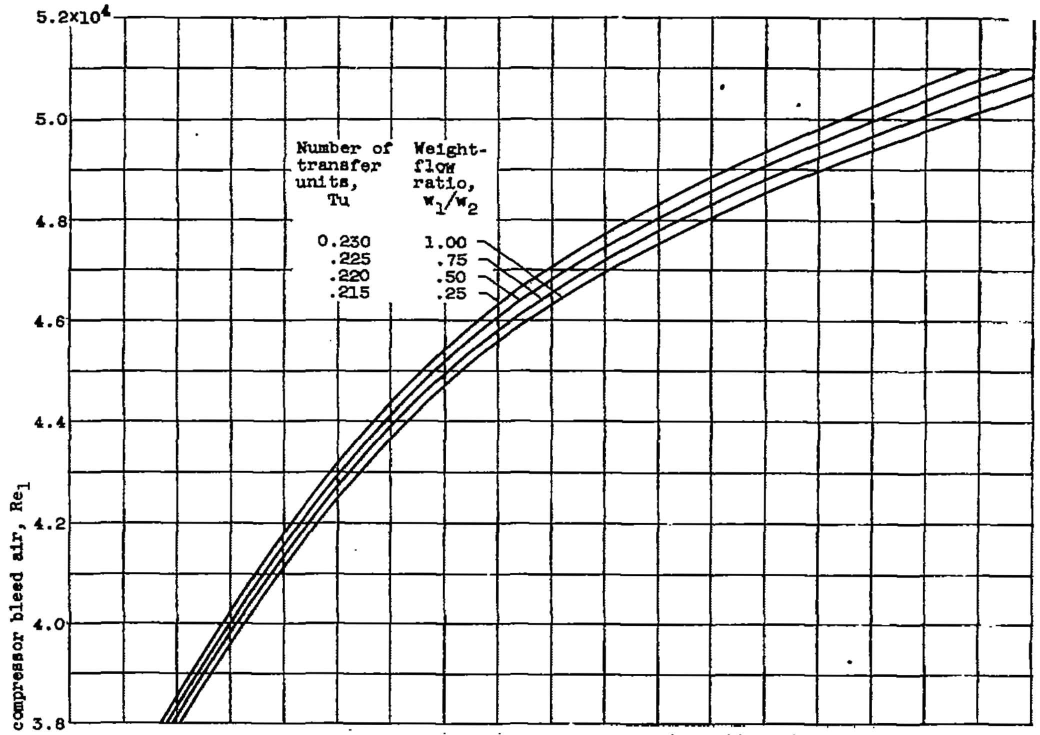
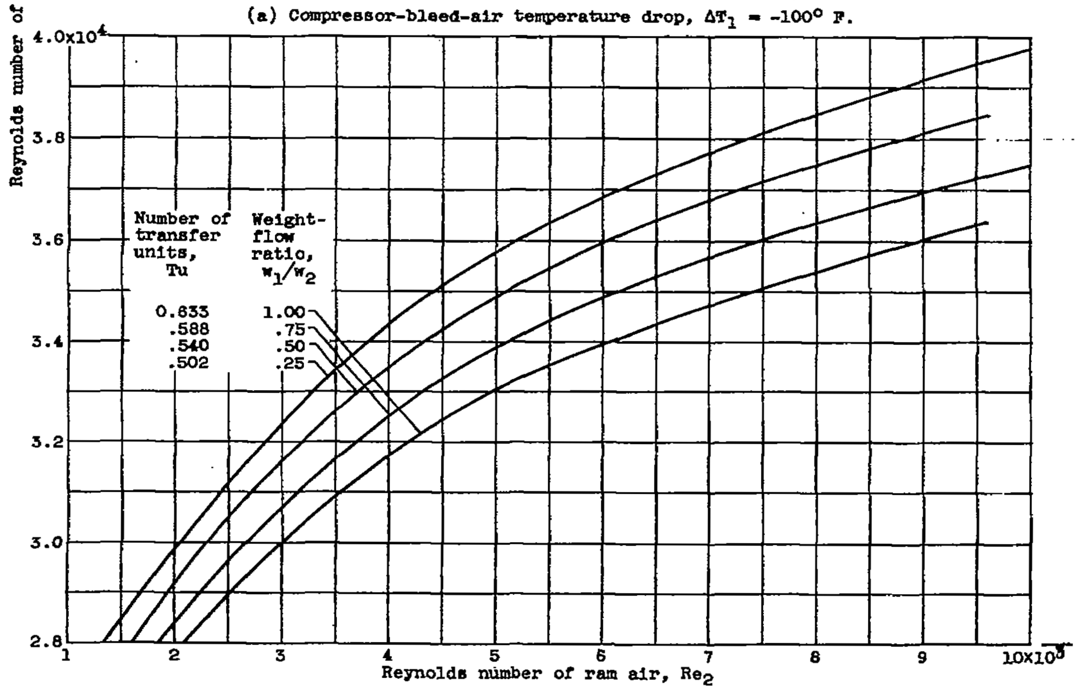


Figure 6. - Graphical representation of equation (3) for inlet state, temperature and pressure drops, and flow characteristics corresponding to compressor bleed air. Pressure drop, 1000 pounds per square foot; inlet pressure, 5300 pounds per square foot; inlet temperature, 1410° R; ratio of free-flow to frontal area, 0.219; hydraulic diameter, 0.018 foot.



