JAN 1 ? 1947

ARR No. 5A12

NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS

WARTIME REPORT

ORIGINALLY ISSUED

April 1945 as
Advance Restricted Report 5A12

AN INVESTIGATION OF AIRCRAFT HEATERS

XXIII - MEASURED AND PREDICTED PERFORMANCE OF A FLAT-PLATE TYPE EXHAUST GAS AND AIR HEAT EXCHANGER

By L. M. K. Boelter, A. G. Guibert,
J. M. Rademacher, and V. D. Sanders
University of California

NACA

WASHINGTON

NACA ARR No. 5A12

NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS

ADVANCE RESTRICTED REPORT

AN INVESTIGATION OF AIRCRAFT HEATERS

XXIII - MEASURED AND PREDICTED PERFORMANCE OF A FLAT-PLATE TYPE EXHAUST GAS AND AIR HEAT EXCHANGER

By L. M. K. Boelter, A. G. Guibert,
J. M. Rademacher, and V. D. Sanders

SUMMARY

Results of tests of the thermal performance and the static pressure drop characteristics of a flat-plate type exhaust gas and air heat exchanger are presented. The ventilating air shroud built into this heat exchanger gives characteristics of cross flow and parallel flow.

The range of the exhaust gas weight rates used in tests for the determination of the heat transfer rates was from about 1700 lb/hr to about 5200 lb/hr. The temperature of the exhaust gas was maintained at approximately 1600° F.

Static pressure drop measurements were made across both sides of the heat exchanger under isothermal and non-isothermal conditions. Heater surface temperatures at the entrance and exit of the heat exchanger were recorded.

Predictions of the thermal outputs and pressure drops are compared with the experimentally determined values.

INTRODUCTION

An investigation of the performance characteristics of this flat-plate heat exchanger, designed for use in the exhaust gas streams of aircraft engines for cabin heating systems and for wing and tail surface anti-icing systems was undertaken, using the large test stand of the Mechanical Engineering laboratories of the University of California.

The following data were obtained:

1. Weight rates of exhaust gas and ventilating air

2. Temperatures of the exhaust gas and ventilating air at the inlet and outlet of the heat exchanger

3. Static pressure drops across both sides of the heat exchanger under isothermal and non-isothermal conditions

4. Surface temperatures of the heat exchanger at the inlet and outlet of the ventilating air side of the heat exchanger

This investigation, part of a research program conducted on aircraft heat exchangers at the University of California, was sponsored by and conducted with the financial assistance of the National Advisory Committee for Aeronautics.

SYMBOLS

A area of heat transfer, ft^2

A_e area of heat transfer of "equivalent" cylindrical ends (see fig. 1a) of fluid passages on either side of the heater, ft^2

A_f area of heat transfer of flat plate section of fluid passages on either air or gas side of heater, ft^2

A_h cross-sectional area of flow for either fluid measured within heater, ft^2

A_1 cross-sectional area of flow at inlet pressure measuring station, ft^2

A_2 cross-sectional area of flow at outlet pressure measuring station, ft^2

D hydraulic diameter, ft

D_o diameter of equivalent cylindrical ends of fluid passage on either side of heater, ft

f_c unit thermal convective conductance (average with length), Btu/hr ft^2 $^{\circ}$F

f_{ce} unit thermal convective conductance of fluid flowing over equivalent cylindrical ends on either air or gas side of heater (average with length), Btu/hr ft^2 $^{\circ}$F

f_{cf} unit thermal convective conductance of fluid flowing over flat plate section of passages on either side of heater (average with length), Btu/hr ft^2 $^{\circ}$F

f_cA thermal convective conductance of either fluid, Btu/hr $^{\circ}$F

F_a tube arrangement modulus for in-line tube banks

g gravitational force per unit of mass, lb/(lb sec^2/ft)

G weight rate of fluid per unit of area, lb/hr ft^2

G_o weight rate of fluid per unit of area, based on minimum area, lb/hr ft^2

K isothermal head loss coefficient defined by equation

$$\frac{\Delta P}{\gamma} = K \frac{{u_m}^2}{2g}$$

l length of a duct measured from entrance, ft

m ratio of cross-sectional area of flow before expansion of fluid passage to that after expansion of fluid passage

P_w wetted perimeter of a duct, ft

q measured rate of enthalpy change of either fluid, Btu/hr or k Btu/hr (1000 Btu/hr)

T_{av} arithmetic average mixed-mean absolute temperature of fluid, $\frac{T_1 + T_2}{2}$, $^{\circ}$R

T mixed-mean absolute temperature of the fluid, $^{\circ}$R

T_f arithmetic average of the absolute temperature of the surface and of the absolute mixed-mean temperature of the fluid, $^{\circ}$R

T_{iso} mixed-mean absolute temperature of the fluid for the isothermal pressure drop tests, °R

u_m mean velocity of fluid within fluid passage, ft/sec

U over-all unit thermal conductance, Btu/hr ft² °F

UA over-all thermal conductance, Btu/hr °F

W weight rate of fluid, lb/hr

γ weight density of fluid, lb/ft³

ΔP static pressure drop, lb/ft²

$\Delta P''$ static pressure drop, in. H_2O

$\Delta \tau_{mp}$ mean temperature difference for parallel flow of fluids, °F

ζ isothermal friction factor defined by equation

$$\frac{\Delta P_{fric}}{\gamma} = \zeta \frac{l}{D} \frac{u_m^2}{2g}$$

τ mixed-mean temperature of fluid, °F

Subscripts

a ventilating air side

b bends (ΔP_b, K_b)

c convective conductance (f_c, etc.) and also sudden contraction (ΔP_c, K_c)

e equivalent cylindrical ends of passages (A_e, f_{ce})

f flat plate section of passages on either side of heater (A_f, f_{cf}) and also fluid "film" (T_f)

g exhaust gas side

h heater (A_h)

m mean values at any section of the heater (u_m)

o maximum values (G_o) and also outer diameter of a cylinder (D_o)

p	parallel flow
av	arithmetic average (T_{av})
exp	sudden expansion (ΔP_{exp}, K_{exp})
fric	friction
iso	isothermal conditions
non-iso	non-isothermal conditions
1	point 1, entrance section
2	point 2, exit section

DESCRIPTION OF HEATER AND TESTING PROCEDURE

The flat-plate heater tested is a parallel-flow unit with a total of twenty alternate exhaust gas and ventilating air passages. At the entrance and exit of the ventilating air side, characteristics of crossflow prevail. (See photographs, fig. 2 to 4, diagrammatic sketch of fig. 1, and also cut-away view of heater in fig. 1a.) The diagrammatic sketch in figure 1 gives all pertinent dimensions and the approximate location of the thermocouples used for determining the surface temperatures. A schematic diagram of the test setup for this heat exchanger, showing the distances from all temperature and pressure measuring stations, is presented in figure 1b.

The ventilating air shroud, which is part of the unit, conveys the ventilating air diagonally across the exhaust gas side at the entrance and exit; but, for the purposes of this report, the flow characteristics will be considered as those for parallel flow of the fluids.

METHOD OF ANALYSIS

Heat Transfer

The evaluation of the thermal output of the heat exchanger from the experimental data was determined in the usual manner (see any previous heat exchanger report of this series).

The results and data are compiled in table I. The over-all thermal conductance UA is based upon the equation:

$$q_a = (UA)\ \Delta T_{mp} \tag{1}$$

where ΔT_{mp} is the mean temperature difference for parallel-flow of the fluids. It was determined from figure 29 of reference 1.

