# NACA

# RESEARCH MEMORANDUM

for the

Air Materiel Command, Army Air Forces

COOLING CHARACTERISTICS OF THE V-1650-7 ENGINE

I - COOLANT-FLOW DISTRIBUTION, CYLINDER TEMPERATURES, AND

HEAT REJECTIONS AT TYPICAL OPERATING CONDITIONS

By John H. Povolny and Louis J. Bogdan

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# NATIONAL ADVISORY COMMITTEE
# FOR AERONAUTICS

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SUMMARY

An investigation was conducted to determine the coolant-flow distribution, the cylinder temperatures, and the heat rejections of the V-1650-7 engine. The tests were run at several power levels varying from minimum fuel consumption to war emergency power and at each power level the coolant flows corresponded to the extremes of those likely to be encountered in typical airplane installations. A mixture of 30-percent ethylene glycol and 70-percent water was used as the coolant. The temperature of each cylinder was measured between the exhaust valves, between the intake valves, in the center of the head, on the exhaust-valve guide, at the top of the barrel on the exhaust side, and on each exhaust spark-plug gasket.

For an increase in engine power from 628 to approximately 1700 brake horsepower the average temperature for the cylinder heads between the exhaust valves increased from 437° to 517° F, the engine-coolant heat rejection increased from 12,600 to 22,700 Btu per minute, the oil heat rejection increased from 1030 to 4600 Btu per minute, and the aftercooler-coolant heat rejection increased from 450 to 3500 Btu per minute.

INTRODUCTION

At the request of the Air Materiel Command, Army Air Forces, an investigation has been conducted at the NACA Cleveland laboratory on a dynamometer test stand to determine the cooling characteristics of

the V-1650-7 engine. The results of the first phase of this investigation, which consisted of a determination of the coolant-flow distribution, the cylinder temperatures, and the heat rejection to the engine coolant, to the oil, and to the aftercooler coolant at typical operating conditions, are reported. A range of power outputs from 630 to 1700 brake horsepower was covered with engine coolant flows at each power corresponding to the extremes of those likely to be encountered in typical airplane installations. Both aftercooler and engine coolants consisted of a mixture of 30-percent ethylene glycol and 70-percent water. In order to obtain a comprehensive survey, the temperatures of each cylinder were measured at six different locations.


## APPARATUS AND INSTRUMENTATION

### Engine

The V-1650-7 engine is a 12-cylinder, liquid-cooled engine with a bore of 5.4 inches and a stroke of 6.0 inches. It is fitted with a two-stage supercharger having impeller diameters of 12.0 and 10.1 inches. The two impellers are mounted on the same shaft and can be operated at a speed of either 5.802 or 7.349 times the engine speed. A liquid-type aftercooler is interposed between the supercharger outlet and the intake manifold. The spark timing is controlled by the throttle position and varies from $29^{\circ}$ B.T.C. for full-closed throttle to approximately $45^{\circ}$ B.T.C. for half-open to full-open throttle. Both the intake and exhaust spark plugs are timed to fire simultaneously.

The coolant flow path through the cylinder bank is schematically shown in figures 1 and 2. The coolant is distributed to the six barrels of each cylinder bank by an external coolant branch tube. This coolant branch tube, which is connected to the discharge of the engine coolant pump, has three outlets, each of which supplies coolant to two adjacent cylinder barrels. After entering the barrels, the coolant flows around each cylinder barrel and up into the cylinder head through 14 connector tubes. The coolant then passes over the cylinder heads and is discharged at a single outlet at the forward end of the cylinder bank. Two pressure-equalizing holes (about 5/8-in. diameter) are provided at the bottom of each bank between the two pairs of adjacent cylinders that are not supplied by a common branch-tube outlet. It may be seen from figure 2 that the flow over each cylinder head is equal to the total of all the flows through the connector tubes upstream of the cylinder head in question.

For convenient identification, the manufacturer's designation of the banks and cylinders is used. Thus, when facing the rear of the engine, the right-hand bank is designated bank A and the left-hand bank is designated bank B. The cylinders of each bank are numbered from 1 to 6 starting at the front of the engine.

## Flow-Distribution Setup

A bench setup in which the connector tubes in bank B were calibrated with water at room temperature was used to determine the coolant-flow distribution through the engine. As shown in figures 3 and 4, pressure taps were installed both across the barrel inlets and across the connector tubes. The numbering system for identifying the connector tubes is shown in figure 4. Thin-plate orifices were installed in the connector tubes to give a measurable pressure drop. A thin-plate orifice was also installed in the water supply line to measure the total flow of water to the bank.

## Engine Setup

The test setup is shown in figure 5. The V-1650-7 engine on which the investigation was conducted was mounted on a dynamometer stand equipped with a 3000-horsepower eddy-current water-gap dynamometer.

The inlet-manifold pressure of the engine was measured at the priming-pipe filter-housing fitting and the inlet-manifold temperature was measured with a single, unshielded, iron-constantan thermocouple located approximately 10 inches downstream of the aftercooler outlet. The fuel used was AN-F-28, Amendment 2; the flow was measured with calibrated rotameters.

Combustion-air and exhaust systems. - Combustion air was supplied to the engine by the laboratory central system and was metered by means of an adjustable orifice installed in the supply duct. A filter was installed in the duct between the orifice and the carburetor. The temperature and the pressure at the carburetor inlet were maintained at the desired values by means of automatic equipment. Air temperatures at the orifice and at the carburetor were measured with thermocouple rakes.

The engine exhaust stacks, which were so constructed that the stack openings were equal in area to the engine exhaust-port openings, were partly water-jacketed and were connected to a header of

12-inch diameter with flexible hoses.  The headers were connected to
the laboratory central exhaust system by means of which the desired
exhaust pressures could be maintained.  The spark plugs, and the
thermocouple leads were shielded from the unjacketed portion of the
exhaust stacks.

Engine coolant system. - A schematic diagram of the engine cool-
ant system is shown in figure 6.  An auxiliary pump was placed in
series with the engine pump to permit the coolant flow to be varied
independently of the engine speed.  The coolant flow was measured with
a venturi.  A throttle valve installed downstream of the venturi was
used to increase the throat pressure enough to prevent cavitation.
The system pressure was regulated with a compressed-air-bleed combina-
tion on the coolant expansion tank.  A centrifugal-type vapor separa-
tor was installed in each of the block-outlet lines to remove air or
any vapor that may have formed during runs in which boiling occurred.
Vent lines were run from both the vapor separators and the block out-
lets to the expansion tank.  Sight glasses were installed in both sets
of vent lines to permit visual observation of the coolant.

The coolant-temperature control unit consisted of two aircraft-
type coolers connected in parallel and a three-way, air-operated,
thermostatically controlled mixing valve installed at the junction of
the main line from and the bypass line around the coolers.  Water was
used as the cooling medium in the coolers and was metered with cali-
brated rotameters.  The coolant used was a mixture of 30-percent
AN-E-2 ethylene glycol and 70-percent water with 0.2 percent by volume
of sodium mercaptobenzothiazole (NaMBT) added as a corrosion inhibitor.
Pressure taps and thermocouples were installed in the coolant and
cooling-water lines, as shown in figure 6.

Lubricating oil system. - A schematic diagram of the lubricating
oil system including thermocouple and pressure-tap locations is shown
in figure 7.  A flow-rate and oil-level indicator similar to the one
described in reference 1 was incorporated in the reservoir tank.  The
temperature-control unit was similar to that for the coolant system.
A filter unit was installed in the engine outlet line and a chip
detector was installed in the drain plug of the filter.  The lubri-
cating oil used was Navy 1120.

Aftercooler coolant system. - Figure 8 is a schematic diagram of
the aftercooler coolant system.  In addition to passing through the
aftercooler unit, the aftercooler coolant also passed through a cooling
jacket that surrounded the supercharger.  The aftercooler coolant flow
was regulated by means of a throttle valve and metered with a

calibrated venturi. The temperature-control unit consisted of water-cooled coolers and a three-way mixing valve that were similar to those in the engine coolant system. The expansion tank is incor-porated in the housing of the aftercooler unit and a relief valve set to relieve at a pressure of 20 pounds per square inch gage is mounted on the tank. The composition of the coolant used was the same as that used in the engine coolant system. Thermocouples and pressure taps were located in the aftercooler-coolant and cooling-water lines, as shown in figure 8.

## Temperature Measurements

Thermocouples for liquid-temperature measurement. - Two types of thermocouple, iron-constantan and copper-constantan, were installed in the engine-coolant, aftercooler-coolant, lubricating-oil, and cooling-water lines in the locations shown in figures 6 to 8. The iron-constantan thermocouples were connected to both a self-balancing recording potentiometer and to a self-balancing direct-reading poten-tiometer. The copper-constantan thermocouples were connected to a portable precision-type potentiometer and balance was indicated on a light-beam galvanometer to provide an accurate determination of the  temperature differences across the engine and the coolers.

