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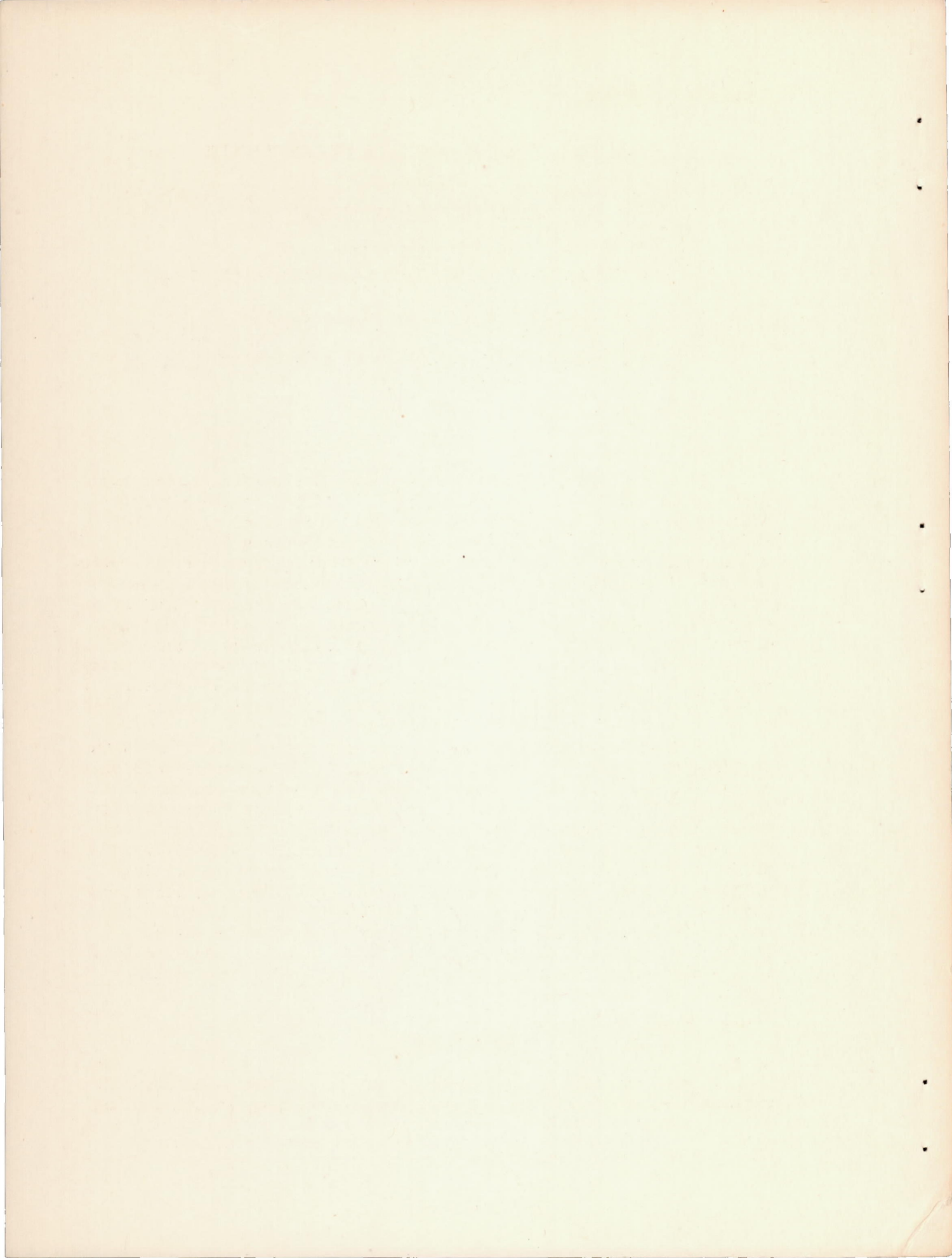
SYSTEMS FOR A LIQUID-COOLED ENGINE