(a) Compressor-bleed-air temperature drop, $\Delta T_1 = -100^{\circ}$ F.

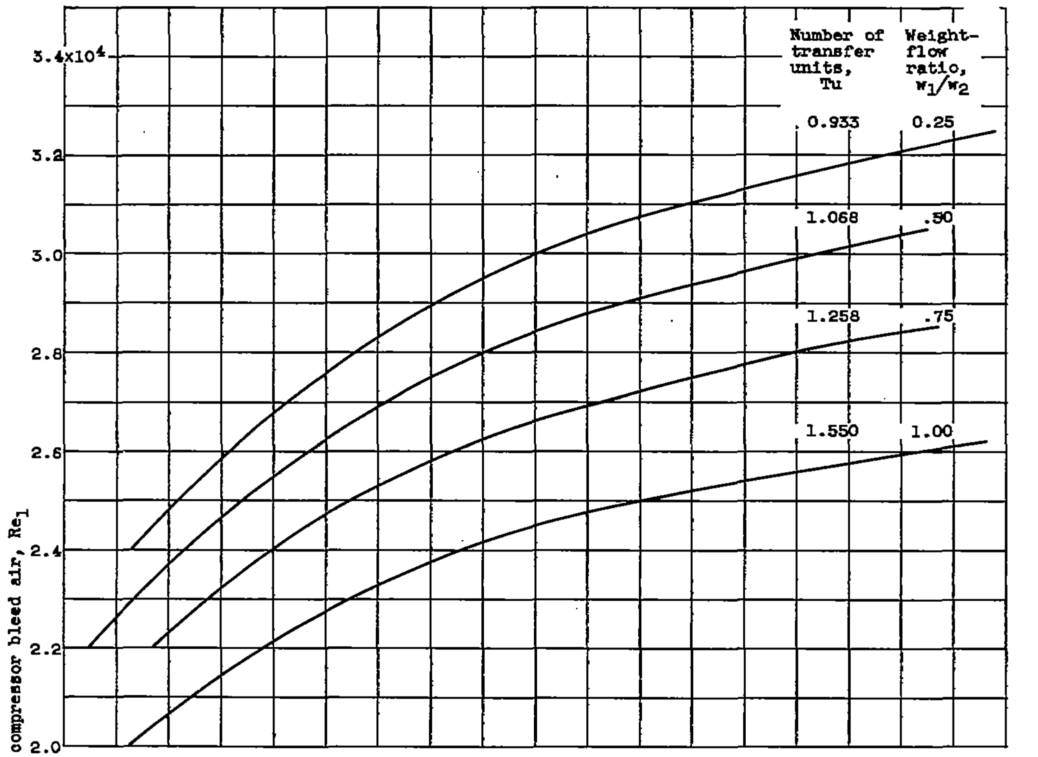


(b) Compressor-bleed-air temperature drop, $\Delta T_1 = -200^{\circ}$ F.

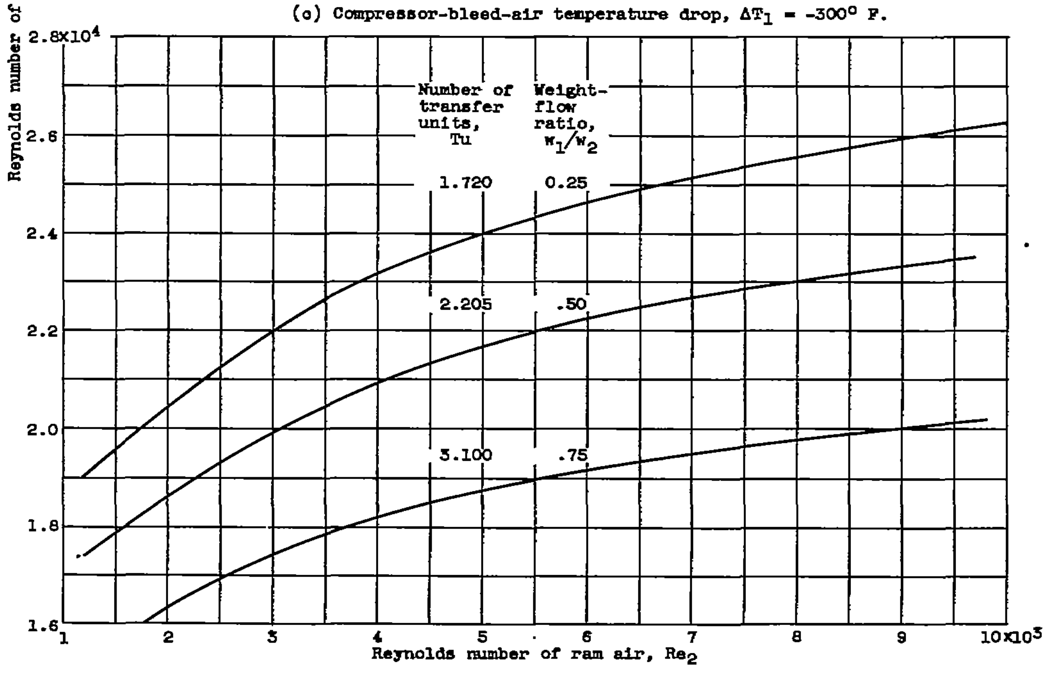
Figure 7. - Graphical representation of equation (1) for prescribed heat-transfer characteristics.