Thermocouples for engine-temperature measurement. - The cylinder-thermocouple installation is shown in figures 9 and 10. Thermocouples were located in each cylinder between the exhaust valves, between the intake valves, in the center of the head, on the exhaust-valve guide, at the top of the cylinder barrel on the exhaust side, and on the exhaust-spark-plug gasket. The thermo-couple holes in the cylinder head were drilled with the aid of jigs to insure uniformity and accuracy of location. The cylinder-head thermocouples were silver-soldered into brass plugs and these plugs were peened into the bottom of the drilled holes. The cylinder-barrel thermocouples were spot-welded to the cylinder liner and the leads were brought through the cylinder coolant jacket by means of a pressure-tight fitting. The exhaust-spark-plug-gasket thermo-couples were silver-soldered into a hole drilled in the edge of the gasket. All of the thermocouple leads were brought to the outside of the engine through stainless-steel tubing. The leads outside the engine were insulated with a flexible glass sleeve and protected with flexible ignition-type shielding. The temperatures were read on the self-balancing direct-reading potentiometer and recorded on the self-balancing recording potentiometer.

## PROCEDURE

### Flow-Distribution Investigation

Inasmuch as both cylinder banks are similar in construction, the coolant-flow distribution was determined for only one bank. The pressure drops across both the connector tubes and the barrel inlets were measured for a range of total flows to the bank varying from 30 to 95 gallons per minute. From these data a plot of the pressure drop across each connector-tube orifice against the total flow to the bank was made (fig. 11). Using the value of pressure drop and the slope given by these curves and assuming the same flow coefficient for all the connector-tube orifices, the flow through each connector tube was calculated for the various total flows to the bank. The flow distribution to the various cylinders within the bank was then determined from the calculated flow through the connector tubes on the basis of the following assumptions: (1) The flow over each individual cylinder barrel leaves the barrel through the four connector tubes adjacent to it; for the case where a connector tube is common to two cylinders, the flow through that connector tube is equally supplied by each barrel; and (2) the flow through the pressure-equalizing holes is zero.

The tests were repeated with the connector-tube orifices removed in order to check the effect on the flow distribution. For each total flow to the bank the pressure drop across every barrel inlet was the same for both the runs with and without the connector-tube orifices in place. The installation of the orifices in the connector tubes therefore did not affect the coolant-flow distribution.

The pressure drop across the entire bank was also measured over the range of total flows investigated and a calibration (fig. 12) was computed for a 30-70 ethylene glycol-water mixture at 245° F. This calibration was used in conjunction with the pressure drops across the banks measured during the engine investigation to determine the division of flow between the banks.

### Engine Investigation

Cylinder temperatures, and coolant and oil heat rejections were determined at the following sea-level engine conditions:

| Power condition | Rated engine power (bhp) (a) | Engine speed (rpm) | Manifold pressure (in. Hg abs.) | Manifold temperature (°F) | Mixture setting | Coolant flow (gal/min) | |
|---|---|---|---|---|---|---|---|
| | | | | | | High | Low |
| 55-percent normal rated (minimum bsfc) | 580 | 1600 | 40 | 82 | Auto. lean | 111 | 91 |
| Maximum cruise | 850 | 2400 | 42 | 120 | Auto. lean | 184 | 134 |
| Normal rated | 1050 | 2700 | 46 | 138 | Auto. rich | 189 | 151 |
| Take-off or military rating | 1490 | 3000 | 61 | 164 | Auto. rich | 209 | 168 |
| War emergency | 1650 | 3000 | 67 | 174 | Auto. rich | 209 | 168 |

[a]Actual powers measured differed slightly.

At each power condition one run was made at each of the two coolant flows listed. These coolant flows are considered to correspond to the extremes of those encountered in typical airplane installations and were determined from consideration of the pump characteristics and the coolant-system resistance of several airplane installations. The coolant-outlet temperature was regulated to $247° ±1°$ F and a block-outlet pressure of 30 pounds per square inch gage was maintained for all runs. The oil-inlet temperature was held at $173° ±6°$ F and the carburetor air was controlled at $80° ±3°$ F. The pressure at the carburetor inlet and in the exhaust header was maintained at one atmosphere. The engine was operated in low blower and

the aftercooler coolant temperature was regulated to give the manifold temperatures listed. These manifold temperatures are the rated values given by the manufacturer for the various power conditions.

The heat rejections to the coolant and oil were determined in two ways: (1) from the measured flow and temperature rise of the coolant and the oil, and (2) from the measured flow and temperature rise of the coolant- and the oil-cooling waters.

## RESULTS AND DISCUSSION

### Coolant-Flow Distribution Data

The results of the flow distribution tests on the cylinder bank are shown in figure 13, where the flow to each cylinder is given as a percentage of the total flow to the bank. This curve applies over the entire range of flows tested.

It can be seen from the figure that the flow pattern for the cylinder barrels is irregular. The flow over the barrels ranged from about 12 percent of the total flow for cylinder 5 to 32 percent for cylinder 1. The flow over the heads progressively increased from cylinder to cylinder with cylinder 6 receiving about 12 percent and cylinder 1 receiving about 76 percent of the total coolant flow to the bank.

The division of flow between the two banks remained constant for the entire range of total flows from 91 to 209 gallons per minute. Bank A received 47 percent and bank B 53 percent of the total coolant flow to the engine.

### Cylinder-Temperature Data

The cylinder-temperature data is summarized in table I. The maximum, minimum, and average temperatures for each thermocouple location are listed for each run and the cylinder in which the maximum and the minimum temperatures occurred are noted in each case.

Temperature distribution. - Figures 14 and 15 show the temperature distribution among the cylinders at the various thermocouple locations for minimum brake specific fuel consumption and war-emergency-power conditions, respectively. The temperature-distribution patterns for both power conditions are similar, but the

spread (difference between the maximum and minimum temperature)
increased with power. For the head temperature between the exhaust
valves, the spread ranged from approximately 40° F (419° to 458° F)
at 630 brake horsepower to approximately 60° F (488° to 546° F) at
1690 brake horsepower. Comparison of these temperature-distribution
patterns with the coolant-flow distribution presented in figure 13
shows that the temperature-distribution patterns for the cylinder
head between the exhaust valves, in the center of the head, and at
the exhaust-valve guide appear to be affected by the coolant-flow-
distribution pattern; whereas those for the cylinder head between
the intake valves, on the exhaust-spark-plug gasket, and at the top
of the barrel on the exhaust side do not. The temperature-
distribution patterns for the exhaust-spark-plug-gasket location are
more erratic than those for the other locations in the cylinder heads.
Subsequent tests have shown that an individual gasket temperature may
vary as much as ±30° F depending upon the location of the thermocouple
on the periphery of the spark-plug gasket or upon the amount of
exhaust-gas leakage past the exhaust-stack gaskets.

For all the conditions investigated, the maximum temperature was
measured in the cylinder head between the exhaust valves on either
one or the other of cylinders 6. At approximately 1700 brake horse-
power, this maximum temperature was 546° F and occurred in cylinder A6.
Subsequent unpublished tests on this engine have shown, however, that
a considerable increase in the cylinder temperatures with engine run-
ning time occurs (approximately 45° F in the hot portion of the cylin-
der head for 100 hr running time); hence an engine that has been run
longer than the one tested (total running time for this engine, 16 hr)
would probably be hotter for the same engine and coolant conditions.

Variation of temperatures with engine power. - The variation of
the average cylinder temperature with engine power for a high and low
coolant flow at each power investigated is presented in figure 16.
In general, the greater the magnitude of the temperature for a certain
location with respect to other locations the greater the temperature
increase with increase in engine power. Thus, for an increase in
engine power from 628 brake horsepower to approximately 1700 brake
horsepower, the average temperature in the cylinder head between the
exhaust valves increased about 80° F (437° to 517° F) and the average
spark-plug-gasket temperature increased about 90° F (from 420° to
508° F); whereas the increase in average temperature at the top of
the barrel, exhaust side, was negligible (from about 265° to 270° F).
In spite of the erratic temperature-distribution pattern of the
exhaust-spark-plug-gasket temperatures, the average spark-plug-gasket

data gave a good indication of the temperature level and changes in
the temperature of the cylinder head between the exhaust valves. A
change in the coolant flow from one extreme to the other of those con-
sidered typical of current aircraft installations resulted in negli-
gible change in the average cylinder temperature in each case.

Comparison of temperatures at various locations. - The relation
between the average cylinder temperature and the temperature of the
hottest cylinder for the various locations in the cylinder head where
the temperatures were measured is shown in figure 17. All of the
cylinder temperature data fall on the same straight line except that
measured at the exhaust-valve guide, which was probably influenced by
the friction between the valve and the guide. The difference between
the maximum and average temperatures for all of the locations except
the exhaust-valve guide varied from about 10° to about 30° F, depend-
ing upon the magnitude of the temperatures involved.