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ADVANCE RESTRICTED REPORT

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SUMMARY

Tests have been conducted to determine the performance of two pressurized shunt-type cooling systems for liquid-cooled aircraft engines using a mixture of 30 percent AN-E-2 ethylene glycol and 70 percent water as the coolant. One of the systems (system A) employed an expansion tank typical of those in current use on unpressurized AN-E-2 ethylene glycol systems; the other system (system B) used an expansion tank designed by the Linde Air Products Company and modified for production by the Bell Aircraft Corporation. Neither system incorporated a venturi nor other pressure-boost arrangement at the pump inlet. Coolant-flow rates were determined for both systems over (a) a range of engine speeds at constant pump-inlet pressure and (b) a range of expansion-tank pressures at constant engine speed.

The results of tests of the two cooling systems show that with the expansion tank of system B higher pump-inlet pressures (approximately 6 lb/sq in.) and attendant better pump performance may be obtained than with the expansion tank used in system A for any fixed expansion-tank pressure. For an expansion-tank pressure equal to the vapor pressure of the coolant at a block-outlet temperature of 250° F, a coolant-flow rate of 255 gallons per minute is obtained with system B as compared with a flow rate of 200 gallons per minute obtained with system A. Both of these flow rates are adequate to cool 12-cylinder liquid-cooled engines at present power ratings. The higher pump-inlet pressures of system B, however, resulted in a more stable and safer operation of the coolant pump than could be obtained with system A.

INTRODUCTION

Recent increases in the power output of liquid-cooled aircraft engines have indicated the necessity for improvement in the cooling

of this type of engine. Although satisfactory cooling of American-built engines has been obtained with an unpressurized shunt-type cooling system using AN-E-2 ethylene glycol as the coolant, the results of unpublished tests conducted at the NACA Cleveland laboratory on an Allison V-1710 engine indicate that the cylinder-head temperatures in this engine approach limiting values for safe operation at present power ratings and exceed them at projected higher power ratings. The results of these tests also indicate that appreciable reductions in cylinder-head temperatures may be obtained by the use of a mixture of 30 percent ethylene glycol and 70 percent water as the coolant.

The relatively low boiling point of the 30-70 glycol-water mixture necessitates the use of a pressurized cooling system if coolant temperatures (250° F and higher) compatible with reasonable radiator size are to be used. Pressurized systems of the series type have been utilized for some time in British Rolls Royce engine installations and more recently in the American-built Packard versions of the Rolls Royce engine. Pressurized systems of the shunt type theoretically possess inherent advantages over pressurized systems of the series type; the principal advantages of the shunt-type system are higher pump-inlet pressures and concomitant superior pump performance for a given expansion-tank pressure. No experimental data are available, however, on their performance.

As part of a research program being conducted by the NACA on the cooling of liquid-cooled engines, the performance of pressurized shunt-type cooling systems has been investigated by means of engine tests using a mixture of 30 percent AN-E-2 ethylene glycol and 70 percent water as the coolant. Two different shunt-type systems were tested: system A used an expansion tank typical of those in current use on unpressurized AN-E-2 ethylene glycol systems; system B used an expansion tank designed by the Linde Air Products Company and modified for production by the Bell Aircraft Corporation. The tests described herein were conducted at Cleveland during June and July 1944 and consisted of a determination of coolant-flow rates over (a) a range of engine speeds at constant pump-inlet pressure and (b) a range of expansion-tank pressures at constant engine speed.