4073

CV-4



(c) Compressor-bleed-air temperature drop, $\Delta T_1 = -300^\circ$ F.



(d) Compressor-bleed-air temperature drop, $\Delta T_1 = -400^\circ$ F.

Figure 7. - Concluded. Graphical representation of equation (1) for prescribed heat-transfer characteristics.

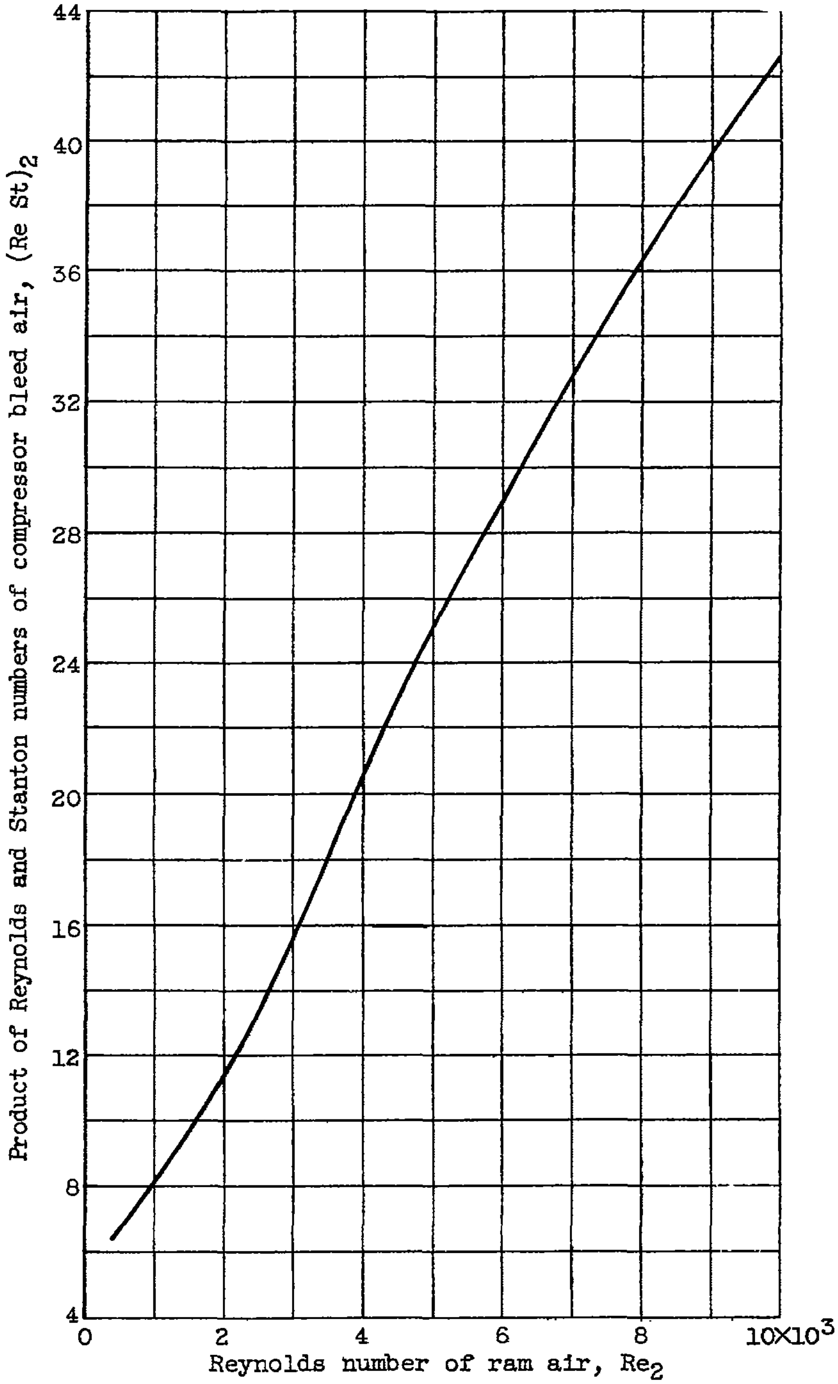


Figure 8. - Rearrangement of heat-transfer characteristics of finned-tube configuration. Prandtl number $Pr^{2/3} = 0.75$.