The relation between the average temperature in the various loca-
tions of the cylinder and the average temperature in the cylinder head
between the exhaust valves is shown in figure 18. In every case a
linear relation was obtained and the slope of the line is greater for
the locations where the magnitude of the temperature is greater.


Heat-Rejection Data

The heat balance between the oil and the oil-cooling water and
the coolant and coolant-cooling water is shown in figure 19. Good
agreement between the two methods of determining the heat rejections
is seen to exist.

The variation of the heat rejections with engine power is shown
in figure 20. For an increase of engine power from 630 to approxi-
mately 1700 brake horsepower, the engine-coolant heat rejection
increased from 12,600 to 22,700 Btu per minute, the oil heat rejection
from 1030 to 4600 Btu per minute, and the aftercooler-coolant heat
rejection from 450 to 3500 Btu per minute. Based on a percentage of
brake horsepower, the engine-coolant heat rejection decreased from 47
to 31 percent, the oil heat rejection increased from 4 to 6 percent
and the aftercooler-coolant heat rejection increased from 2 to 5 per-
cent for this same increase in power. A change in the coolant flow
from one extreme to the other of those considered to be typical of
current aircraft installations caused no change in the heat rejec-
tions.

## SUMMARY OF RESULTS

An investigation of the cooling characteristics of the
V-1650-7 engine at typical operating conditions for a range of powers
from approximately 630 to 1700 brake horsepower with a mixture of
30-percent ethylene glycol and 70-percent water as the coolant gave
the following results:

1. The coolant-flow distribution among the cylinder barrels was
irregular. The flow over the individual barrels ranged from about
12 to about 32 percent of the total flow to the bank. The flow over
the cylinder heads progressively increased from cylinder to cylinder
in each bank with cylinder 6 receiving about 12 percent and cylinder 1
receiving about 76 percent of the total flow to the bank.

2. The temperature-distribution patterns appeared to be affected
by the coolant-flow distribution pattern for the temperatures meas-
ured in the cylinder head between the exhaust valves, in the center
of the head, and at the exhaust-valve guide, but did not appear to
be affected by the coolant-flow distribution pattern for the tempera-
tures measured in the cylinder head between the intake valves, on the
exhaust-spark-plug gasket, and at the top of the barrel on the exhaust
side. The difference between the maximum and minimum temperatures at
a particular location increased with engine power. A linear relation
was found to exist between the maximum and average temperatures at
each location where the temperatures were measured, and between the
average temperatures for all locations.

3. For all the conditions investigated, the maximum temperature
was measured in the cylinder head between the exhaust valves on either
one or the other of cylinders 6. At approximately 1700 brake horse-
power, this maximum temperature was 546° F and occurred in cylinder A6.

4. The average temperature for the cylinder heads between the
exhaust valves increased from 437° to 517° F for a power increase from
628 to approximately 1700 brake horsepower.

5. For a change in power from 628 to approximately 1700 brake
horsepower, the engine-coolant heat rejection increased from 12,600
to 22,700 Btu per minute, the oil heat rejection from 1030 to 4600 Btu
per minute, and the aftercooler-coolant heat rejection from 450 to
3500 Btu per minute. Based on a percentage of brake horsepower,

the engine-coolant heat rejection decreased from 47 to 31 percent, the oil heat rejection increased from 4 to 6 percent, and the aftercooler-coolant heat rejection increased from 2 to 5 percent for this same increase in power.


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<div align="center">REFERENCE</div>

1. Koffel, William K., and Bierman, Arnold E.:  A Unit Laboratory
    Engine Oil System Providing for a Remote Indication of Oil Flow
    and Oil Consumption Together with Blow-by Measurement.  NACA TN
    No. 952, 1944.

TABLE I - SUMMARY OF CYLINDER-TEMPERATURE

| Power condition at sea level | Measured engine power (bhp) | Engine speed (rpm) | Manifold pressure (in. Hg absolute) | Mixture setting | Fuel-air ratio | Carburetor-air temperature (°F) | Manifold temperature (°F) | Oil inlet temperature (°F) | Spark setting (deg B.T.C.) | Block-outlet pressure (lb/sq in. gage) | Coolant flow (gal/min) | Coolant-out temperature (°F) |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 55-percent normal rated (minimum bsfc) | 630 | 1602 | 40.05 | Auto. lean | 0.075 | 78 | 79 | 168 | 46.2 | 30 | 91 | 247.0 |
| | 628 | 1596 | 40.05 | Auto. lean | 0.075 | 80 | 81 | 168 | 46.2 | 30 | 111 | 247.5 |
| Maximum cruise | 924 | 2399 | 42.20 | Auto. lean | 0.072 | 80 | 117 | 168 | 46.3 | 30 | 134 | 248.1 |
| | 924 | 2398 | 42.15 | Auto. lean | 0.072 | 82 | 119 | 168 | 46.3 | 30 | 184 | 247.9 |
| Normal rated | 1099 | 2695 | 46.25 | Auto. rich | 0.078 | 83 | 136 | 174 | 46.4 | 30 | 151 | 248.0 |
| | 1101 | 2699 | 46.20 | Auto. rich | 0.078 | 82 | 135 | 174 | 46.4 | 30 | 189 | 246.8 |
| Take-off or military rating | 1506 | 2999 | 60.80 | Auto. rich | 0.093 | 83 | 161 | 179 | 45.4 | 30 | 168 | 248.0 |
| | 1526 | 3005 | 60.70 | Auto. rich | 0.093 | 82 | 160 | 176 | 45.4 | 30 | 209 | 247.2 |
| War emergency | 1698 | 3027 | 66.90 | Auto. rich | 0.090 | 82 | 173 | 176 | 44.4 | 30 | 168 | 247.9 |
| | 1690 | 2999 | 66.90 | Auto. rich | 0.090 | 82 | 172 | 175 | 44.4 | 30 | 209 | 248.1 |

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DATA OBTAINED ON THE V-1650-7 ENGINE

| Cylinder temperature (°F) | | | | | | | | | | | | | | | | | |
| Cylinder head between exhaust valves | | | Cylinder head between intake valves | | | Center of head | | | Exhaust-valve guide | | | Top of cylinder barrel, exhaust side | | | Exhaust spark-plug gasket | | |
| Max-imum | Min-imum | Aver-age | Max-imum | Min-imum | Aver-age | Max-imum | Min-imum | Aver-age | Max-imum | Min-imum | Aver-age | Max-imum | Min-imum | Aver-age | Max-imum | Min-imum | Aver-age |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 458 (B6) | 419 (B2) | 438 | 411 (B3) | 387 (B1) | 398 | 405 (B3,5,6) | 375 (B1) | 392 | 331 (A6) | 280 (B1) | 312 | 298 (B6) | 249 (A2) | 267 | 439 (A5) | 398 (B3) | 422 |
| 458 (B6) | 417 (A1) | 437 | 408 (B3) | 389 (B1,A1) | 396 | 404 (A6,B5,6) | 374 (A1) | 390 | 331 (B6) | 276 (B1) | 308 | 294 (B6) | 248 (A2) | 263 | 434 (A5,B5) | 395 (A1,B3) | 417 |
| 502 (A6) | 461 (B2) | 478 | 426 (B3,5) | 406 (A2) | 415 | 433 (B5) | 397 (A2) | 415 | 347 (A5) | 297 (A1) | 322 | 278 (B6) | 249 (A2) | 263 | 468 (A3) | 422 (B3) | 450 |
| 503 (A6) | 451 (A2) | 477 | 424 (B5) | 403 (A2) | 414 | 434 (A6) | 393 (A2) | 415 | 350 (A6) | 284 (B1) | 318 | 274 (B6) | 247 (A2) | 261 | 467 (B3) | 419 (B1) | 448 |
| 516 (A6,B6) | 465 (A1) | 492 | 439 (B5) | 420 (A2) | 429 | 446 (B5) | 405 (A2) | 427 | 354 (A6) | 291 (B1) | 325 | 276 (B6) | 247 (A2) | 263 | 491 (A3) | 440 (B3) | 469 |
| 518 (A6) | 462 (A1) | 492 | 438 (B5) | 415 (A2) | 428 | 448 (B5,6) | 402 (A1) | 427 | 355 (A6) | 288 (B1) | 321 | 277 (B3,5) | 246 (A2) | 263 | 488 (A3) | 440 (B3) | 468 |
| 528 (A6) | 473 (A1) | 500 | 446 (B5) | 421 (A6) | 434 | 452 (B6) | 408 (A1) | 432 | 358 (A6) | 293 (B1) | 327 | 286 (B6) | 247 (A2) | 266 | 523 (A3) | 471 (A1) | 492 |
| 528 (A6) | 471 (A1) | 500 | 445 (B5) | 418 (A5) | 433 | 455 (B5) | 404 (A2) | 432 | 358 (A6) | 288 (B1) | 322 | 289 (B5) | 248 (A2) | 266 | 515 (A3) | 443 (B3) | 489 |
| 545 (A6) | 487 (A2) | 517 | 465 (B5) | 437 (A6) | 451 | 466 (B5) | 420 (A2) | 445 | 360 (A6) | 296 (B1) | 332 | 289 (B5) | 249 (A2) | 271 | 540 (A3) | 462 (B3) | 509 |
| 546 (A6) | 488 (A1) | 517 | 462 (B3) | 435 (A5) | 448 | 468 (B5) | 416 (A2) | 444 | 364 (A6) | 292 (B1) | 327 | 306 (B5) | 249 (A2) | 269 | 534 (A3) | 475 (A2) | 506 |