THEORY OF COOLING SYSTEMS

Much of the theory on cooling systems discussed herein has been presented in references 1 and 2. The satisfactory operation of a cooling system for a liquid-cooled aircraft engine depends upon the ability of the system to keep the flow rate as high as required to maintain the engine temperatures within safe limits. In order to

obtain adequate coolant-flow rates and stable pump operation it is
necessary that the pressure at the pump inlet be higher than the
vapor pressure of the coolant by an amount sufficient to prevent
excessive pump cavitation. The tendency of a pump to cavitate is
usually considered as a function of the "pressure proximity" at the
pump inlet; the pressure proximity is defined as the difference
between the absolute static pressure of the liquid and the vapor
pressure of the liquid at the same point. This parameter may be
used to correlate the effect of liquid composition and temperature
on the cavitation characteristics of a pump.

Because the decrease of flow rate caused by pump cavitation is
due to the formation of vapor within the pump, the presence of
entrained air or vapor in the coolant will have the same effect on
the pump performance as too low an inlet pressure. It is therefore
essential that satisfactory air-vapor separators be incorporated in
the cooling system.

In an aircraft-engine cooling system the main coolant flow is
from the pump to the engine, through the engine to a radiator, and
through the radiator back to the pump. An expansion tank is located
at the engine outlet. Two different circuits, the series circuit
and the shunt circuit, are in current use. These circuits differ
mainly in the manner in which the expansion tank is connected into
the system. A schematic diagram of a series circuit is shown in fig-
ure 1(a). In this circuit the expansion tank is located in the main
coolant line between the engine and the radiator. The shunt circuit
is schematically shown in figure 1(b). The flow in this circuit
differs from that of the series circuit in that only a small portion
of the flow from the engine, about 1 or 2 percent, goes to the
expansion tank. This flow through the expansion tank does not go
through the radiator but returns directly to the pump inlet.

In any cooling system, the pressure in the expansion tank deter-
mines the absolute pressure level of the entire system. In a pres-
surized system the expansion-tank pressure is maintained higher than
the atmospheric pressure by means of a pressure cap fitted to the
tank. This pressure cap usually incorporates a pressure-relief valve
to relieve excessive pressures and a vacuum or "sniffle" valve to
protect the system against excessively low pressures. Because it is
difficult to keep these valves and other piping connections from
leaking slightly and because variations in flight conditions and
engine power cause the coolant temperature to change, the highest
pressure that can be reliably maintained in the expansion tank is
the vapor pressure of the coolant at the block-outlet temperature.

The vapor pressure of the coolant at the pump inlet is lower than the vapor pressure in the expansion tank owing to the temperature drop across the radiator. This temperature drop is not very large (about 10° F); hence the decrease in coolant vapor pressure between the expansion tank and the pump inlet is small. This small change in coolant vapor pressure therefore limits the pressure proximity at the pump inlet to a relatively small amount. The absolute pressure at the pump inlet is equal to the expansion-tank pressure plus the differential-elevation head minus any pressure losses in the shunt line. It is therefore desirable to have as low a line pressure drop as possible between the expansion tank and the pump inlet. In this respect, the shunt system is better than the series system because the difference in pressure between the expansion tank and the pump inlet is not decreased by a pressure drop through the radiator and the main coolant piping. It is important, however, that the tank be so designed that the pressure loss in the shunt line does not become excessive.

The pump-inlet pressure may be increased by the use of special arrangements, such as applying heat to the expansion tank or by a venturi boost at the pump inlet, in which the shunt line from the expansion tank is connected to the throat of a venturi and the static pressure at the pump inlet is thereby increased by the amount of pressure recovery of the venturi. It should be noted, however, that the most desirable pressure proximity at the pump inlet is that value just sufficient to give satisfactory flow conditions. A system with an unnecessarily high pressure level is undesirable because of the difficulty of maintaining tight engine-jacket seals and piping connections.

As the engine power is varied, the temperature rise of the coolant through the engine, and hence the pressure proximity at the pump inlet, will vary for constant engine speed. If the pump is operating under cavitating conditions but still in a relatively stable region, the variation in the pressure proximity will result in a change of flow rate with a change in power; that is, an increase in power will raise the flow rate and a decrease of power will lower the flow rate.