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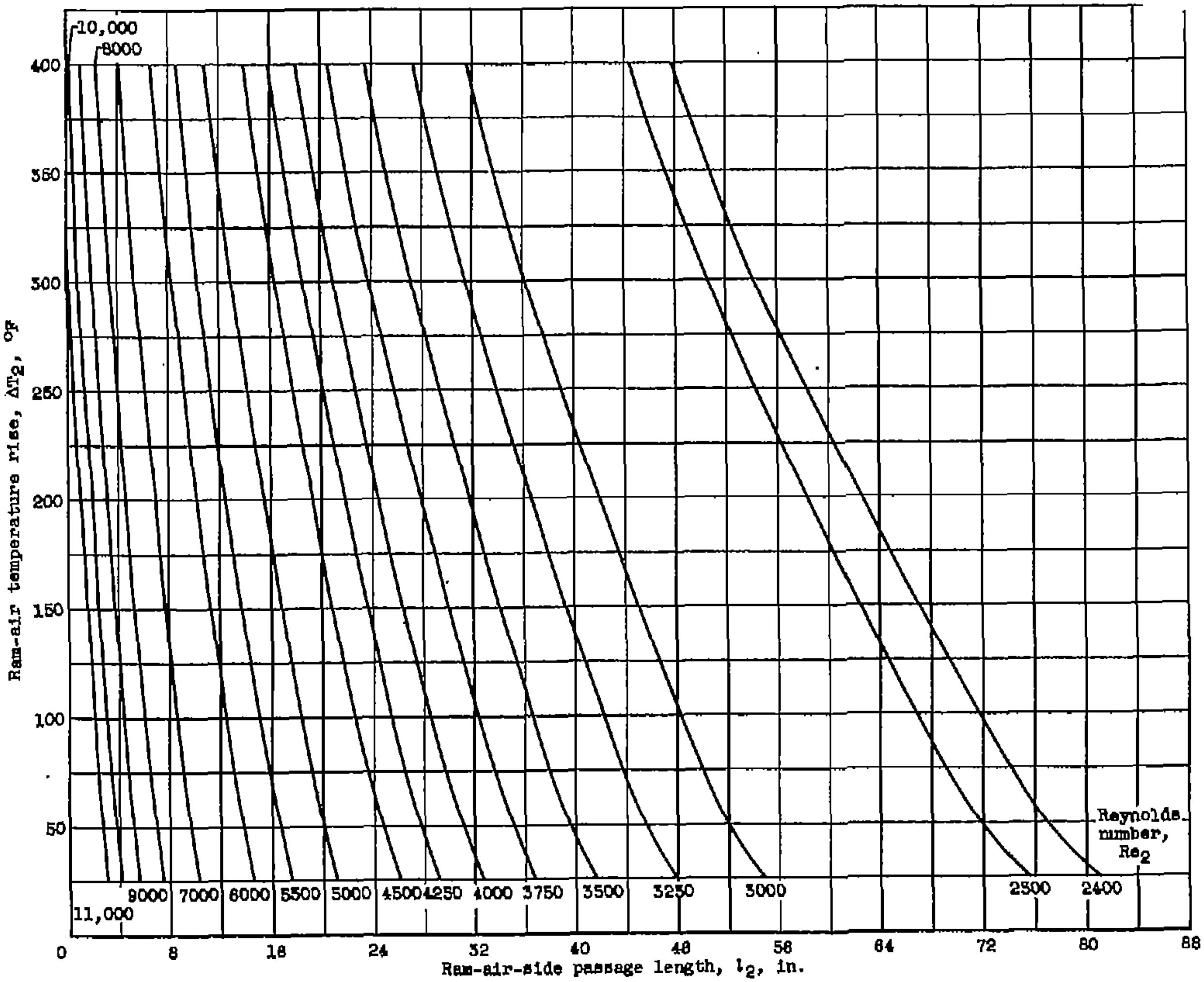

Figure 9. - Graphical representation of equation (3) for inlet-state; temperature and pressure drops and flow characteristics corresponding to ram air bleed. Ram-air pressure drop, 400 pounds per square foot; inlet pressure, 1080 pounds per square foot; inlet temperature, 860° R; ratio of free-flow to frontal area, 0.697; hydraulic diameter, 0.0118 foot.

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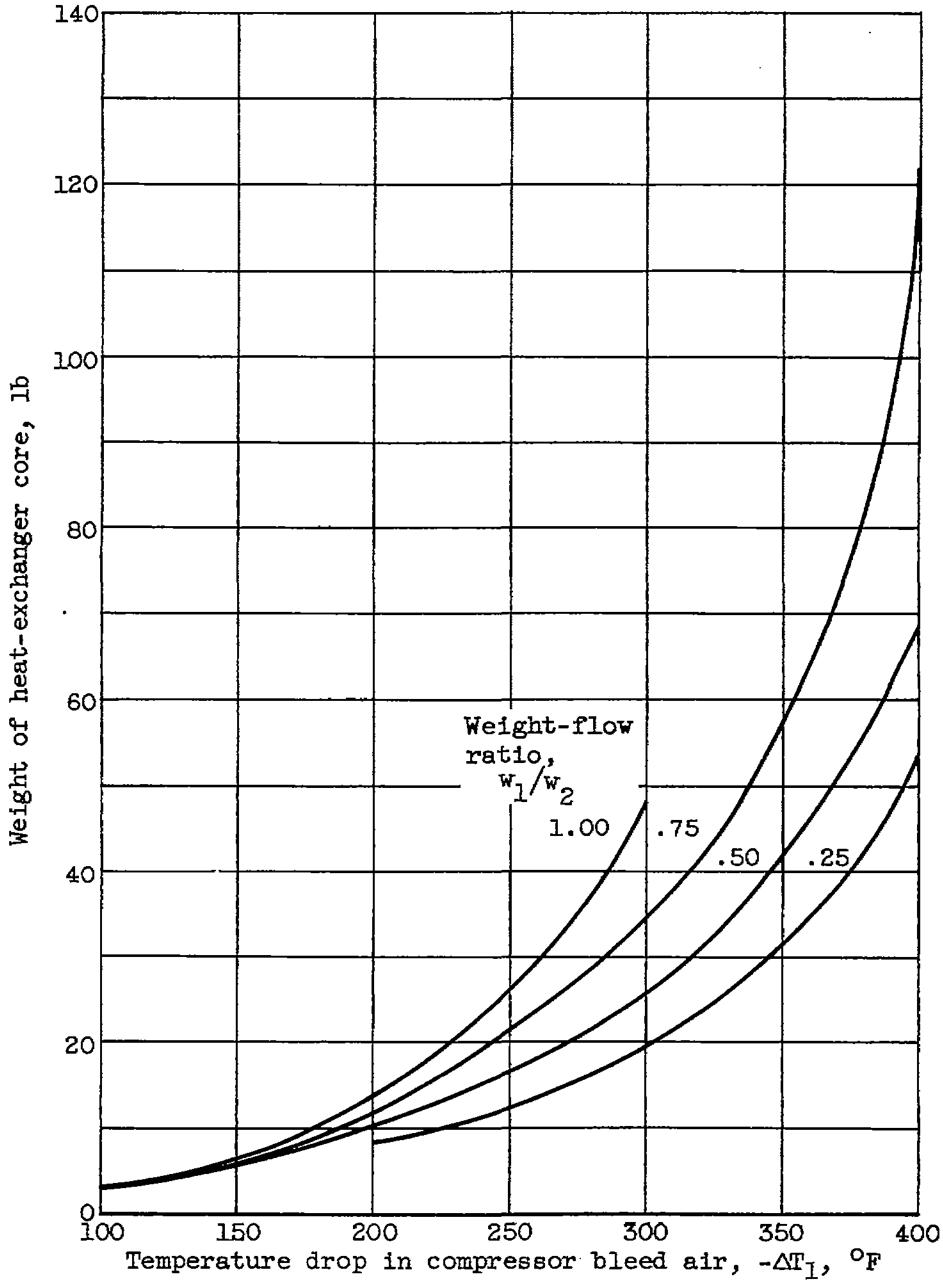