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Fig. 1

Outlet

Connector
tubes

Front

Pressure-equalizing holes

Coolant branch tube

Inlet
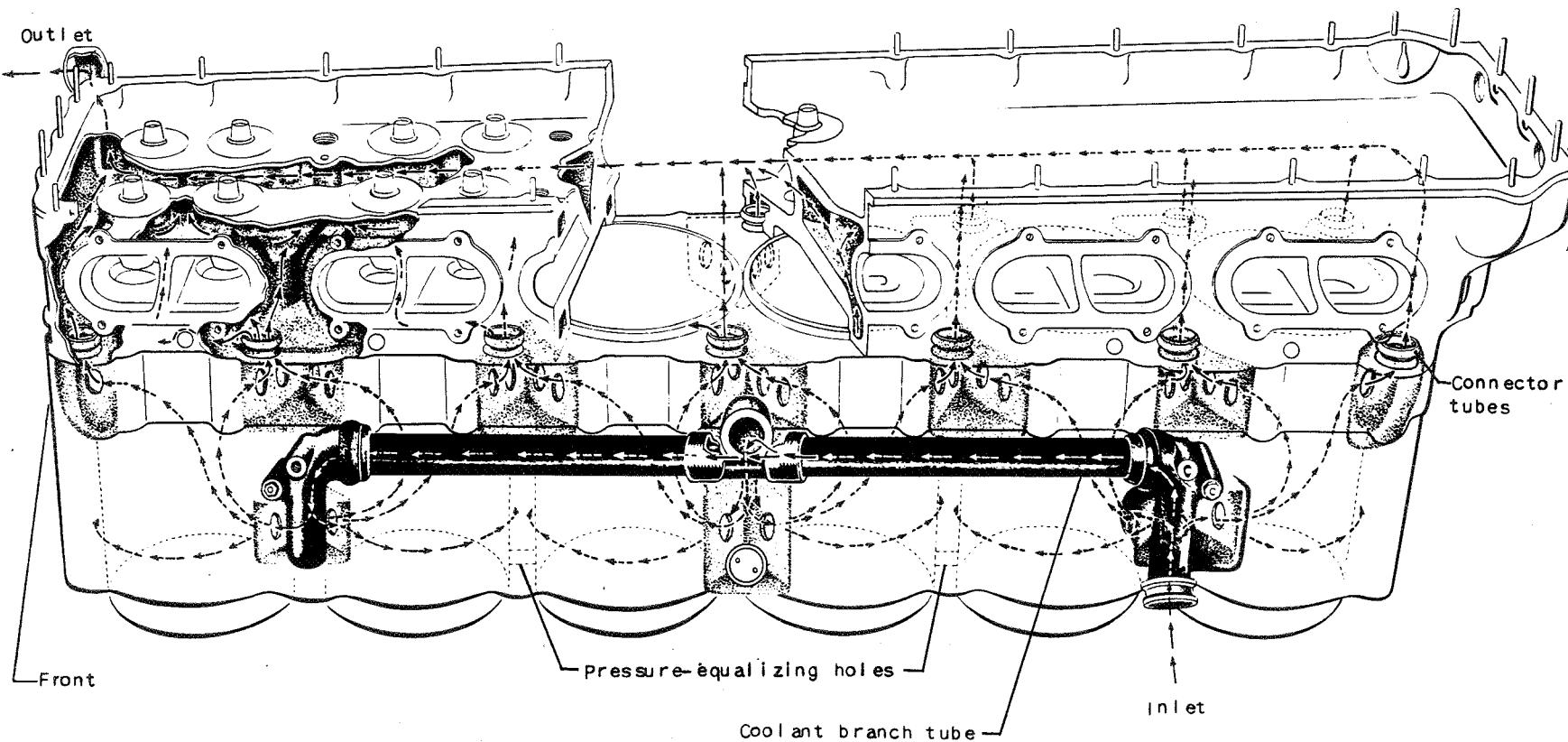
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Figure 1. - Sketch of cylinder bank showing network of coolant passages.

Outlet

7,14    6,13    5,12    4,11    3, 10    2,9    1,8    Connector tubes

Cylinder heads

Front

Cylinder barrels

Engine coolant pump

Coolant branch tube

To other bank

Coolant path
Pressure-equalizing holes
Pressure tap

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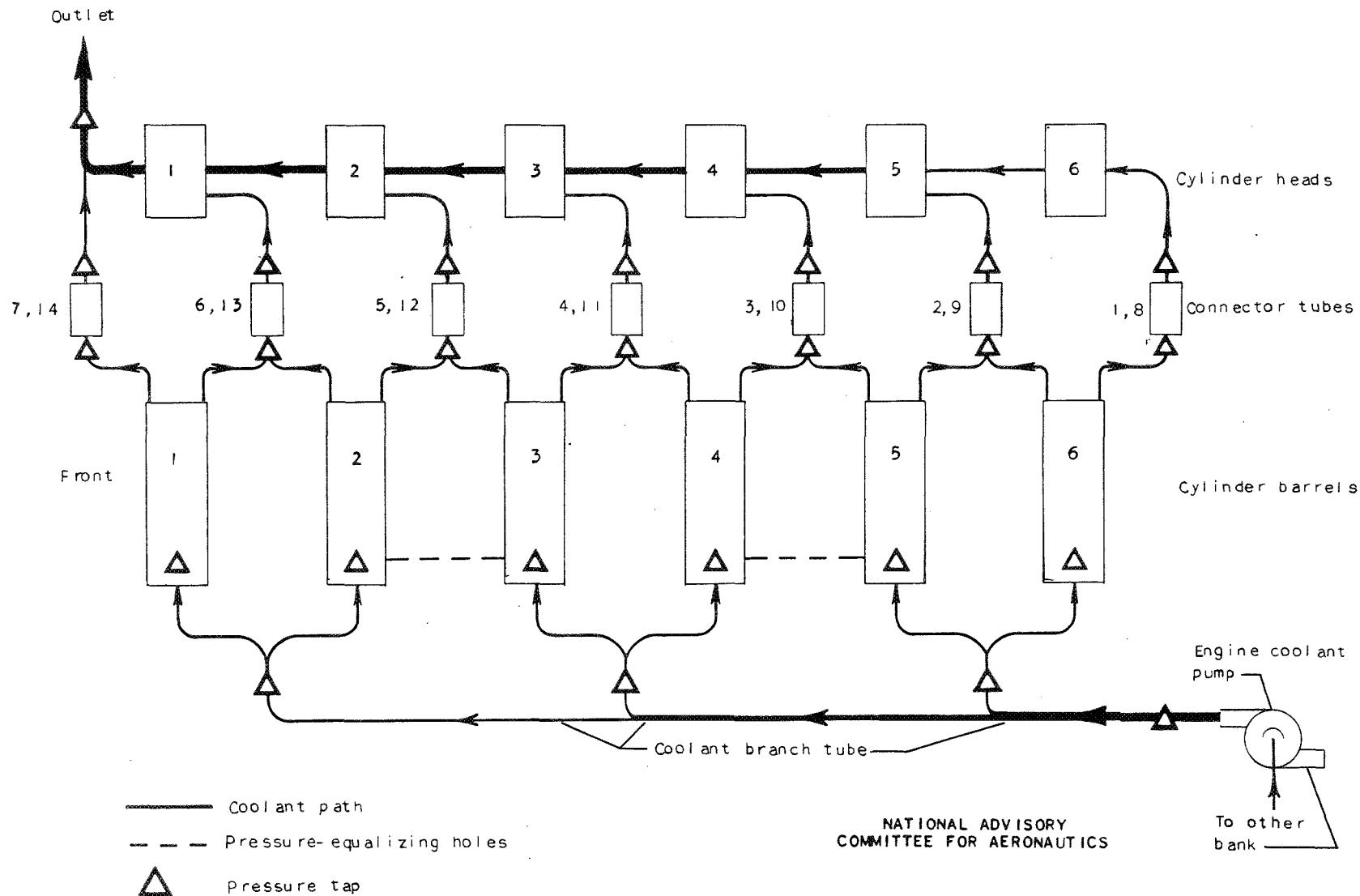
Figure 2. - Schematic diagram of cylinder bank showing path of coolant flow.