APPARATUS

Engine Installation

The tests were conducted with an Allison V-1710-81 multicylinder engine mounted on a dynamometer stand equipped with a 2000-horsepower eddy-current dynamometer. Oil, refrigerated air, and atmospheric exhaust were supplied to the engine at specified conditions by auxiliary equipment.

Cooling Systems

System using unpressurized-type expansion tank. - Cooling
system A, a conventional shunt type, used an expansion tank of the
type in current use on unpressurized AN-E-2 ethylene glycol sys-
tems. Air-vapor separators were added to the system and two ver-
sions of their installation were tested: (a) when installed in
the main coolant lines at the block outlets and (b) when installed
in the shunt lines at the block outlets. Figure 2 is a schematic
diagram of the system showing the air-vapor separators installed
in the main coolant lines together with a cross-sectional drawing of
the expansion tank. The version of the system with the air-vapor
separators installed in the shunt lines at the block outlets is
shown schematically in figure 3. A cross-sectional view of the
expansion tank, which was modified for tests of this version of
the system by the installation of a horizontal baffle inside the
tank, is also shown in figure 3.

The system was constructed to simulate the installation in a
typical pursuit-type military airplane with regard to length, size,
and resistance of piping. The coolant flow divides at the pump
outlet, follows parallel paths through each engine block and cooler,
and returns to a Y connection at the pump inlet. The shunt circuit
of the system consists of separate shunt lines at each block outlet,
which join and enter the expansion tank, and a return line from the
bottom of the expansion tank to the pump inlet.

The shunt lines from the block outlets, which were fitted with
1/8-inch orifices to restrict the flow, were joined together and
connected to the expansion tank as shown in figures 2 and 3. The
lines were mounted in a horizontal plane in order to decrease the
possibility of air traps. The common line from the block shunt
lines was extended halfway down into the tank and discharged tan-
gentially along the inner wall. The vent lines from the vapor sep-
arators were connected to each side of the expansion tank. The
return line from the bottom of the expansion tank was connected to
the pump-inlet cover as shown in figures 2 and 3.

The expansion tank was mounted between the cylinder blocks and
was fitted with an altitude-compensated pressure-relief valve set to
relieve at 35 pounds per square inch absolute. The horizontal baf-
fle, which was fitted to the tank for the tests of the second ver-
sion of the system, was installed in an attempt to improve vapor and
air separation from the liquid in the tank by shielding the inlet of
the pump return line from the direct discharge of the vent lines.
The main-line air-vapor separator shown in figure 4(a) is of the cen-
trifugal type and has a ratio of inlet area to throat area of 3:1.
The shunt-line air-vapor separator shown in figure 4(b) was made by

flattening a bend in the tubing to provide a reduction of area at
the throat and adding a vent line to the inside radius of the bend.
The Allison shrouded-impeller coolant pump fitted with the Y inlet
cover was used. The pump speed is 1.234 times the engine speed.
The coolant temperature was controlled by means of a water-cooled
heat exchanger and an air-operated bypass valve installed in the
water line. The normal pressure drop across the system from the
pump outlet to the pump inlet was about 34 pounds per square inch
at a flow rate of 250 gallons per minute.

The coolant-flow rate was determined by two similar venturi
tubes installed in the main coolant lines of each block. System
pressures were measured at the locations indicated in figures 2 and
3 by calibrated Bourdon-tube gages. Iron-constantan thermocouples
connected to a self-balancing potentiometer were used to measure
the coolant temperature at the cylinder-block outlets, at the expan-
sion tank, and at the pump inlet from the right bank. Sight glasses
installed in the main coolant lines, in the vent lines, and in the
expansion tank return line, at the locations shown in figures 2 and
3, permitted observation of the coolant. A sight glass and an elec-
tric light were also installed in the baffled expansion tank to per-
mit observation of the internal-flow conditions. Compressed-air
lines were connected to the main coolant lines at the locations indi-
cated for aerating the coolant in order to test the vapor separators
and to control the pressure of the system at the desired levels.