Figure 10. - Weight of heat-exchanger core against temperature drop in compressor bleed air. Compressor-bleed-air weight-flow rate, 2.70 pounds per second; metal thickness, 0.004 inch; density of metal, 0.300 pound per cubic inch.

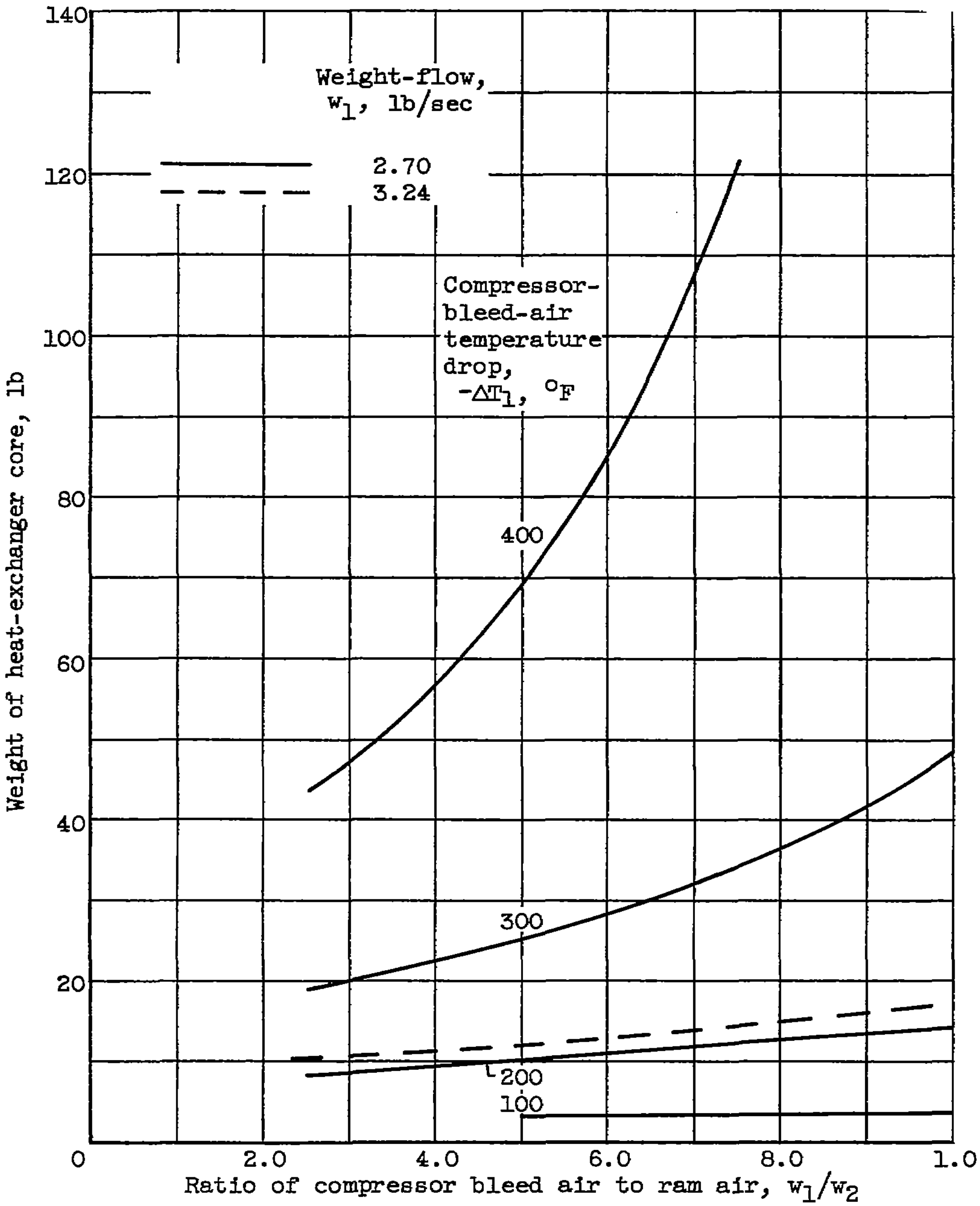

Figure 11. - Weight of heat-exchanger core against rate of compressor bleed air to ram bleed air. Compressor-bleed-air weight-flow rate, 2.70 pounds per second; metal thickness, 0.004 inch; density of metal, 0.300 pound per cubic inch.

