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

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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.
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 5.602 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° B.T.C. for full-closed throttle to approximately 45° 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 coolant 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 combination on the coolant expansion tank. A centrifugal-type vapor separator 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 outlets 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 calibrated 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 lubricating 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 incorporated 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 potentiometer. 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 thermocouple 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 thermocouples 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:

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

*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 horsepower, 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 running time occurs (approximately 45° F in the hot portion of the cylinder 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 temperature 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 lo-
cations 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 measured 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 temperatures 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 horsepower, 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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Mechanical Engineer.

Benjamin Pinkel,
Physicist.

REFERENCE

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<tr>
<th>Power condition at sea level</th>
<th>Measured engine power (bhp)</th>
<th>Engine speed (rpm)</th>
<th>Manifold pressure (in. Hg absolute)</th>
<th>Mixture setting</th>
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<th>Carborator-air temperature (°F)</th>
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<th>Oil inlet temperature (°F)</th>
<th>Spark setting (deg B.T.C.)</th>
<th>Blockout pressure (lb/sq in. gage)</th>
<th>Coolant flow (gal/min)</th>
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DATA OBTAINED ON THE V-1650-7 ENGINE

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<th>Cylinder head between intake valves</th>
<th>Center of head</th>
<th>Exhaust-valve guide</th>
<th>Top of cylinder barrel, exhaust side</th>
<th>Exhaust spark-plug gasket</th>
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<td>438 (B6)</td>
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<td>418 (A5)</td>
<td>433</td>
</tr>
<tr>
<td>545 (A6)</td>
<td>487 (A2)</td>
<td>517</td>
<td>465 (B5)</td>
<td>437 (A6)</td>
<td>451</td>
</tr>
<tr>
<td>546 (A6)</td>
<td>488 (A1)</td>
<td>517</td>
<td>482 (B5)</td>
<td>435 (A5)</td>
<td>448</td>
</tr>
</tbody>
</table>
Figure 1. - Sketch of cylinder bank showing network of coolant passages.
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Average cylinder-head temperature between exhaust valves, °F

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