System using Linde expansion tank. - The cooling system using
the Linde expansion tank (system B) is the same as system A except
that no main and shunt-line air-vapor separators were installed and
the expansion tank is different. Figure 5 is a schematic diagram
of system B. The expansion tank (figs. 6 and 7) consisted essen-
tially of an outer or main tank inside of which was located a cen-
trifugal air-vapor separator. This air-vapor separator received
the flow from the block shunt lines and discharged it directly to
the pump inlet. The outer expansion tank was also connected to the
pump inlet by a static line. Because the flow from the block shunt
lines did not enter the outer tank (except for the negligible flow
from the separator vent), there was practically no flow in this
static line. The air-vapor separator, which was at a higher pres-
sure than the outer static tank, was provided with a restricted
vent in order to obviate excessive flows into the outer tank and
through the static line. Heat supplied by the hot coolant in the
air-vapor separator maintained the temperature of the liquid in the
outer tank at approximately the same temperature as the outlet cool-
ant, thus assuring maximum coolant-vapor pressure in the tank.

In order to permit the installation of the expansion tank in
the engine used in these tests it was necessary to fabricate a tank

at the Cleveland laboratory that was 2 inches shorter than the Bell
production model of the Linde design. The same style pressure-relief
valve was used as on the expansion tank of system A and a screen
was installed, as shown in figure 6, to prevent dirt from passing
into the valve and depositing on the neoprene seal.

Tests were conducted with two different sets of flow-restriction
orifices in the block shunt lines. The first set was 1/8 inch in
diameter and the second set was 3/16 inch in diameter. A compressed-
air line and a blow-off cock were connected to the expansion tank at
the location shown in figure 6.

The instrumentation was the same as for the tests of system A
except for measurement of the coolant condition at the pump inlet
and the flow rate through the shunt lines. Mercury manometers were
connected to the pump suction lines from both banks and an additional
thermocouple was so installed in the left bank suction line that the
coolant temperature and pressure could be determined in both of the
pump-suction lines. Manometers were fitted across each of the flow-
restriction orifices in the shunt lines to measure the flow rate to
the expansion tank. The complete shunt-line assembly was subsequently
calibrated by a bench test.

Venturi-Calibration Setup

One of the venturi tubes used to measure the coolant-flow rate
was calibrated by a bench test over the range of flow rates and pres-
sure proximities encountered in the engine tests. A sight glass was
installed immediately downstream from the venturi to permit observa-
tion of the flow. Both the flow rate and the upstream pressure on
the venturi were controlled by valves located at the venturi entrance
and downstream from the sight glass. Differential manometers were
fitted across both the venturi and the sight glass; single-tube
manometers were connected to the entrance of both the venturi and the
sight glass to provide a complete pressure survey. The flow rate was
determined by a weighing tank.

TEST PROCEDURE

Tests were conducted to determine coolant-flow rates for both
systems over a range of engine speeds at constant pump-inlet pres-
sure and over a range of expansion-tank pressures at constant engine
speed. The following table gives a summary of the test conditions
for both systems:

System	Engine power (bhp)	Engine speed (rpm)	Pump-inlet pressure (lb/sq in. abs.)	Flow-restriction orifice in shunt line (in.)
	Variable engine speed			
A	Constant throttle (635 to 725)	2000 to 3000	27.9	1/8
B	725 ±35	2000 to 3000	27.0	1/8
			28.0	1/8
			28.0	3/16
	Variable expansion-tank pressure			
A	1035 ±5	3000 ±5	26.1 to 32.8	1/8
B	735 ±5	3000 ±5	23.2 to 31.2	1/8

The coolant-outlet temperature was held at 250° ±2° F and the carburetor-mixture control was maintained in automatic-rich position for all runs.