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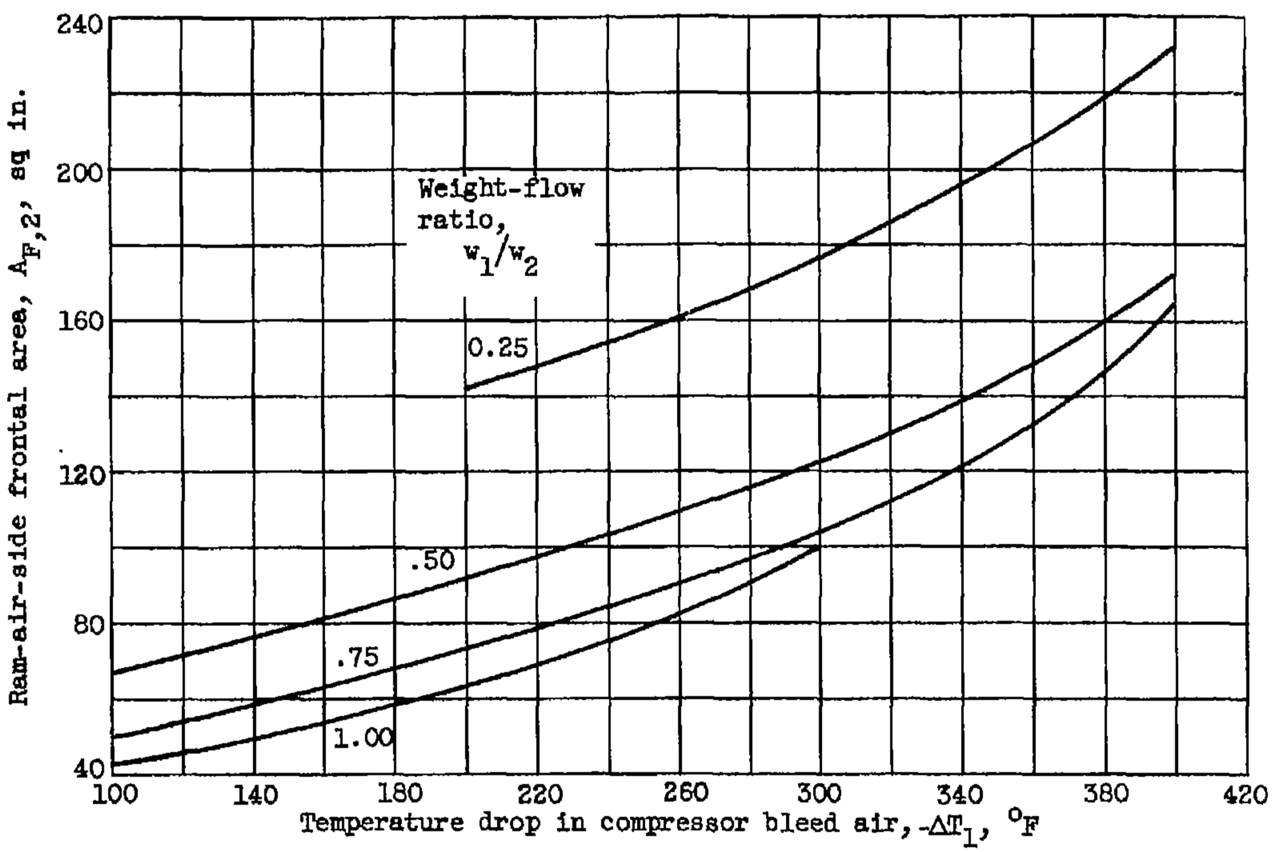


Figure 12. - Ram-air-side frontal area against temperature drop of compressor bleed air. Compressor-bleed-air weight-flow rate, 2.70 pounds per second; ratio of free-flow to frontal area, 0.697.