Prediction of the over-all thermal conductance of the heat exchanger was attempted. The heat flow was considered as occurring through the flat plates separating the fluid passages and through equivalent cylindrical surfaces at the edges of the ductlike passages over which one fluid flows diagonally with respect to the fluid flowing within the passage.

The equation used to determine the unit thermal conductances of the fluids within the passages of the heater (reference 2) is:

$$f_{cf} = 5.4 \times 10^{-4}\ T_{av}{}^{0.3}\ \frac{G^{0.8}}{D^{0.2}}\left(1 + 1.1\ \frac{D}{l}\right) \tag{2}$$

where

T_{av} average temperature of fluid

l mean length of passage (in the entrance length correction factor, see reference 3)

The equation used to determine the unit thermal conductance of the fluid flowing over the equivalent cylindrical edges (equation (29), reference 1) is:

$$f_{co} = 14.5 \times 10^{-4}\ F_a\ T_f{}^{0.43}\ \frac{G_o{}^{0.6}}{D_o{}^{0.4}} \tag{3}$$

where

G_o maximum weight rate of fluid per unit of area for flow past tube banks

D_c diameter of cylindrical edges

F_a tube arrangement modulus for in-line tube banks ($F_a = 1$ for 1 row of tubes)

The use of equation (3) means that the unit thermal conductance over the equivalent cylindrical leading and trailing edges of the flat plates has been postulated to be the same as that over the tubes of a single-row tube bank.

The unit thermal conductances obtained using equations (2) and (3) are employed for calculation of the thermal conductances f_cA over the equivalent cylindrical edges and also over the flat plate section of the heater. The equation used for calculating the over-all thermal conductance (see reference 1, equation (46)) was:

$$UA = \frac{1}{\left(\frac{1}{f_cA}\right)_a + \left(\frac{1}{f_cA}\right)_g} \qquad (4)$$

Equation (4) therefore was used to obtain the over-all thermal conductances over the equivalent cylindrical leading and trailing surfaces both on the air side, where the ventilating air flows over them, and also on the gas side, where the exhaust gas flows over similar surfaces. The predicted total over-all thermal conductance was computed as being the sum of the over-all thermal conductances over the cylindrical leading and trailing edges and over the flat plate section of the heater.

$$(UA)_{total} = (UA)_{plates} + (UA)_{ends}$$

The predicted and measured values of the total over-all thermal conductance are presented in figure 6 as a function of the ventilating air rate.

Sample Calculation

Predict the over-all thermal conductance UA of the flat plate heat exchanger at an exhaust gas weight rate of 2960 lb/hr and a ventilating air weight rate of 4000 lb/hr. The inlet temperatures of the exhaust gas and ventilating air are 1600° and 100° F, respectively.

1. <u>Thermal conductance on the ventilating air side</u>:

(a) Flat plate section

$$f_{cf} = 5.4 \times 10^{-4}\ T_{av}^{0.3}\ \frac{G^{0.8}}{D^{0.2}}\left(1 + 1.1\ \frac{D}{l}\right) \qquad (2)$$

Estimate the average temperature of the air side fluid to be about 250° F (710° R).

$T_{av} = 710°\ R$ $\qquad\qquad T_{av}^{0.3} = 7.17$

$G = \dfrac{W}{A} = \dfrac{4000}{0.222} = 18{,}000\ lb/hr\ ft^2$ $\qquad G^{0.8} = 2520$

$D = \dfrac{4 \times A}{P_w} = \dfrac{4 \times 0.222}{17.0} = 0.0522$ $\qquad D^{0.2} = 0.553$

$l = 0.834\ ft$

$$\left(1 + 1.1\ \frac{D}{l}\right) = \left(1 + 1.1\ \frac{0.0522}{0.834}\right) = 1.06$$

$$f_{cf} = 5.4 \times 10^{-4} \times 7.17 \times \frac{2520}{0.553} \times 1.06$$

$$= 18.7\ Btu/hr\ ft^2\ {}^{\circ}F$$

Flat plate heat transfer area, $A_f = 19.9$

$(f_{cf}A_f)_a = 372\ Btu/hr\ {}^{\circ}F$

then

$$\left(\frac{1}{f_{cf}A_f}\right)_a = 0.00269$$

(b) Cylindrical leading and trailing edges

$$f_{ce} = 14.5 \times 10^{-4} F_a T_f^{0.43} \frac{G_o^{0.6}}{D_o^{0.4}} \qquad (3)$$

F_a = tube arrangement modulus = 1 (single row of tubes)

Estimate surface temperature of the cylindrical edges of the heater plates to be about 660° F (for the cylindrical edges with ventilating air flowing over them).

$$T_f = \frac{(660 + 250)}{2} + 460 = 915^\circ \text{ R} \qquad T_f^{0.43} = 18.8$$

$$G_o = G = 18{,}000 \text{ lb/hr} \qquad G^{0.6} = 357$$

$$D_o = 0.0356 \text{ ft} \qquad D_o^{0.4} = 0.266$$

$$f_{ce} = 14.5 \times 10^{-4} \times 18.8 \times \frac{357}{0.266} = 36.6 \text{ Btu/hr ft}^2 \text{ °F}$$

Heat transfer area of cylindrical edges, $A_e = 0.48 \text{ ft}^2$

$$(f_{ce}A_e)_a = 17.6 \text{ Btu/hr °F}$$

then

$$\left(\frac{1}{f_{ce}A_e}\right)_a = 0.0569$$

2. <u>Thermal conductance on the exhaust gas side:</u>

(a) Flat plate section

$$f_{cf} = 5.4 \times 10^{-4} T_{av}^{0.3} \frac{G^{0.8}}{D^{0.2}} \left(1 + 1.1 \frac{D}{l}\right) \qquad (2)$$

Estimate the average temperature of the gas side fluid to be about 1400° F (1860° R)

$T_{av} = 1860° \text{ R}$ $\qquad T_{av}^{0.3} = 9.57$

$G = \frac{W}{A} = \frac{2960}{0.199} = 14,900 \text{ lb/hr ft}^2 \qquad G^{0.8} = 2200$

$D = \frac{4A}{P_w} = \frac{4 \times 0.199}{14.2} = 0.0560 \text{ ft} \qquad D^{0.2} = 0.561$

$l = 1.30 \text{ ft}$

$\left(1 + 1.1 \frac{D}{l}\right) = \left(1 + 1.1 \frac{0.056}{1.30}\right) = 1.05$

$f_{cf} = 5.4 \times 10^{-4} \ (9.57) \frac{(2200)}{(0.561)} \ (1.05)$

$= 21.0 \text{ Btu/hr ft}^2 \ °\text{F}$

Flat plate heat transfer area, $A_f = 19.9$

$(f_{cf}A_f)_g = 418 \text{ Btu/hr } °\text{F}$

$\left(\frac{1}{f_{cf}A_f}\right)_g = 0.00239$

(b) Cylindrical leading and trailing edges

$$f_{ce} = 14.5 \times 10^{-4} \ F_a \ T_f^{0.43} \ \frac{G_o^{0.6}}{D_o^{0.4}} \qquad (3)$$

F_a = tube arrangement modulus = 1 (single row of tubes)

Estimate surface temperature of the cylindrical edges of the heater plates to be about 1000° F (for the cylindrical edges with exhaust gas flowing over them).

$$T_f = \frac{(1000 + 1400)}{2} + 460 = 1660^\circ \text{ R} \qquad T_f^{0.43} = 24.2$$

$$G_o = G = 14{,}900 \text{ lb/hr ft}^2 \qquad G_o^{0.8} = 320$$

$$D_o = 0.0560 \text{ ft} \qquad D_o^{0.4} = 0.315$$

$$f_{ce} = 14.5 \times 10^{-4} \; (24.2) \; \frac{(320)}{(0.315)} = 35.7 \text{ Btu/hr ft}^2 \; ^\circ\text{F}$$