For the tests of system A, the expansion-tank pressure was increased by the introduction of compressed air into the system at the main block outlet line; coolant was discharged through a bleed-off valve in order to decrease the pressure. During the tests of system B the expansion-tank pressure was increased by admitting compressed air into the tank and decreased by bleeding-off vapor from the tank through a blow-off valve. For the variable engine-speed runs of both systems the pump-inlet pressure was maintained constant by adjusting the expansion-tank pressure.

The pump-inlet pressure taps were located about 1 foot upstream from the pump suction to avoid error from the uneven pressure distribution in the long-radius elbows. The measurements of pump-inlet pressure thus obtained were subsequently corrected for this line loss and therefore the data presented herein represent the pressure immediately upstream from the Y suction connection of the coolant pump.

RESULTS AND DISCUSSION

Effect of engine speed on coolant-flow rate. - The variation of coolant-flow rate with engine speed for system A at a pressure proximity (static pressure minus vapor pressure) of 7 pounds per square inch at the pump inlet and for both types of air-vapor separator is shown in figure 8(a). Similar data for system B for two values of

the pressure proximity at the pump inlet and with two different sized orifices in the shunt lines are shown in figure 8(b). The data points of both systems have been corrected for slight variations in pressure proximity encountered in the tests.

Figure 8(a) shows that the coolant-flow rate of system A varies directly with engine speed up to nearly 3000 rpm, indicating that the pump operates almost free of cavitation at a pressure proximity of 7.0 pounds per square inch. A slightly higher flow rate, particularly at the higher engine speeds, is noted for system A with the air-vapor separators installed in the block shunt lines. The loss in flow rate caused by decreasing the pressure proximity from 6.0 to 5.0 pounds per square inch, shown in figure 8(b), is greater at the higher engine speeds than at the lower engine speeds because the higher velocity of the liquid through the pump results in greater entrance pressure losses, which increases the tendency to vapor formation. For a pressure proximity of 6.0 pounds per square inch (corresponding to a pump-inlet pressure of 28.0 lb/sq in. absolute), the coolant-flow rate varies directly with engine speed up to 2600 rpm, indicating the pump to be operating free from cavitation in this region; whereas for a pressure proximity of 5.0 pounds per square inch (corresponding to a pump-inlet pressure of 27.0 lb/sq in. absolute), a slight amount of cavitation is noted at all engine speeds above 2000 rpm. The change from 1/8-inch to 3/16-inch orifices in the shunt lines of system B has no apparent effect on the coolant-flow rate.

Effect of expansion-tank pressure on coolant-flow rate. The variation of coolant-flow rate with expansion-tank pressure is shown in figures 9(a) and 9(b) for systems A and B, respectively. The data for both types of vapor separator used in system A fall on one curve, indicating no significant difference in system operation between the two types of vapor separator. A maximum coolant-flow rate of about 275 gallons per minute was obtained for both systems at an expansion-tank pressure of about 32 pounds per square inch absolute. The coolant-flow rate falls off more rapidly with a decrease in expansion-tank pressure for system A, however, than for system B. The temperature of the liquid in the expansion tank was generally about 1° F lower than the block-outlet temperature. Thus, for a coolant-outlet temperature of 250° F the vapor pressure in the expansion tank is about 26 pounds per square inch absolute, resulting in a coolant-flow rate of about 200 gallons per minute for system A and of about 255 gallons per minute for system B. At this point the curve for system A is almost vertical, which indicates that a further slight reduction in expansion-tank pressure due to leaks in the tank or in other parts of the system would probably cause a breakdown of coolant flow. For system B, however, the flow-rate curve is sufficiently flat in this region to indicate stable and safe operation.