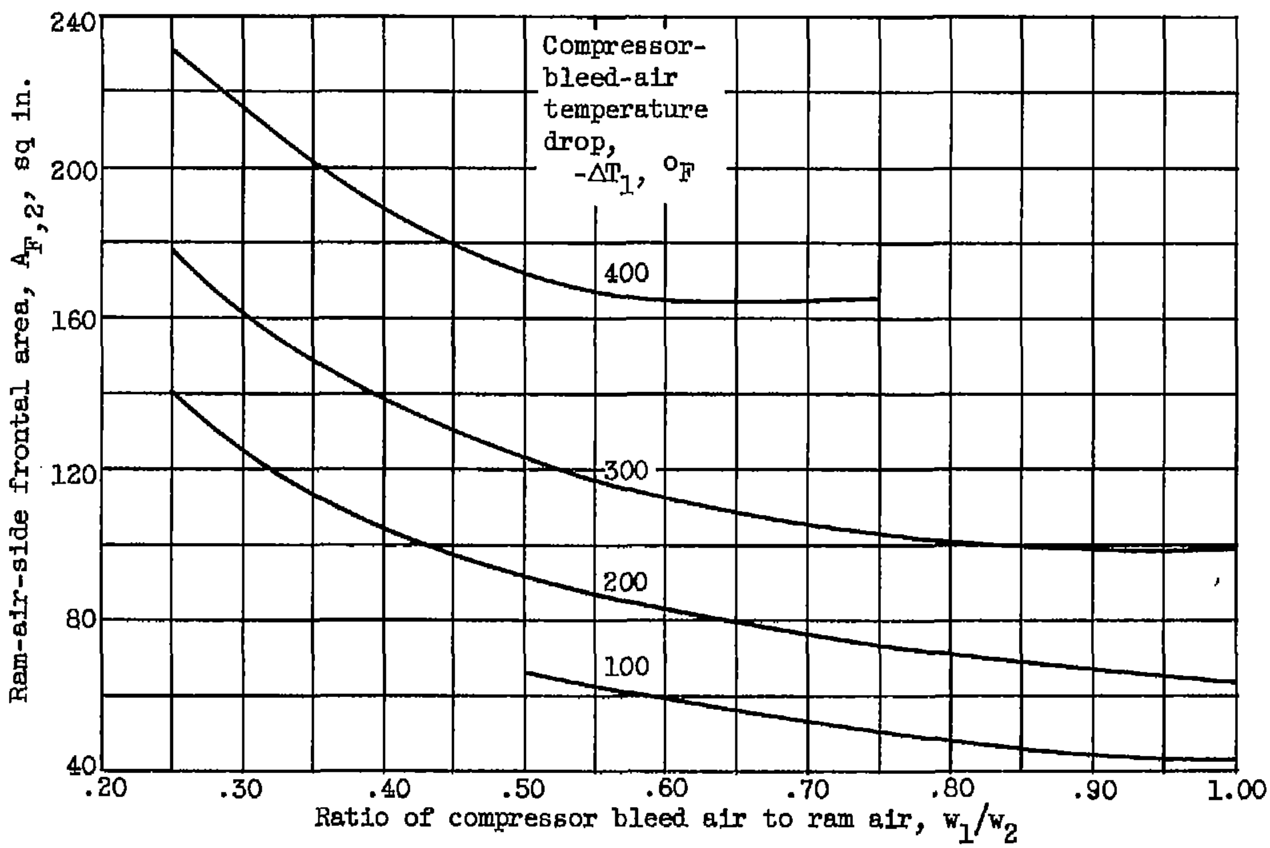
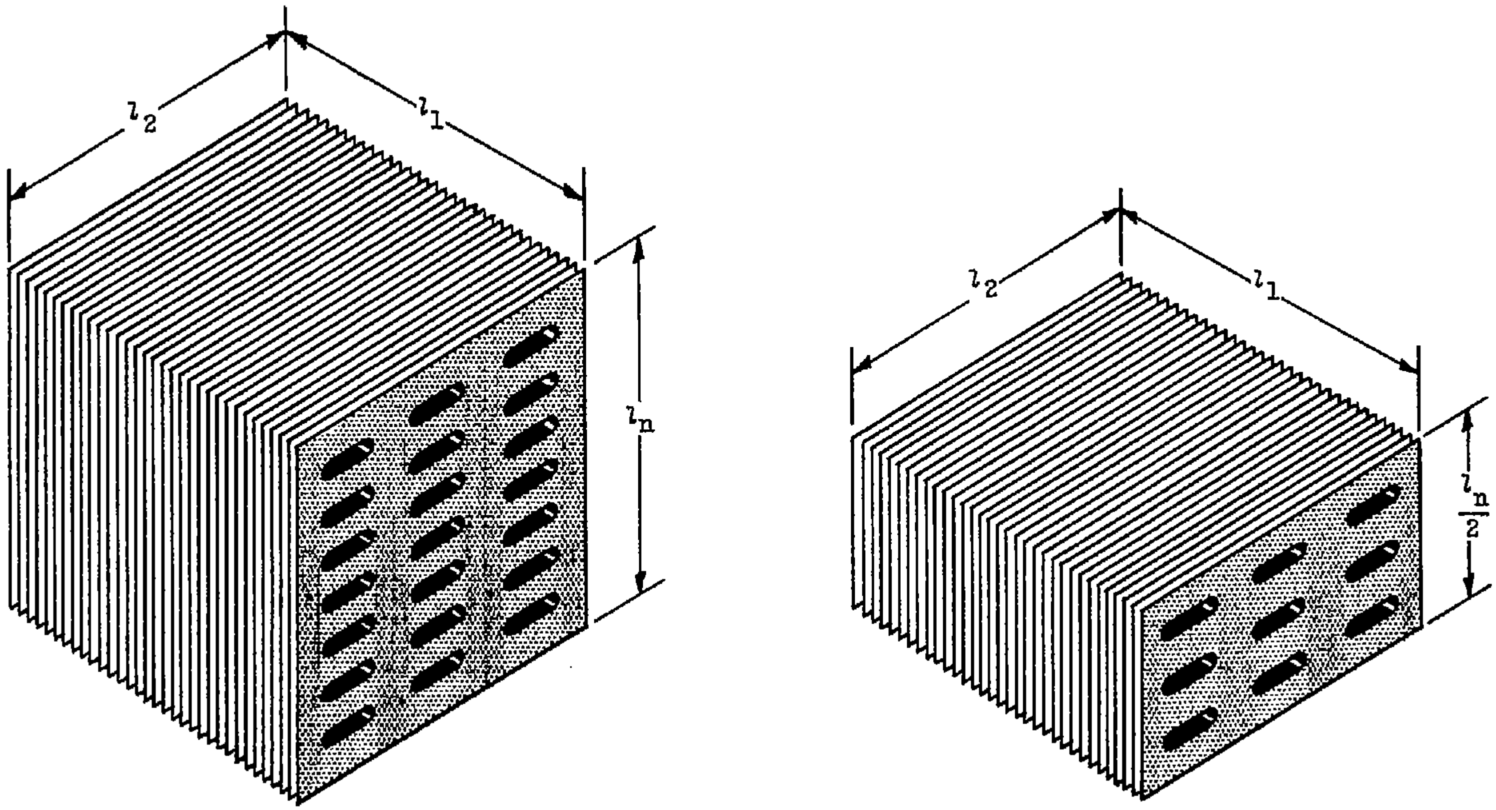