Heat transfer area of cylindrical edges, $A_e = 0.48$

$$(f_{ce}A_e)_g = 17.1 \text{ Btu/hr } ^\circ\text{F}$$

$$\left(\frac{1}{f_{ce}A_e}\right)_g = 0.0585$$

3. Over-all thermal conductances:

(a) Flate plate section

$$UA = \frac{1}{\left(\frac{1}{f_{cf}A_f}\right)_a + \left(\frac{1}{f_{cf}A_f}\right)_g} = \frac{1}{0.00269 + 0.00239} = 197 \; \frac{\text{Btu}}{\text{hr } ^\circ\text{F}}$$

(b) Cylindrical ends (ventilating air flowing over the ends)

$$UA = \frac{1}{\left(\frac{1}{f_{ce}A_e}\right)_a + \left(\frac{1}{f_{cf}A_e}\right)_g} = \frac{1}{0.0569 + 0.0992} = 6.4 \; \frac{\text{Btu}}{\text{hr } ^\circ\text{F}}$$

(c) Cylindrical ends (exhaust gas flowing over the ends)

$$UA = \frac{1}{\left(\frac{1}{f_{cf}A_e}\right)_a + \left(\frac{1}{f_{ce}A_g}\right)_g} = \frac{1}{0.111 + 0.0585} = 5.9 \; \frac{\text{Btu}}{\text{hr } ^\circ\text{F}}$$

$$(UA)_{total} = (UA)_{plates} + (UA)_{ends}$$

$$= 197 + 6.4 + 5.8 = 209 \text{ Btu/hr } {}^{\circ}F$$

UA (calculated from measured q_a) = 269 Btu/hr °F

Percentage deviation = 22 percent

Pressure Drop

Isothermal pressure drop.- The isothermal static pressure drop on the ventilating air side of the heater was predicted, using the following ideal system, which is approximated by the actual system:

(a) Sudden contraction at the entrance to the heater section

$$\frac{\Delta P_c}{\gamma} = K_c \frac{u_m^2}{2g} \quad (K_c = 0.22) \tag{5}$$

where u_m is the mean velocity of flow after contraction has occurred and K_c is a head loss coefficient for sudden contraction obtained from reference 4.

(b) Bend loss within the heater (near the air inlet) as the ventilating air turns through approximately 80°

$$\frac{\Delta P_b}{\gamma} = K_b \frac{u_m^2}{2g} \quad (K_b = 0.51) \tag{6}$$

(See figs. 5 and 6 of reference 4.)

(c) Frictional pressure drop within the heater passages, given by the equation

$$\frac{\Delta P_{fric}}{\gamma} = \zeta \frac{l}{D} \frac{u_m^2}{2g} \tag{7}$$

where ζ is the friction factor for commercial pipe.

(d) Bend loss within the heater (near the air outlet) as the ventilating air again turns through approximately 80°

$$\frac{\Delta P_b}{\gamma} = K_b \frac{u_m^2}{2g} \quad (K_b = 0.60) \tag{6}$$

(See figs. 5 and 6 of reference 4.)

(e) Sudden expansion as the ventilating air leaves the heater section and enters the exit duct.

$$\frac{\Delta P_{exp}}{\gamma} = K_{exp} \frac{u_m^2}{2g} \tag{8}$$

where K_{exp} is a head loss coefficient for sudden expansion of the fluid passage. It is determined from the equation $K_{exp} = (1 - m)^2$.

The summation of the foregoing pressure drops was then the predicted magnitude of the ventilating air side pressure drop. The deviation of the predicted values from the measured values was about 15 percent. The predicted and measured pressure drops are plotted in figure 7 as a function of the ventilating air weight rate.

The isothermal static pressure drop on the exhaust gas side of the heat exchanger was predicted on the basis of a similar ideal system. This system consisted of (1) sudden contraction, (2) sudden expansion, (3) sudden contraction, (4) friction losses as the exhaust gas flowed through the heater passage, (5) sudden expansion as the gas left the heater passages, (6) sudden contraction, and (7) sudden expansion. The basic equations applied to the calculation of the air side static pressure drop (equations (5), (7), and (8)) were again used together with appropriate values of the respective head loss coefficients.

The summation of the preceding pressure drop terms was considered as the predicted value of the isothermal static pressure drop across the exhaust gas side. The deviation of the predicted values from the measured static pressure drops was about 40 percent. A plot of both the measured pressure drops and also of the predicted values as a function of the exhaust gas weight rate is presented in figure 8. The data for this plot are found in table II.

Over-all isothermal head loss coefficients based on the isothermal static pressure drops across either side of the exchanger were also determined, using the equation:

$$\frac{\Delta P}{\gamma} = K \frac{u_m^2}{2g} \tag{9}$$

<u>Non-isothermal pressure drop</u>.- Prediction of the non-isothermal static pressure drop across either side of the heat exchanger was attempted, using the isothermal static pressure drop measurements according to the equation (see reference 1, equation (54)):

$$\Delta P_{non-iso} = \Delta P_{iso} \left(\frac{T_{av}}{T_{iso}}\right)^{1.13}$$

$$+ \left(\frac{W}{3600}\right)^2 \frac{1}{2g\gamma_1 A_h^2} \left[\left(\frac{A_h^2}{A_1^2} + 1\right)\left(\frac{T_2}{T_1} - 1\right)\right] \tag{10}$$

where

ΔP_{iso} isothermal static pressure drop across heat exchanger at temperature T_{iso}

T_{av} arithmetic average of T_1 and T_2, absolute mixed-mean temperatures of the fluid at inlet and outlet ends of heat exchanger, respectively

γ_1 weight density of fluid evaluated at temperature T_1

A_1 cross-sectional area of flow at pressure-measuring station (equal areas upstream and downstream)

The predictions were within 15 percent of the measured values on the ventilating air side and within 20 percent of the measured values on the exhaust gas side. Both measured and predicted values of the non-isothermal static pressure drop on either side of the heat exchanger are plotted in figures 7 and 8 as functions of the weight rate of fluid on that side.

DISCUSSION

Heat Transfer

The deviation of the predicted values of the over-all thermal conductance from the experimentally determined values is on the average about 20 percent. Consideration of the possible sources for discrepancy indicates that the more obvious ones are (a) heat transfer by gaseous radiation to the heater surfaces (see reference 5), (b) heat transfer by radiation from the heater surfaces to the cooler walls of the shroud and then by convection from walls of the shroud to the ventilating air (a rough estimate of this type of heat transfer could account for approximately 5 percent of the discrepancy), (c) the indentations on the gas side which formed the guide vanes on the air side probably increased the unit thermal conductance on the exhaust gas side, and (d) the cross-flow characteristics of the shroud at the entrance and exit of the ventilating air were disregarded when the mean temperature difference for determination of the experimental over-all thermal conductance was chosen as that for parallel flow only.

Temperatures of the heater surfaces (see table I) were measured at several points on the ventilating air side, three at the inlet, and two at the outlet sections. Inspection of the recorded temperatures indicates that the highest temperatures usually were found in the first two of the three thermocouple stations, located at the entrance to the first two air passages. Of the two thermocouples located at the outlets of the ventilating air passages, the one located in the passage near the center gave higher temperatures than those given by the thermocouple located at the end of the second passage down from the top. The highest temperature recorded by any thermocouple was about 1390° F at the entrance and about 1125° F at the outlet of the ventilating air for an exhaust gas inlet temperature of 1600° F.