Relation between pump-inlet pressure and expansion-tank pres-
sure. - Figure 10 shows the relation between pump-inlet pressure
and expansion-tank pressure for systems A and B. Because the expan-
sion tank and the pump inlet are connected by a return line, or static
line, as shown in figures 2, 3, and 5, the pump-inlet pressure is
equal to the expansion-tank pressure plus the differential-elevation
head minus the line pressure drop. For any fixed expansion-tank pres-
sure the pump-inlet pressure is seen to be about 6.0 pounds per square
inch higher in system B than in system A, which results in a pump-
inlet pressure of about 27.5 pounds per square inch absolute in sys-
tem B as compared with a 21.5 pounds per square inch absolute pump-
inlet pressure in system A when normal (approximately 26 lb/sq in.
absolute) coolant vapor pressure exists in the expansion tank. This
difference is due to the fact that the shunt-line flow in system A
goes through the return line connecting the expansion tank to the
pump inlet with attendant line pressure losses, whereas in system B
the pump inlet is connected to the expansion tank by the static line
in which there is very little flow. The pressure losses in the return
line from the air-vapor separator of system B does not decrease the
pump-inlet pressure because the pressure within the separator is cor-
respondingly higher than the expansion-tank pressure. The differential-
elevation head between the pump and the expansion tank is the cause of
the pump-inlet pressure in system B being about 1.5 pounds per square
inch higher than the expansion-tank pressure.

Variation of coolant-flow rate with pump-inlet pressure. - Fig-
ures 11(a) and 11(b) show the variation of coolant-flow rate with
pump-inlet pressure for systems A and B, respectively. A corres-
ponding variation of flow rate is obtained when it is plotted against
the pressure proximity at the pump inlet, thereby giving the cavita-
tion characteristics of the pump. (See fig. 12.) The cavitation
curves obtained for both systems are similar and show that full liq-
uid flow is not obtained until a pressure proximity of about 8 pounds
per square inch is reached. As noted on the curves, when coolant vapor
pressure exists in the expansion tank, the pressure proximity at the
pump inlet is about 5 pounds per square inch higher for system B than
for system A. (The difference in the pressure proximity between the
two systems is not so great as the difference in the pump-inlet pres-
sure owing to the lower engine power for the tests of system B.) This
difference in pressure proximity, for the same expansion-tank pres-
sure, results in a greater degree of pump cavitation and a correspond-
ingly lower flow rate for system A than for system B, as was noted in
figure 9.

The system resistance did not remain constant during these tests
but increased about 25 percent as the pressure in the system was
reduced. This increase was due to an effect introduced by the venturi

in the pump suction line and probably caused the flow rate to start
to fall off at a higher pressure than would the curves of flow rate
for a constant system resistance. The cause of this variation in
system resistance and its effect on the performance of the system
will be discussed in greater detail later in this report.

Flow rate through the shunt lines. - The effect of engine speed
on the flow rate through the shunt lines of system B is shown in
figure 13(a). A relatively small flow rate, which increases with
engine speed, is noted for both sizes of flow-restriction orifices.

Figure 13(b) shows the effect of pump-inlet pressure on the
flow rate through the shunt lines. The pressure drop across the
shunt-line system is the difference between the block-outlet pres-
sure and the pump-inlet pressure; the small effect of pump-inlet
pressure on the flow rate is caused by an increase in the pump-inlet
pressure accompanied by an almost equal increase in the block-outlet
pressure.

Effect of coolant-flow rate on pressure drop. - The pressure
drops both through the entire system (pump outlet to pump inlet) and
from the block outlet to the pump inlet are shown as a function of
the total coolant-flow rate in figure 14(a) for system A and in fig-
ure 14(b) for system B. For the variable-speed runs, the pressure
drop increases with the square of the flow rate in the normal manner
and the agreement between the two systems is very good. The pres-
sure drop for the variable-pressure runs, however, is seen to remain
constant in system B over nearly the full range of flow rates tested;
for system A, the pressure drop remains constant over a considerable
range of flow rates and gradually decreases as the low flow rates
are reached. Inspection of the data showed that, for each system at
the point of intersection, the pump-inlet pressure for the variable-
speed curve is equal to that for the variable-pressure curve.