Fig. 2
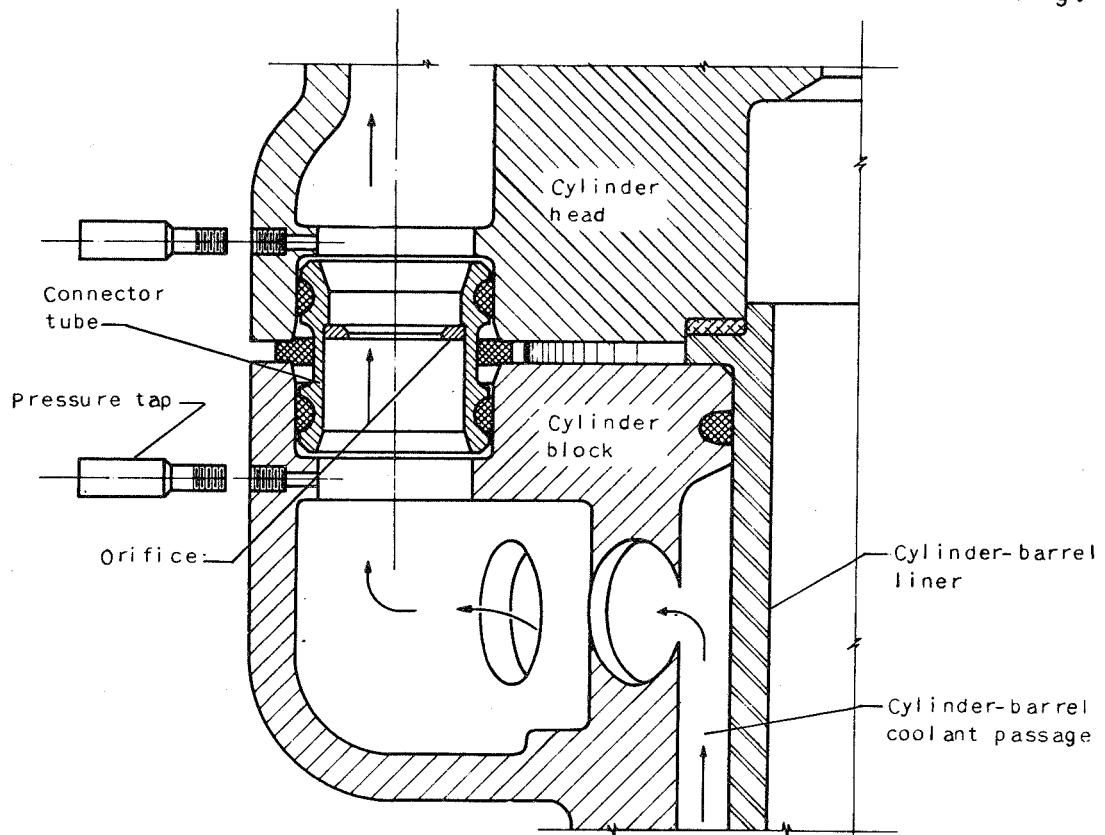
(a)   Connector-tube orifice and pressure-tap installation.
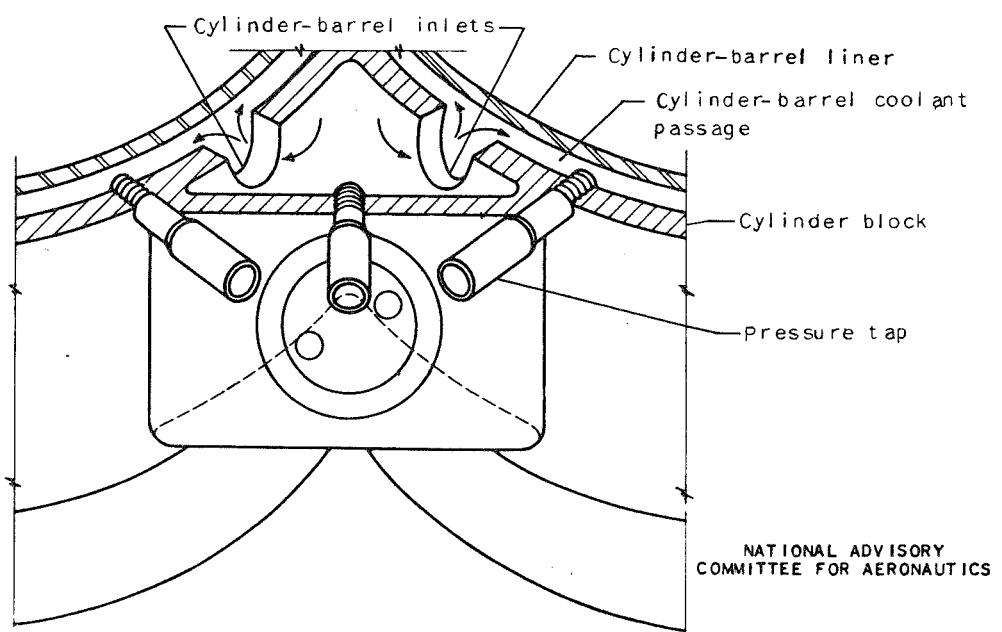
(b)   Cylinder-barrel-inlet pressure-tap installation.

Figure 3. - Orifice and pressure-tap installation.