Pressure Drop

The accurate prediction of the isothermal static pressure drop across either side of this heat exchanger is difficult because of the complexity of the passages (bends, diagonal expansions in the gas side, diagonal flow of the fluids over the leading and trailing edges of the flat plate passages).

Isothermal head loss coefficients (see equation 9) were computed for both sides of the heat exchanger. It was found that the values of K on the exhaust gas side were about 50 percent greater than those on the ventilating air side, which were about 2.5. Comparison of the values of K obtained with previous heat exchangers shows that K for the exhaust gas side is usually much lower than that for the ventilating air side. (See references 6 to 11.)

CONCLUSIONS

The thermal and fluid-dynamic performance of a flat-plate type exhaust gas and air heat exchanger are reported. Methods of prediction of the heat transfer and pressure drop characteristics are presented. The agreement with experimental values is usually within 25 percent.

University of California,
Berkeley, Calif., July 1944.

REFERENCES

1. Boelter, L. M. K., Martinelli, R. C., Romie, F. E., and Morrin, E. H.: An Investigation of Aircraft Heaters: XVIII - A Design Manual for Exhaust Gas and Air Heat Exchangers. NACA ARR No. 5A06, 1945.

2. Martinelli, R. C., Weinberg, E. B., Morrin, E. H., and Boelter, L. M. K.: An Investigation of Aircraft Heaters. III - Measured and Predicted Performance of Double Tube Heat Exchangers. NACA ARR, Oct. 1942.

3. Boelter, L. M. K., Dennison, H. G., Guibert, A. G., and Morrin, E. H.: An Investigation of Aircraft Heaters. X - Measured and Predicted Performance of a Fluted-Type Exhaust Gas and Air Heat Exchanger. NACA ARR, March 1943.

4. SAE Committee A-9 Aircraft Air Conditioning Equipment: Airplane Heating and Ventilating Equipment. Engineering Data. Aero. Inf. Rep. 2, Jan. 1, 1943.

5. Martinelli, R. C., Morrin, E. H., and Boelter, L. M. K.: An Investigation of Aircraft Heaters. VII - Thermal Radiation from Athermanous Exhaust Gases. NACA ARR, Dec. 1942.

6. Boelter, L. M. K., Miller, M. A., Sharp, W. H., and Morrin, E. H.: An Investigation of Aircraft Heaters. XI - Measured and Predicted Performance of a Slotted-Fin Exhaust Gas and Air Heat Exchanger. NACA ARR No. 3D16, 1943.

7. Boelter, L. M. K., Dennison, H. G., Guibert, A. G., and Morrin, E. H.: An Investigation of Aircraft Heaters. XII - Performance of a Formed-Plate Crossflow Exhaust Gas and Air Heat Exchanger. NACA ARR No. 3E10, 1943.

8. Boelter, L. M. K., Guibert, A. G., Miller, M. A., and Morrin, E. H.: An Investigation of Aircraft Heaters. XIII - Performance of Corrugated and Noncorrugated Fluted Type Exhaust Gas-Air Heat Exchangers. NACA ARR No. 3H26, 1943.

9. Boelter, L. M. K., Guibert, A. G., Rademacher, J. M., and Sloggy, L. J. B.: An Investigation of Aircraft Heaters. XIX - Performance of Two Finned-Type Crossflow Exhaust Gas and Air Heat Exchangers. NACA ARR No. 4H21, 1944.

10. Boelter, L. M. K., Guibert, A. G., Rademacher, J. M., Romie, F. E., and Sanders, V.D.: An Investigation of Aircraft Heaters. XX - Measured and Predicted Performance of a Finned-Type Cast-Aluminum Crossflow Exhaust Gas and Air Heat Exchanger. NACA ARR No. 5A08, 1945.

11. Boelter, L. M. K., Guibert, A. G., Rademacher, J. M., Romie, F. E., Sanders, V. D., and Sloggy, L. J. B.: An Investigation of Aircraft Heaters. XXI - Measured and Predicted Performance of a Flattened-Tube Type Crossflow Exhaust Gas and Air Heat Exchanger. NACA ARR No. 5A10, 1945.

TABLE I - EXPERIMENTAL RESULTS ON A FLAT PLATE PARALLEL FLOW TYPE HEAT EXCHANGER

AIR SIDE						EXHAUST GAS SIDE						HEATER TEMPS.					OVERALL PERFORMANCE	
Run No.	T_{a_1} °F	T_{a_2} °F	ΔT_a °F	W_a lb/hr	q_a K Btu/hr	T_{g_1} °F	T_{g_2} °F	ΔT_g °F	W_g lb/hr	q_g K Btu/hr	q_g/q_a	t_1 °F	t_2 °F	t_3 °F	t_4 °F	t_5 °F	ΔT_{m_p} °F	UA Btu/hr °F
23	102	610	508	1280	158	1608	1330	278	1740	132	0.84	1315	1310	1310	880	940	1070	147
24	104	482	378	2250	206	1608	1265	343	1770	165	0.80	1255	1225	1260	720	780	1100	187
17	97	363	266	4000	258	1600	1158	442	1760	212	0.82	1140	1175	1125	575	575	1130	228
18	96	325	229	5200	288	1600	1121	479	1750	228	0.79	1085	1145	1075	510	485	1110	260
25	93	694	601	1000	146	1604	1428	176	2950	143	0.98	1386	1377	1360	1032	1095	1070	136
26	93	541	448	2270	247	1595	1347	248	2960	182	0.74	1278	1278	1269	795	862	1130	218
19	93	405	312	4050	306	1591	1256	335	2910	266	0.87	1195	1200	1150	645	665	1140	269
20	95	366	271	5060	332	1586	1227	359	2840	277	0.83	1160	1155	1110	575	600	1150	289
27	96	698	602	1000	146	1604	1464	140	3510	136	0.92	1394	1386	1381	998	1125	1100	133
28	95	564	469	2280	259	1600	1386	214	3590	212	0.82	1269	1278	1282	799	897	1130	229
21	97	439	342	4000	331	1610	1312	298	3520	286	0.86	1225	1245	1175	720	710	1170	283
22	97	389	292	5060	358	—	—	—	3470	—	—	—	—	—	—	—	—	—

TABLE II

ISOTHERMAL STATIC PRESSURE DROP

Run	W (lb/hr)	$\Delta P''$ (pred.) (in. H_2O)	$\Delta P''$ (meas.) (in. H_2O)	K[a]
Ventilating Air Side				
57	1020	[b]0.18	0.24	3.8
58	1390	.32	.42	3.5
59	1970	.62	.75	3.1
51	2130	.72	.86	3.1
52	4100	2.51	2.64	2.5
53	5850	4.95	4.52	2.1
Exhaust Gas Side				
55	2380	0.97	1.83	4.2
60	3490	1.97	3.67	3.9
48	3510	2.02	3.94	4.2
56	4100	2.70	4.82	3.7
49	4920	3.90	7.37	4.0
61	4950	3.92	6.96	3.7
62	6450	6.55	11.7	3.7
50	6460	6.55	12.4	3.5
63	8120	10.2	18.3	3.6
51	8340	10.7	20.5	3.8

[a]The values of K are based on equation

$$\frac{\Delta P}{\gamma} = K \frac{{u_m}^2}{2g}$$

where $\Delta P = 5.19\ \Delta P''_{meas.}$, lb/ft^2

[b]The values of $\Delta P''_{pred.}$ are interpolated from the curves of figs. 7 and 8.