Figure 13. - Ram-air-side frontal area against ratio of compressor bleed air to ram air bleed. Compressor-bleed-air weight-flow rate, 2.70 pounds per second; ratio of free-flow to frontal area, 0.697.



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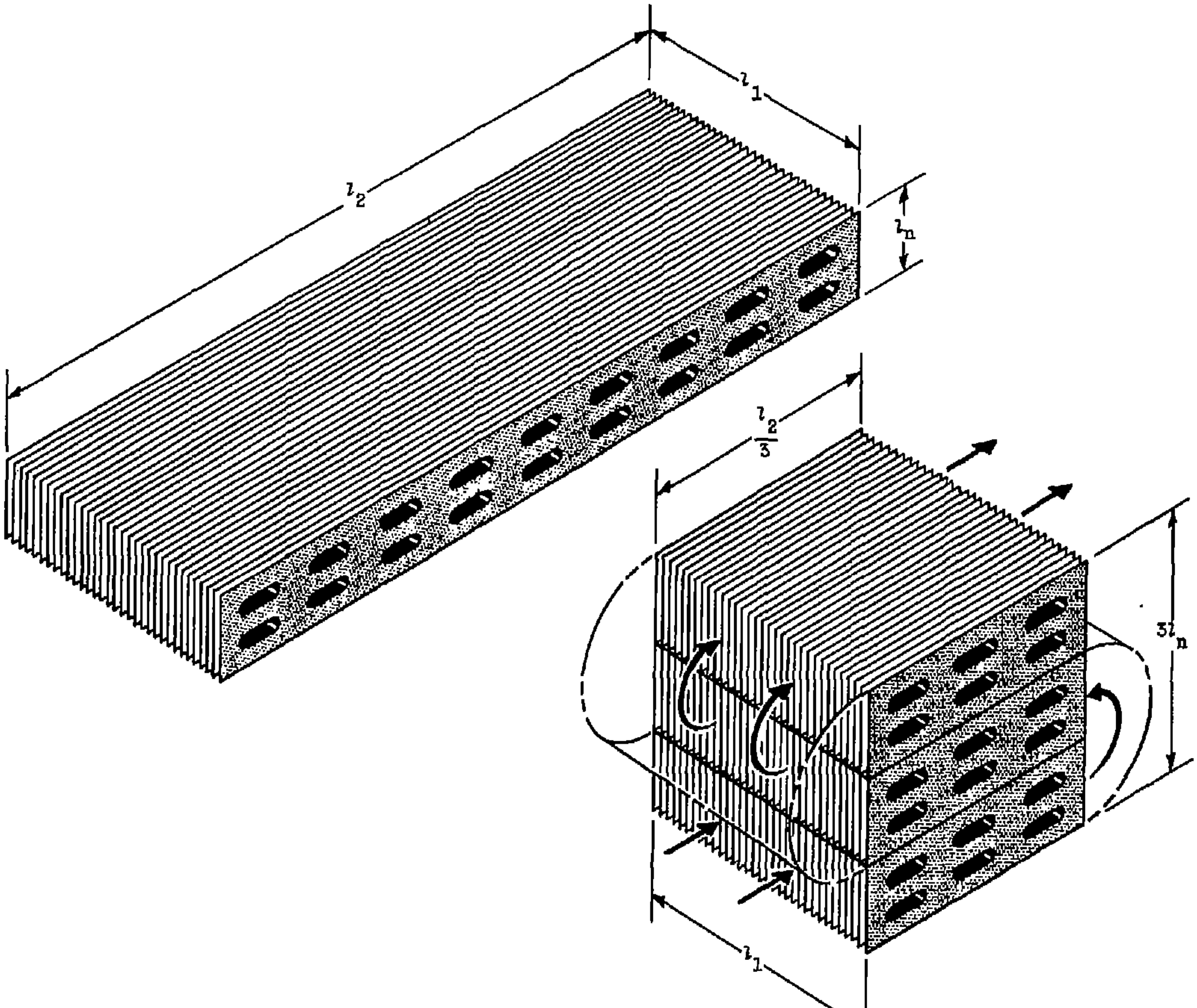
(a) Heat-exchanger core divided into segments.

Figure 14. - Heat-exchanger core arrangement.



(b) Heat-exchanger core folded into layers.

Figure 14. - Concluded. Heat-exchanger core arrangement.

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