Connector tubes

7    6    5    4    3    2

14    13    12    11    10    9    8

Cylinder    1    2    3    4    5    6

Fig. 4
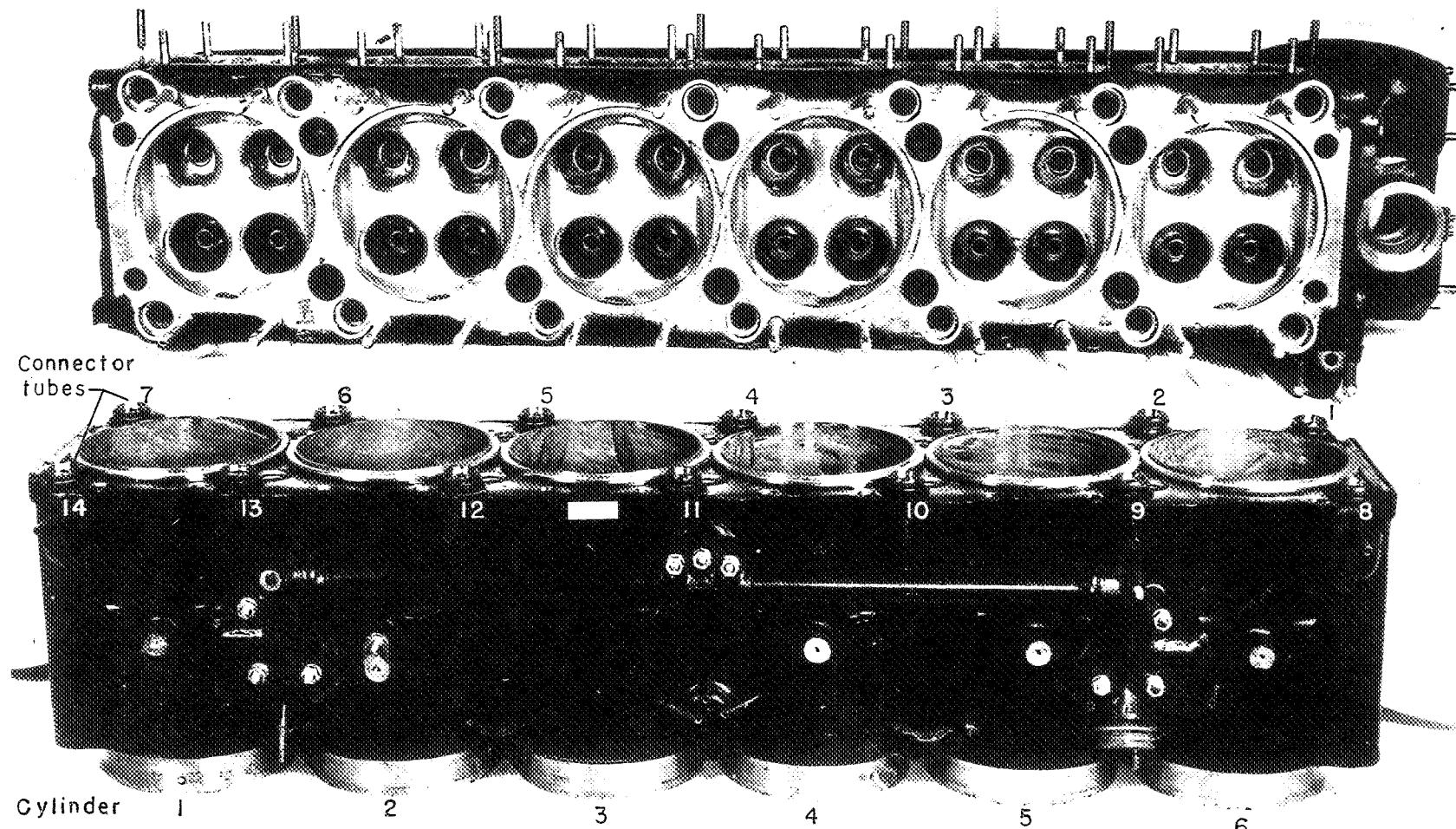
Figure 4. - Pressure-tap installation in cylinder bank.
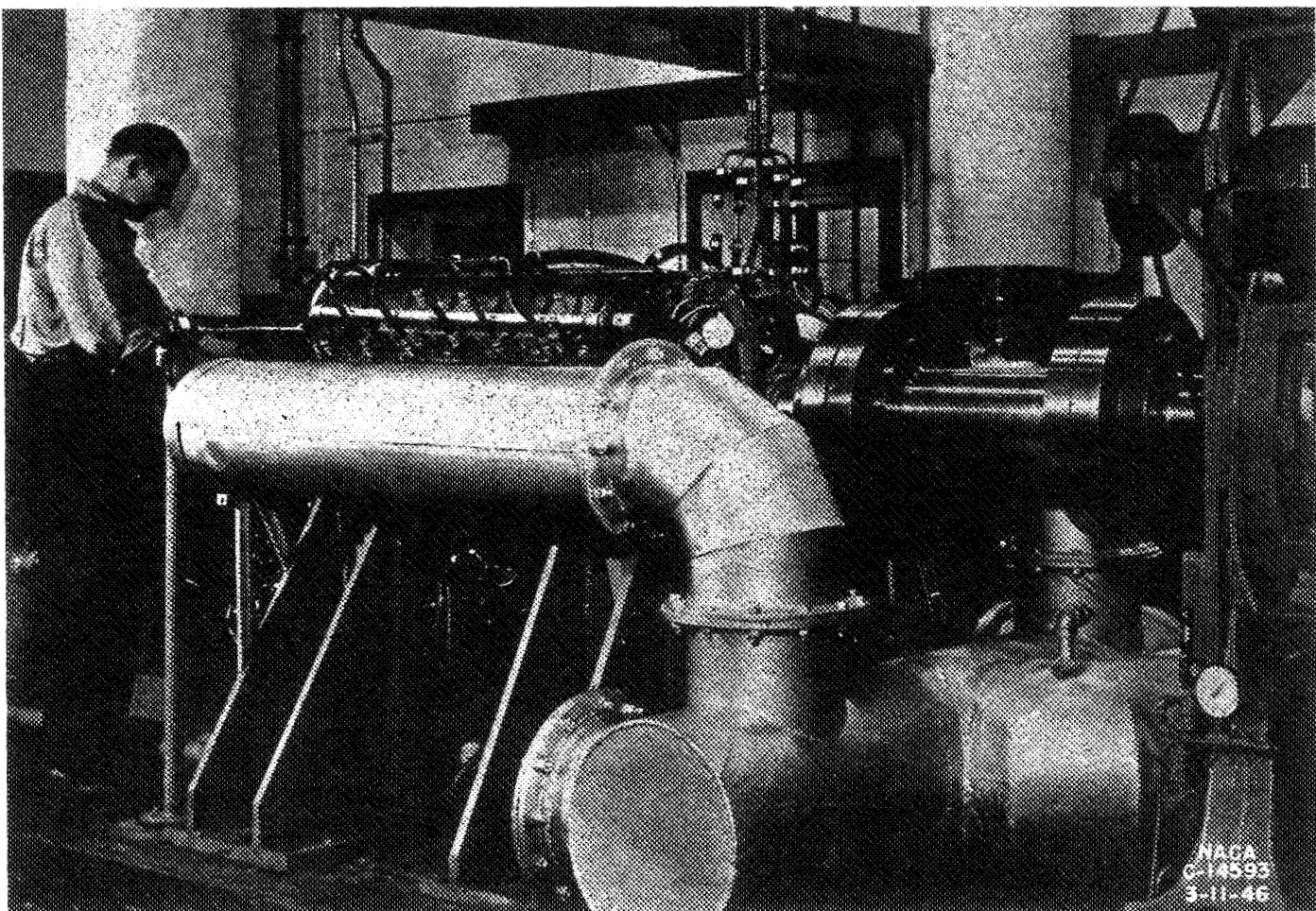
Figure 5. - Engine mounted for cooling investigation.