TABLE III

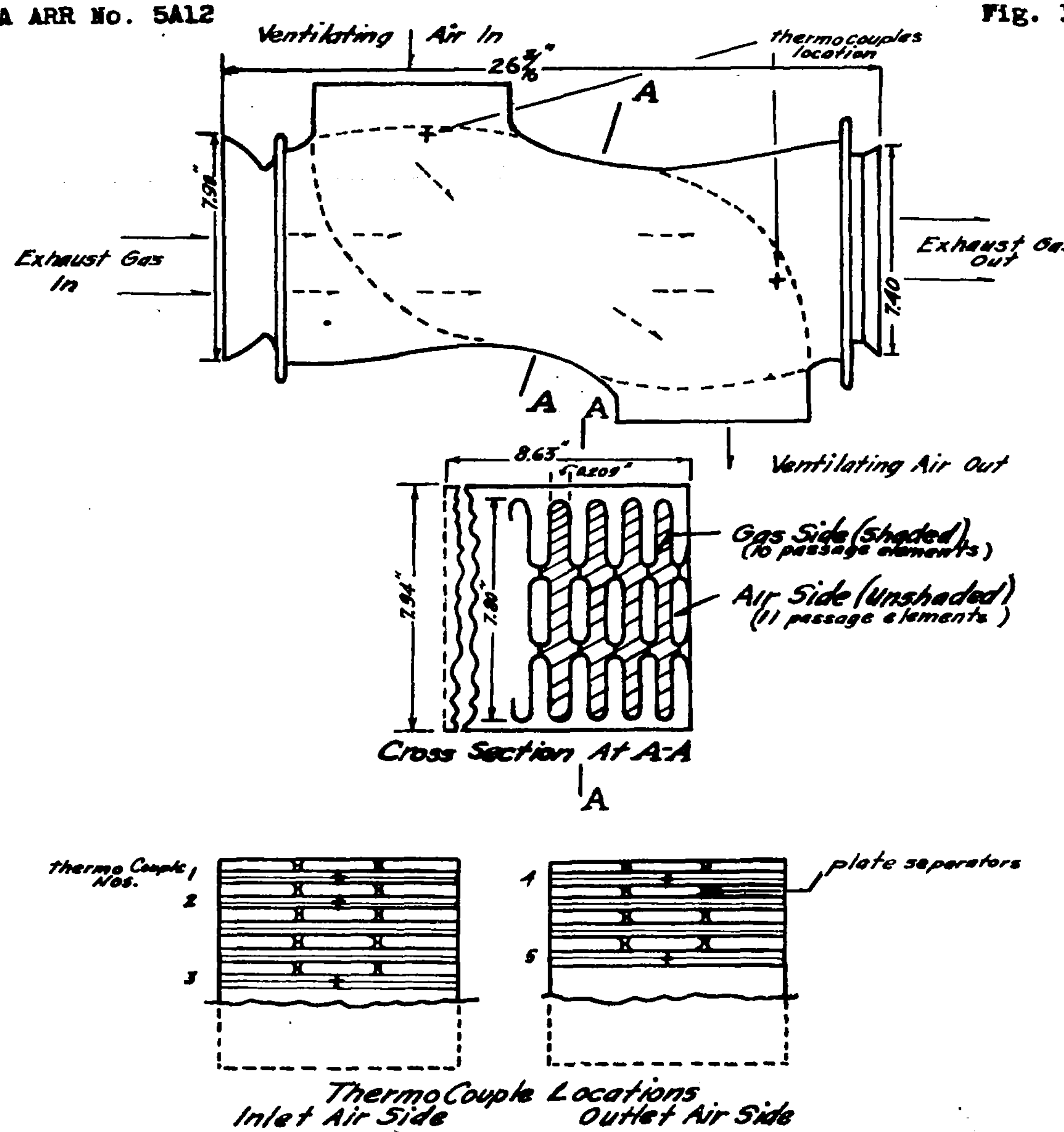
NON-ISOTHERMAL STATIC PRESSURE DROP

Run	W (lb/hr)	T_1 (°R)	T_2 (°R)	T_{av} (°R)	$\Delta P''_{iso}$ (in. H_2O)	$\Delta P''_{non-iso}$ Measured (in. H_2O)	Predicted[a] (in. H_2O)
Ventilating Air Side							
65	1050	561	1153	857	0.25	0.66	0.49
66	1530	564	1083	824	.48	1.16	.89
67	2660	566	975	770	1.24	2.56	2.12
68	3900	561	915	738	2.35	4.25	3.89
Exhaust Gas Side							
24	1770	2065	1725	1895	1.02	5.82	3.80
26	2960	2055	1807	1931	2.72	13.7	10.8
27	3490	2060	1840	1950	3.72	17.3	14.5

[a]Predicted values are based on equation (10)

$$\Delta P_{non-iso} = \Delta P_{iso} \left(\frac{T_{av}}{T_{iso}}\right)^{1.13}$$

$$+ \left(\frac{W}{3600}\right)^2 \frac{1}{2g\gamma_1 A_h^2} \left[\left(\frac{A_h^2}{A_1^2} + 1\right)\left(\frac{T_2}{T_1} - 1\right)\right] \qquad (10)$$



	AIR SIDE	GAS SIDE
Cross Sectional Area Ft^2	0.222	0.199
Wetted Permeter Ft.	17.0	14.2
Hydraulic Diameter Ft.	0.0521	0.0561
Heat Transfer Area Ft^2	20.4	20.4
Flat Plate Portion	19.9	19.9
Cylindrical Ends (see Fig. 1a)	0.48	0.48

Fig. 1.- Schematic Diagram of a Flat Plate Parallel Flow-Type Heat Exchanger.

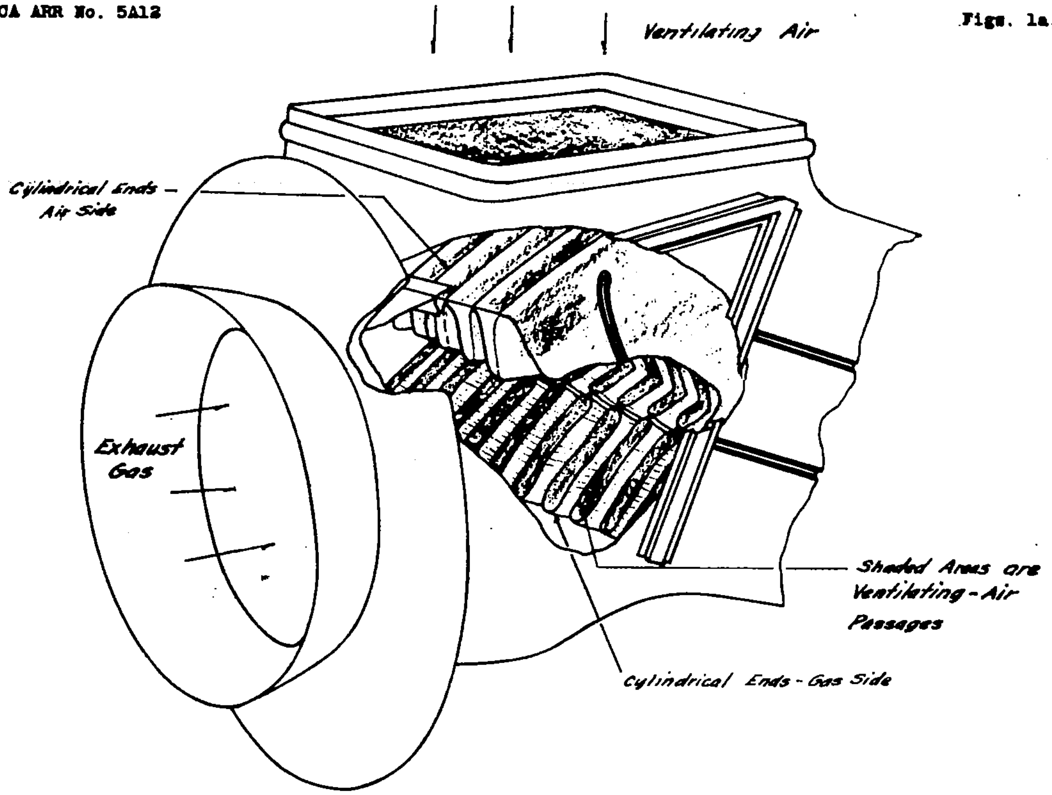


Fig. 1a – Cut-away View of The Flat Plate Heat Exchanger, Showing The Exhaust-Gas and Ventilating-Air Passages.

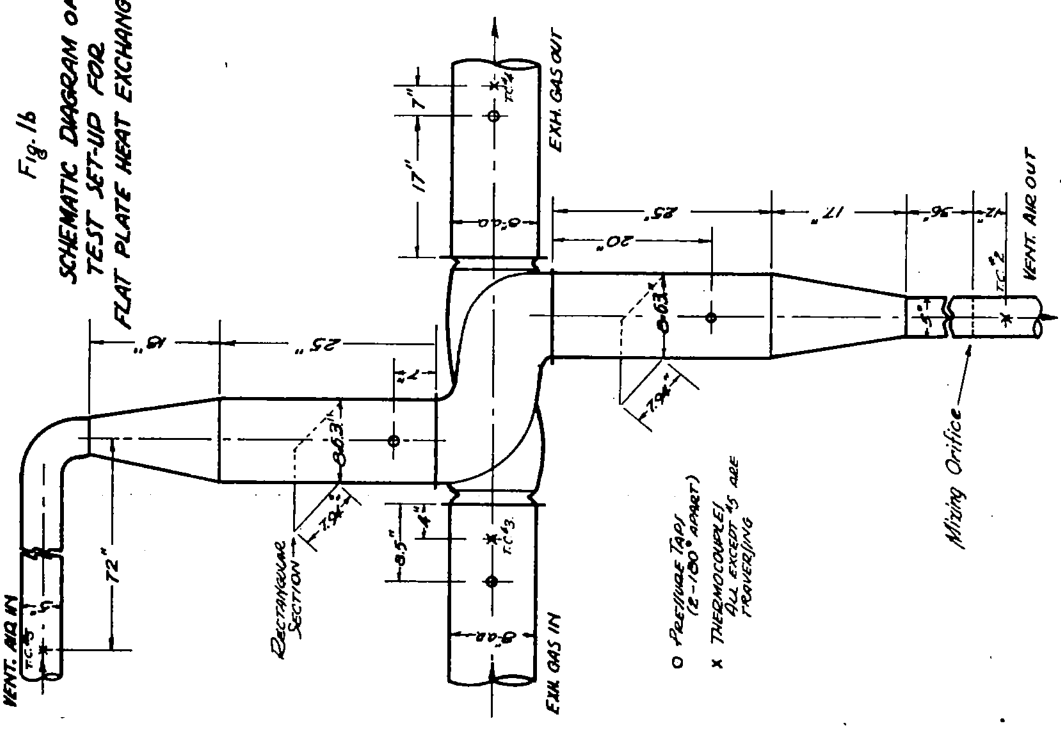


Fig. 1b
SCHEMATIC DIAGRAM OF TEST SET-UP FOR FLAT PLATE HEAT EXCHANGER

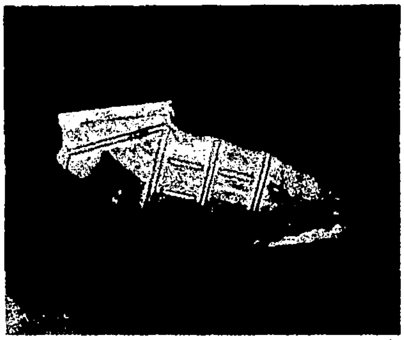


Figure 2.- Photograph of flat plate heat exchanger with parallel-flow shroud.

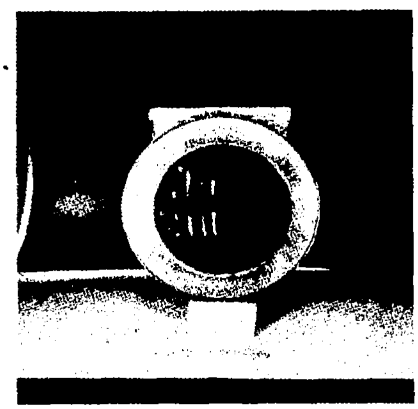


Figure 3.- Photograph of the inlet end of the exhaust-gas passage of the flat plate heat exchanger.

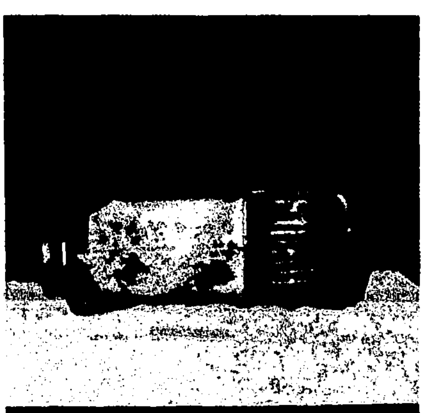


Figure 4.- Photograph of the outlet end of ventilating-air passage of the flat plate heat exchanger.