Fig. 5

Fig. 6

Compressed-air line

Bleed line

Vapor-separator vent line

Block-outlet vent line

Expansion tank

Bank A

Bank B

Front

Vapor
separator

Engine pump

Venturi

Bypass line

Three-way mixing valve

Coolers

Strainer

Cooling water line

Auxiliary pump
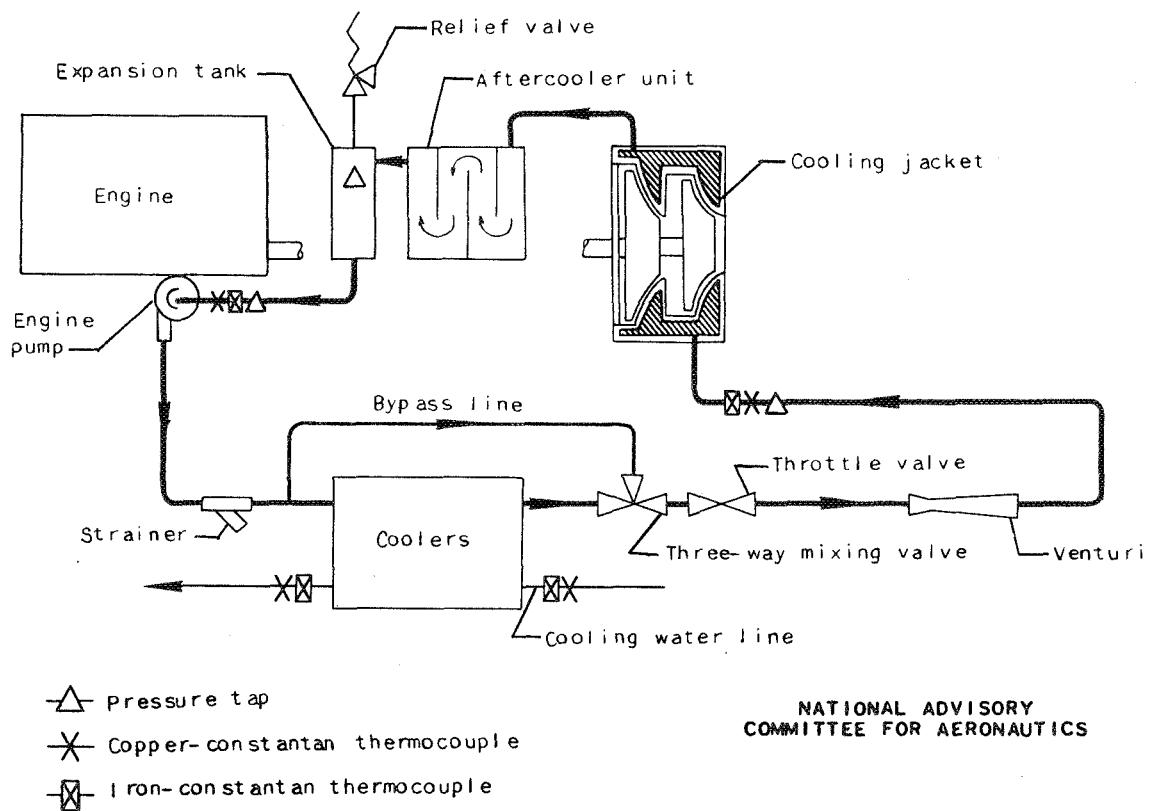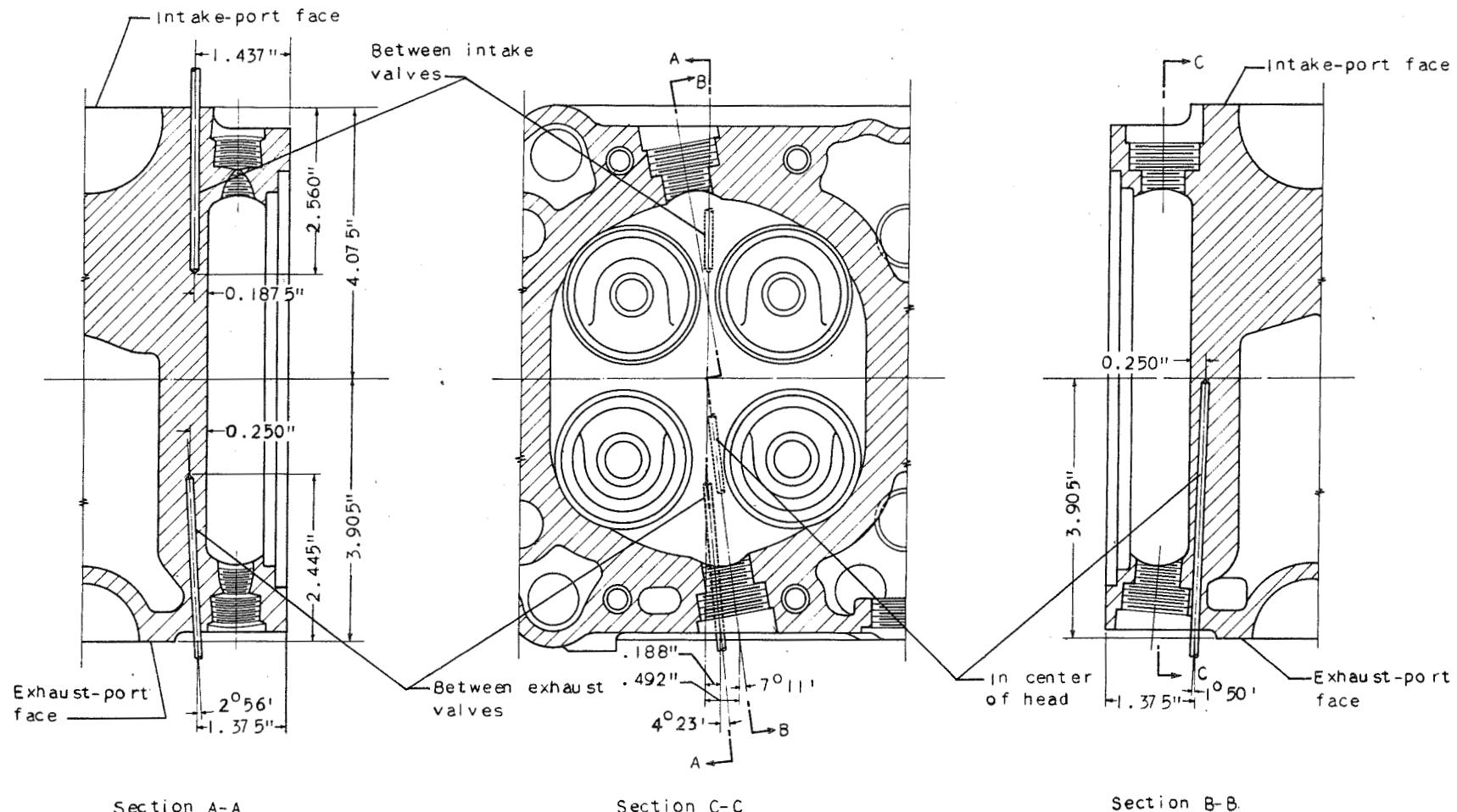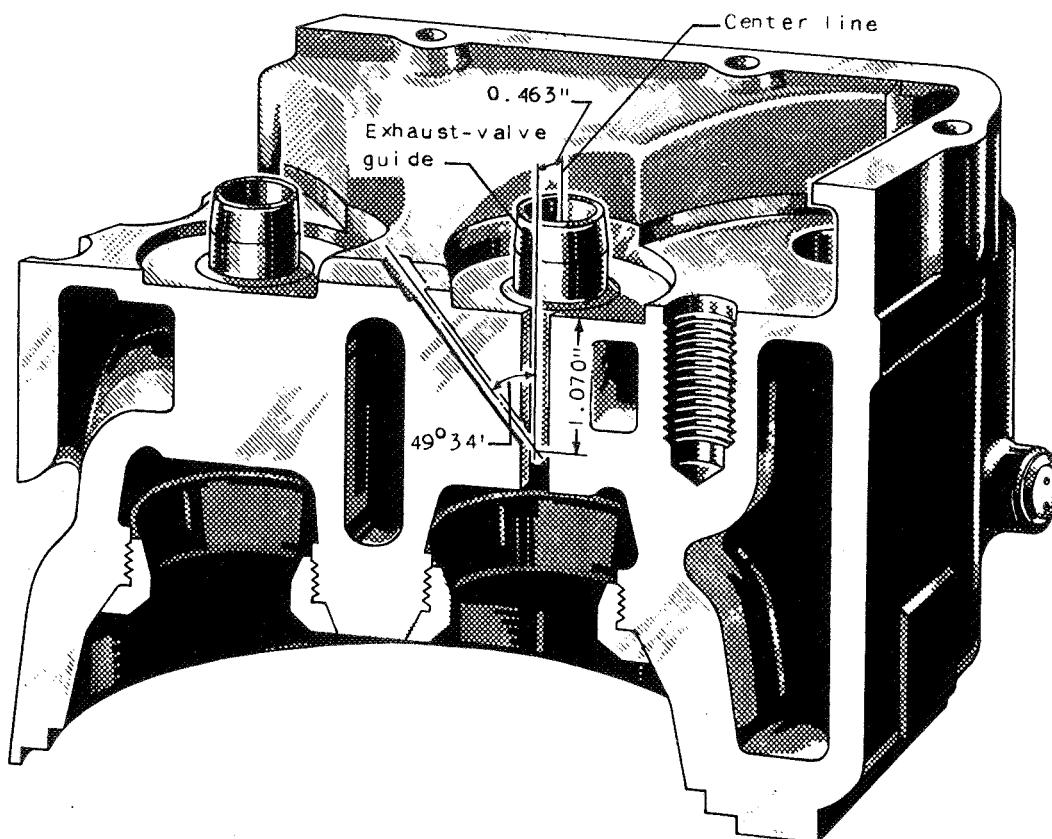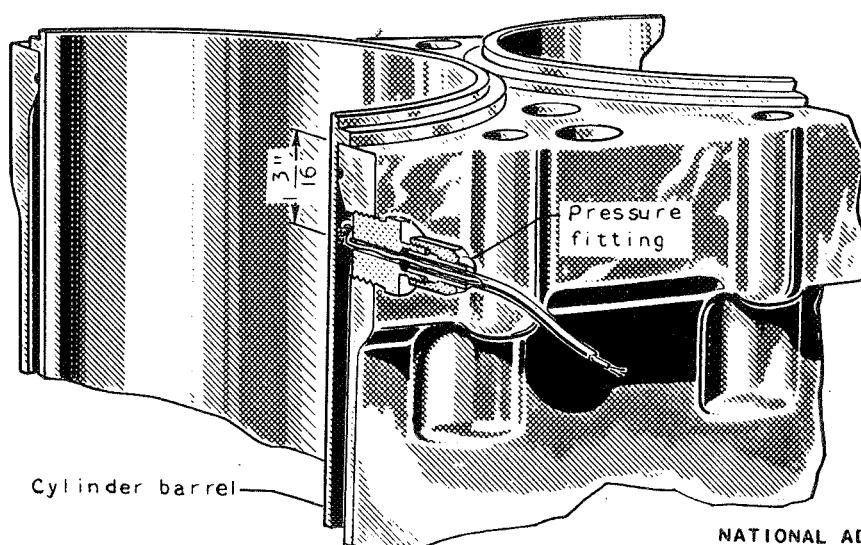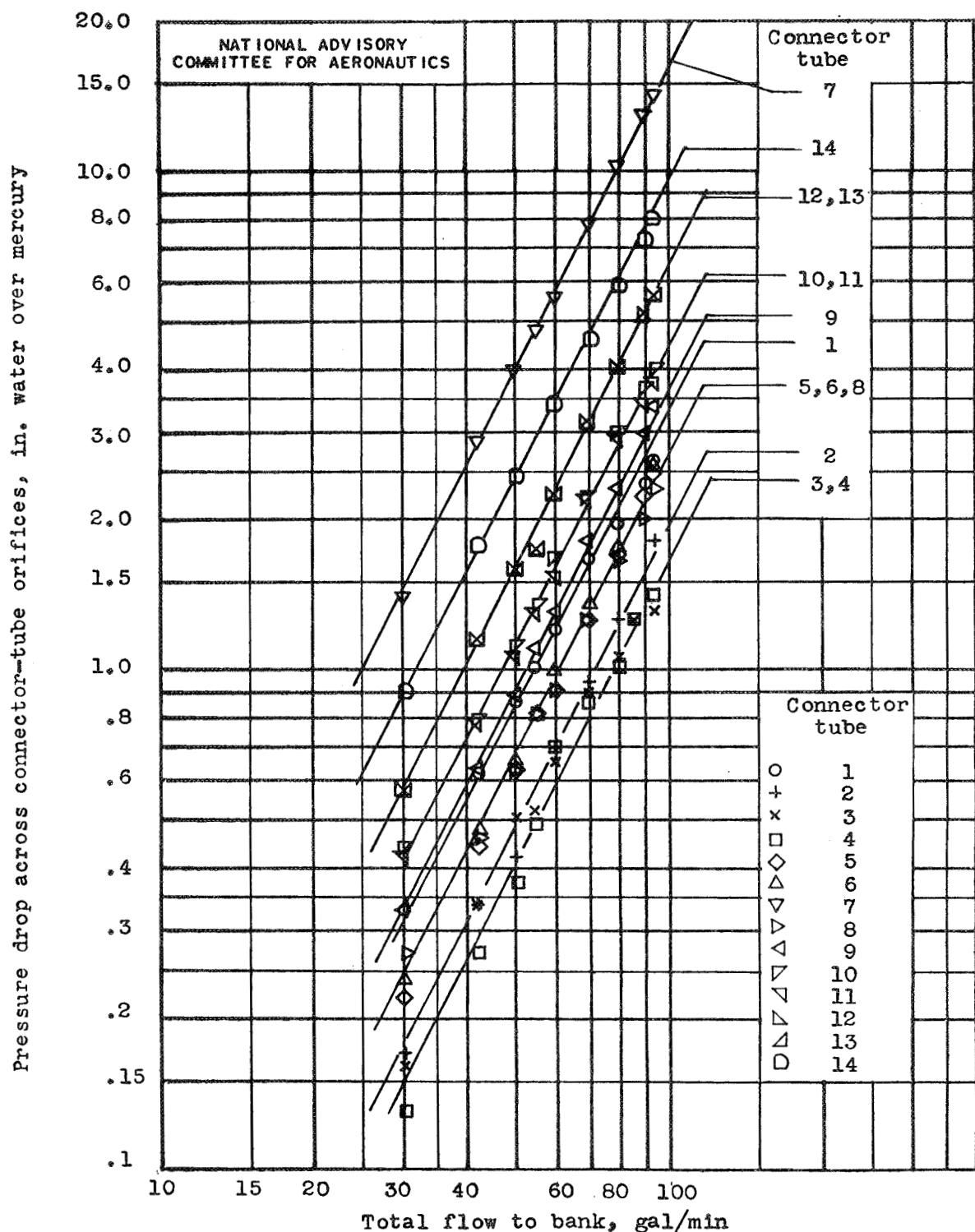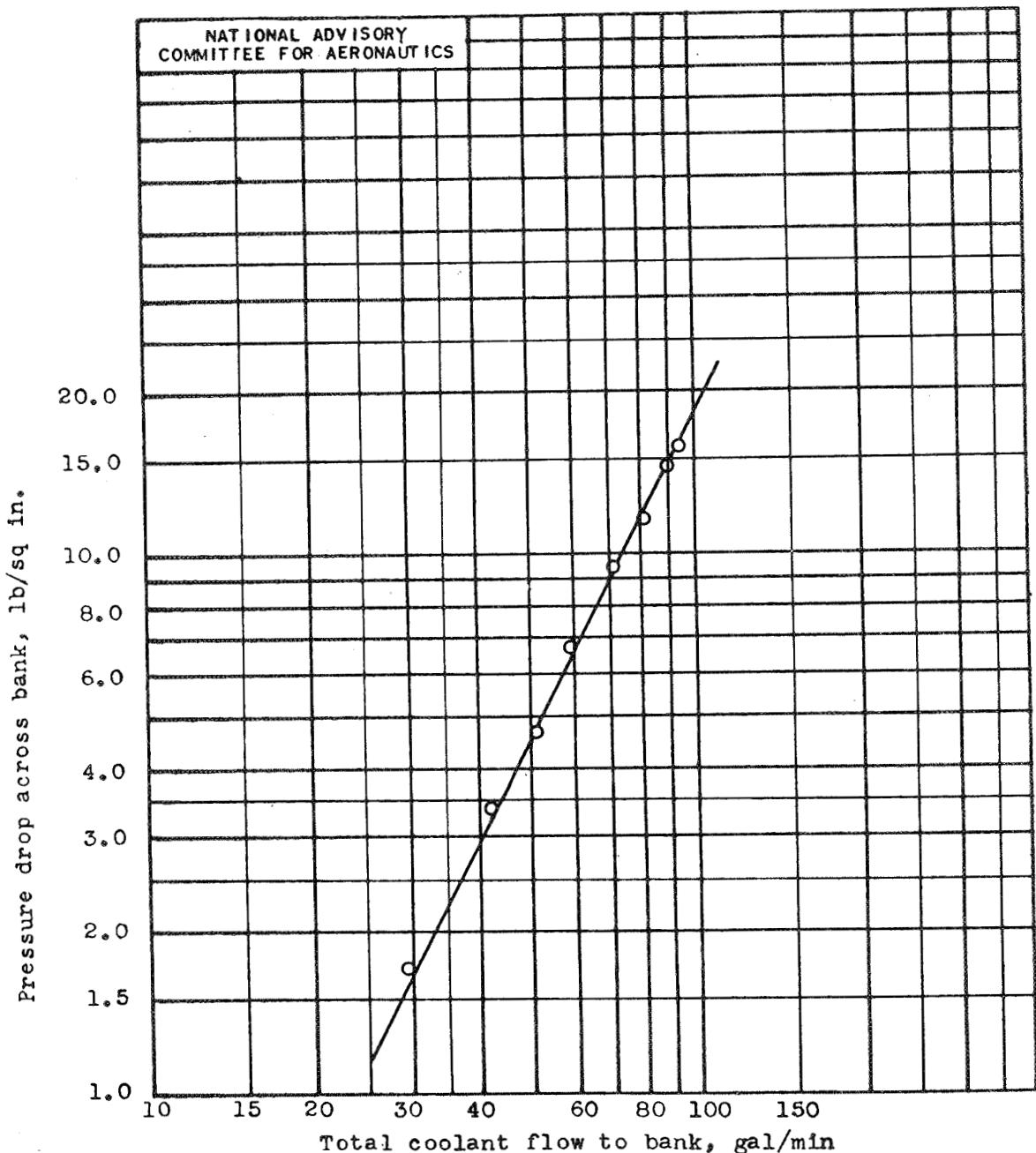
~~~~~~  Main lines

△  Pressure tap

▭  Sight glass

✕  Copper-constantan thermocouple

⊠  Iron-constantan thermocouple

⋈  Throttle valve

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Figure 6. - Schematic diagram of engine coolant system.

Reservoir tank

Engine pump

Engine

Bypass line

Filter unit

Chip detector

Coolers

Cooling water line

Three-way
mixing valve

⊲ Pressure tap

✕ Copper-constantan thermocouple

⊠ Iron-constantan thermocouple

Figure 7. - Schematic diagram of lubricating oil system.

Relief valve

Expansion tank

Aftercooler unit

Engine

Cooling jacket

Engine
pump

Bypass line

Throttle valve

Strainer

Coolers

Three-way mixing valve

Venturi

Cooling water line

⊲ Pressure tap

✕ Copper-constantan thermocouple

⊠ Iron-constantan thermocouple

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Figure 8. - Schematic diagram of aftercooler coolant system.

Figure 9. — Installation of cylinder-head thermocouples.

Fig. 9

(a)   Exhaust-valve guide.



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(b)   Top of barrel, exhaust side.