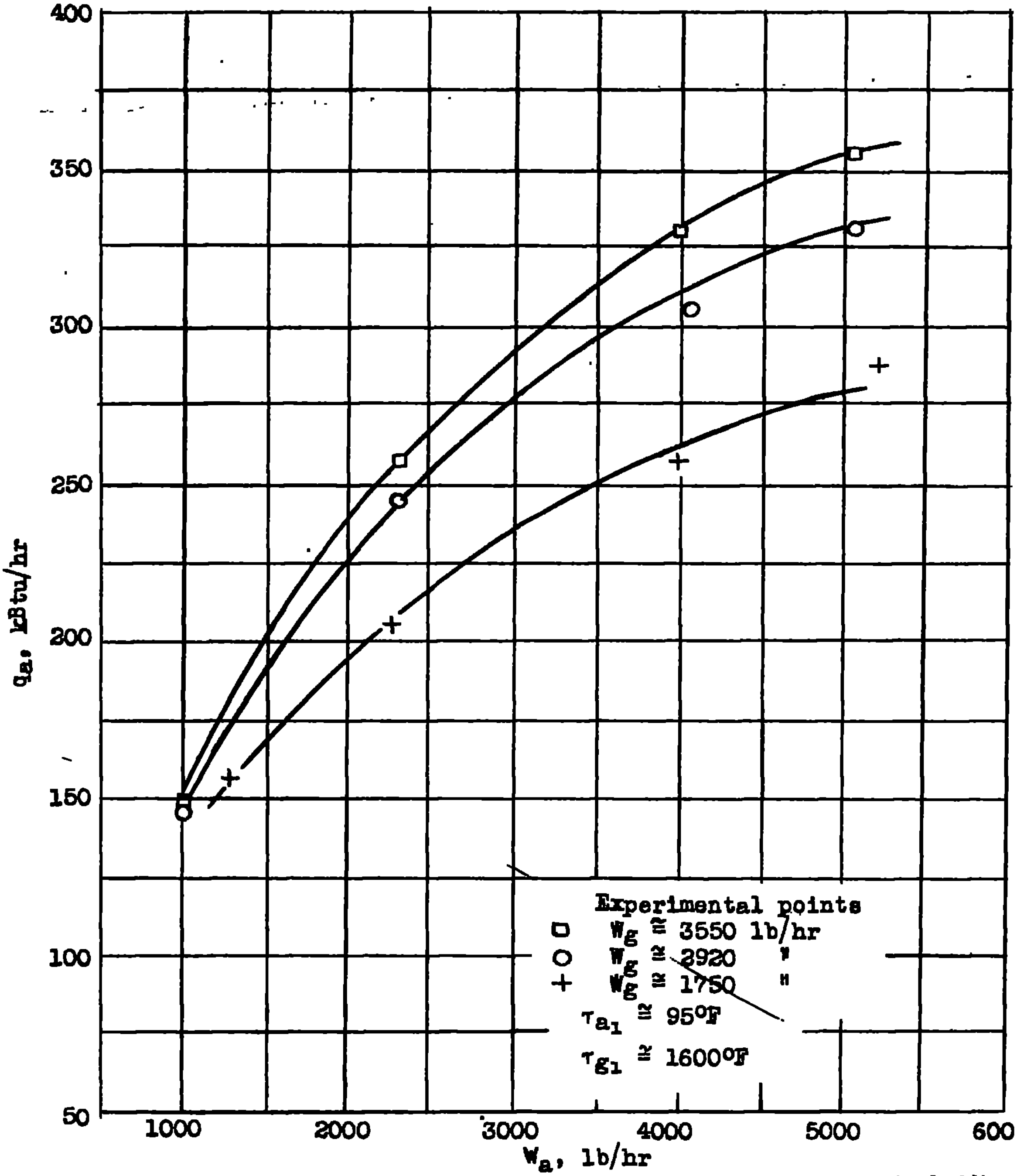


Figure 5.- Thermal output of a flat plate parallel flow type heat exchanger, as a function of the ventilating air rate.

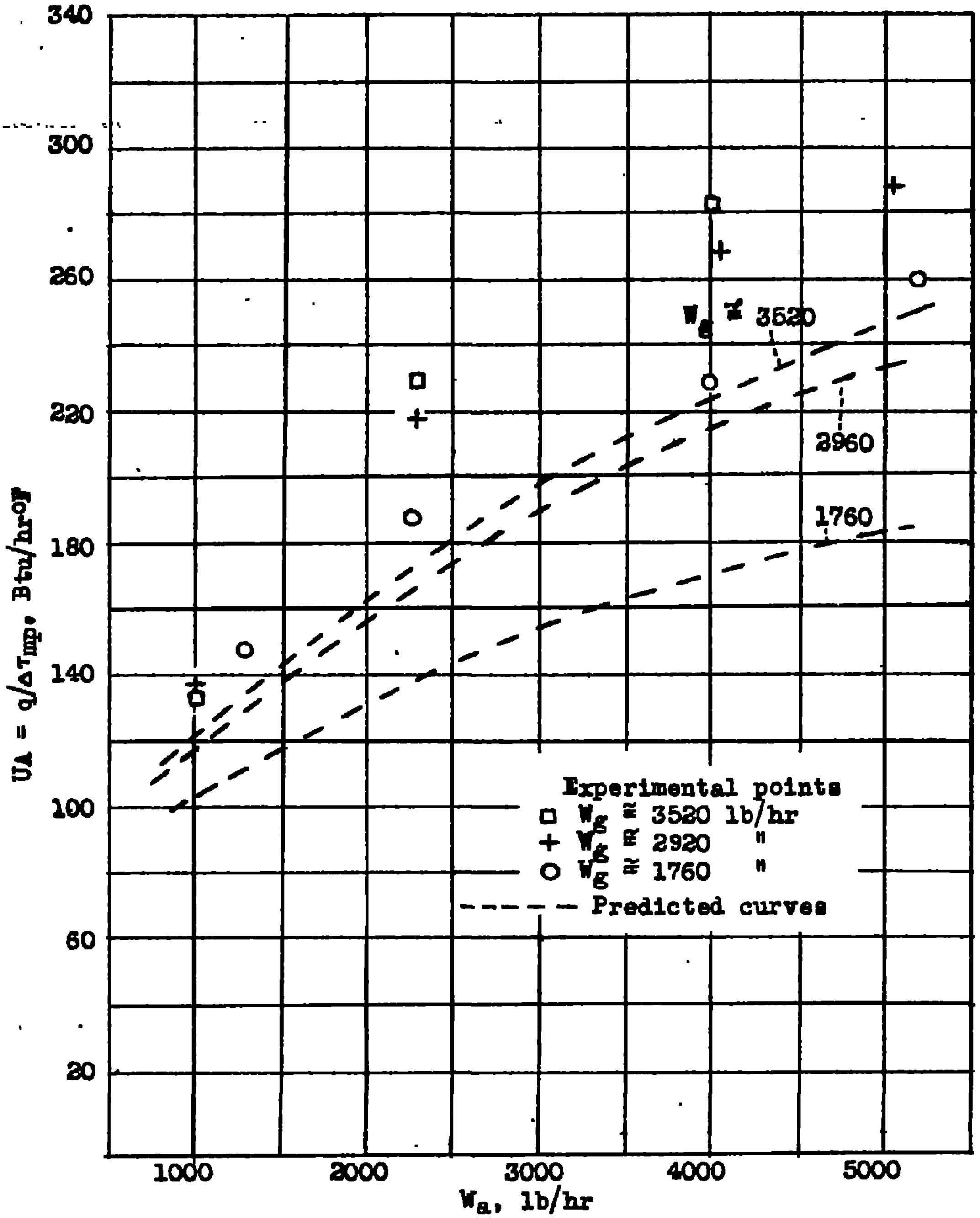


Figure 6.- Overall thermal conductance of a flat plate parallel flow type heat exchanger, as a function of ventilating air rate.