Figure 10. – Installation of thermocouples at exhaust-valve
guide and top of barrel on exhaust side.

Figure 11. — Flow calibration of connector-tube orifices with water at room temperature.

Figure 12. — Variation of pressure drop across bank with coolant flow
for 30—70 ethylene glycol—water mixture at 245° F.

Figure 13. – Coolant-flow distribution through cylinder bank.

Figure 14. — Temperature distribution among cylinders for minimum brake
   specific fuel consumption power condition.  Engine power, 630 brake
   horsepower; engine speed, 1602 rpm; fuel-air ratio, 0.075; manifold
   temperature, 79° F; coolant-out temperature, 247° F; coolant flow,
   91 gallons per minute; block-outlet pressure, 30 pounds per square
   inch gage.

Figure 15. — Temperature distribution among cylinders for war-emergency-power condition. Engine power, 1690 brake horsepower; engine speed, 3000 rpm; fuel-air ratio, 0.090; manifold temperature, 172° F; coolant-out temperature, 248° F; coolant flow, 209 gallons per minute; block-outlet pressure 30 pounds per square inch gage.

Figure 16. — Variation of the average cylinder temperatures with engine power.

Figure 17. — Variation of average cylinder temperature with temperature of hottest cylinder.

Figure 18. - Variation of average cylinder temperature with average
cylinder-head temperature between exhaust valves.

Figure 19. — Heat balance between oil and oil-cooling water and
coolant and coolant-cooling water.

Figure 20. — Variation of heat rejections with engine power.