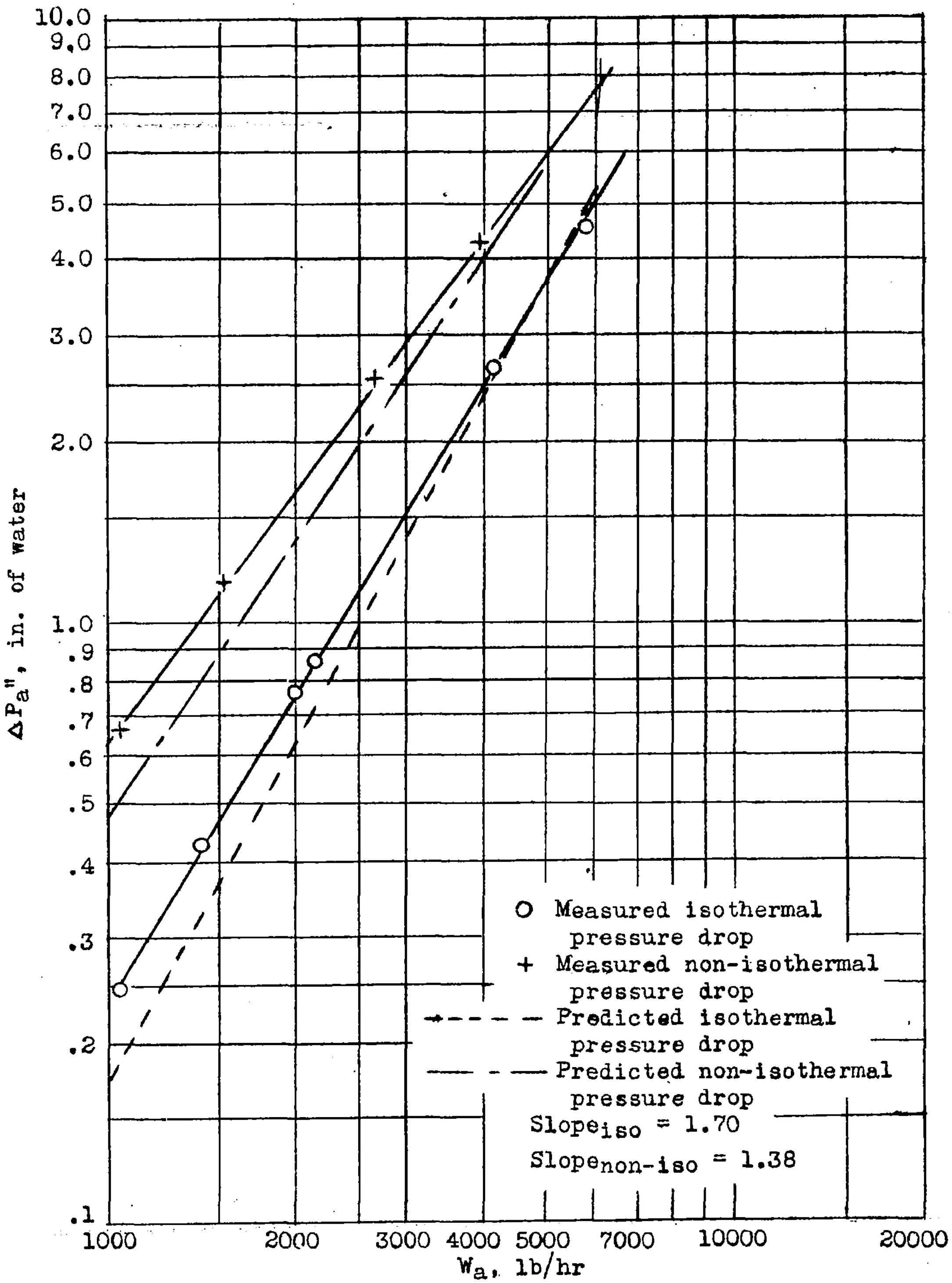


Figure 7.- Static pressure drop across the air side of a flat plate parallel flow type heat exchanger, as a function of ventilating air rate.

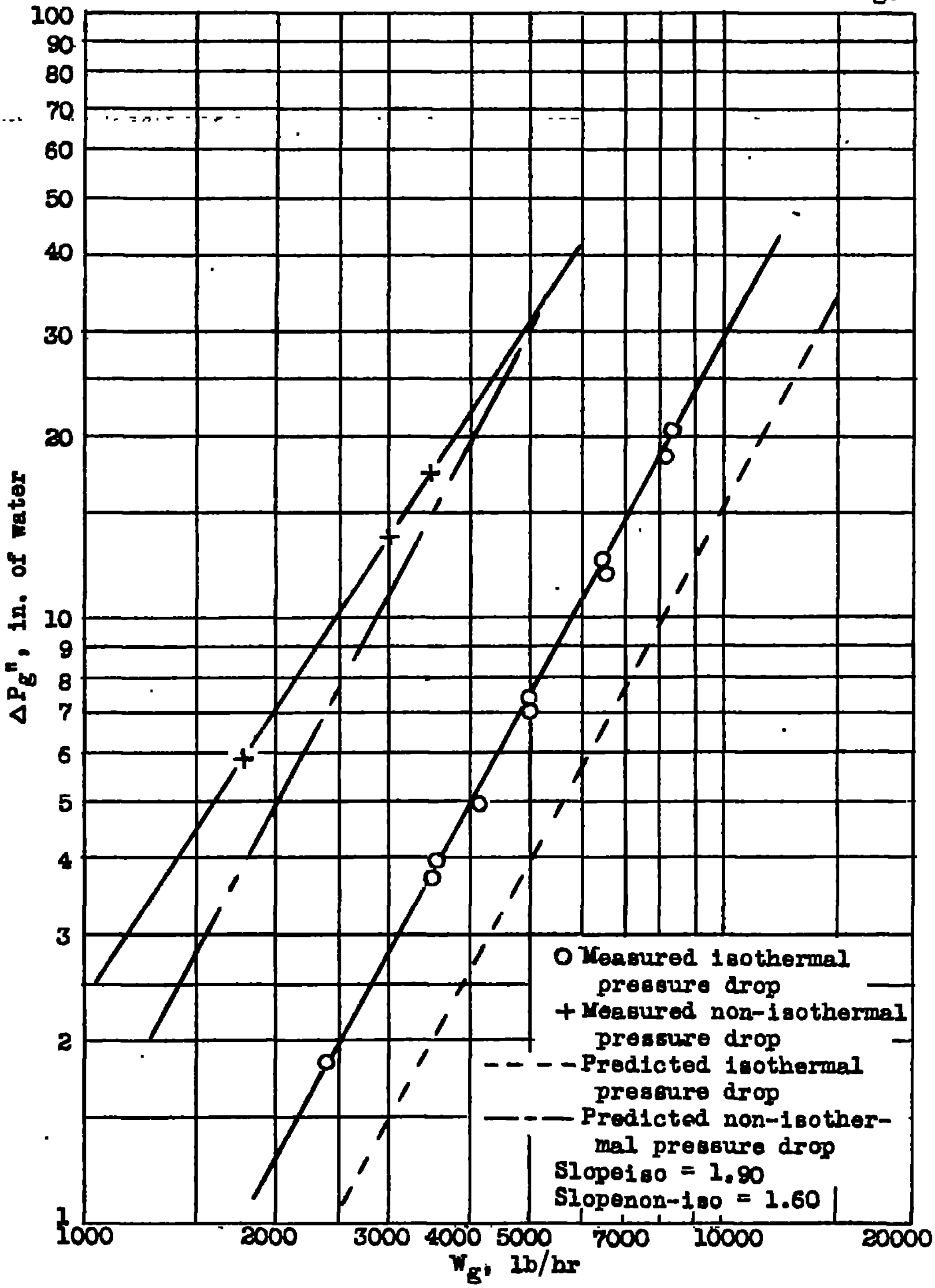


Figure 8.- Static pressure drop across a flat plate parallel flow type heat exchanger, as a function of exhaust gas rate.

ATSC FORM 69 (13 MAR 47)

AUTHOR(S)		ATI- 7947
Boelter, L. M. Guibert, A. G. and others	DIVISION: Power Plants, Reciprocating (6) SECTION: Cooling (1) CROSS REFERENCES: Heat exchangers (48400); Heat transfer (48800); Heat exchangers - Pressure drop (48520)	ORIG. AGENCY NUMBER ARR-5A12 REVISION

AMER. TITLE: An investigation of aircraft heaters - XXIII - Measured and predicted performance of a flat-plate type exhaust gas and air heat exchanger

FORG'N. TITLE:

ORIGINATING AGENCY: National Advisory Committee for Aeronautics, Washington, D. C.

TRANSLATION:

COUNTRY	LANGUAGE	FORG'N.CLASS	U. S.CLASS.	DATE	PAGES	ILLUS.	FEATURES
U.S.	Eng.		Unclass.	Apr'45	27	13	photos, tables, diagrs, graphs

ABSTRACT

Tests of thermal performance of static pressure-drop characteristics of flat-plate exhaust gas and air-heat exchanger are presented. Range of exhaust gas weight rates used for determination of heat transfer varied from about 1700 lb/hr to 5200 lb/hr. Temperature was maintained at 1600°F. Static pressure drop measured across both sides of heat exchanger under isothermal and nonisothermal conditions and surface temperatures were recorded. Predictions and experimental values of thermal output and pressure drops are compared.

NOTE: Requests for copies of this report must be addressed to: N.A.C.A., Washington, D. C.