This unusual variation of pressure drop with flow rate during
the variable-pressure runs may be explained by the results of the
venturi-calibration tests. During the calibration tests at low pres-
sures, large quantities of vapor were observed in the sight glass,
indicating that considerable flashing or boiling of the coolant occur-
red in the venturi. As the upstream pressure on the venturi was
increased, the flashing was suppressed until finally full liquid flow
was obtained. The pressure survey through the venturi and sight glass
for both flashing and nonflashing runs is given in figure 15. For
clarity, only a few typical runs are presented. For the flashing runs,
the upstream pressure was adjusted to a value that resulted in recon-
version of two-phase flow (vapor and liquid) to full liquid flow in
the downstream portion of the sight glass. It is noted that for the

flashing runs the pressure recovery did not take place in the down-
stream cone of the venturi but occurred, to a limited extent (about
one-fourth of that obtained in nonflashing runs), in the region
where the reconversion to liquid flow occurred. This result is
attributed to boundary separation in the venturi cone and the attend-
ant eddy losses. The higher over-all pressure loss for the flashing
runs is shown in figure 16 in which the pressure drop from the ven-
turi entrance to the downstream end of the sight glass is plotted
against the flow rate. Figure 17 gives the venturi calibration for
both flashing and nonflashing runs. In spite of this wide variation
in flow conditions, all data points are seen to be within 8 percent
of the calculated calibration curve.

This higher over-all pressure loss through the venturi and down-
stream piping when falshing of the coolant occurred affords the expla-
nation of the unusual variation of pressure drop with flow rate noted
in figure 14. As the coolant-flow rate through the cooling systems
was reduced by decreasing the pump-inlet pressure, vaporization occur-
red in the venturi, which increased the resistance of the system. This
increase in system resistance, together with the decreased flow rate,
resulted in a constant over-all pressure drop over a considerable range
of flow rates. At the lower flow rates encountered in the tests of
system A, the pressure drop varies with flow rate in the normal man-
ner but is higher than the variable-speed pressure-drop curve. It is
believed that in this region the vaporization in the venturi reached
a maximum and did not contribute further increase to the system
resistance.

Effect of venturi on performance of the cooling system. -
From the foregoing considerations, it should be noted that actual
aircraft installations of the cooling systems (that is, without the
flow-measuring venturi) may be expected to perform with somewhat
higher flow rates at low pump-inlet pressures than is shown by the
test results presented herein because the effective system resistance
will not be increased by vaporization in the venturi. Inspection of
the data indicates that the maximum increase in system resistance
caused by vaporization in the venturi amounted to about 25 percent
of the normal resistance and occurred at a pump-inlet pressure of
about 24 pounds per square inch absolute (pump-inlet pressure prox-
imity of 2 lb/sq in.). At this point the coolant-flow rate without
the venturi in the system would be expected to be about 12 percent
higher than obtained in the tests. Inasmuch as the pump-inlet pres-
sure is controlled by the shunt-line branch of the system, the pres-
sure proximity at pump inlet is not affected by the presence of the
flow-measuring venturi in the main coolant line. System A, which
was found to be operating with a pressure proximity of less than
1 pound per square inch at the pump-inlet (see fig. 12(a)) when

coolant-vapor pressure existed in the expansion tank, would there-
fore still be operating in an unstable region even with the venturi
removed.

System operating experience. - Both systems were found to be
difficult to fill after they had been completely drained. In order
to fill the systems completely, it was necessary first to fill as
much as possible, run the engine for a short period, and then shut
down and add more coolant.

Although the expansion tanks were vented to the atmosphere
during filling, this difficulty was not completely eliminated because
the expansion tank is located at a lower level than the engine blocks.
It was therefore also necessary to vent the engine blocks to the
atmosphere. Connections for both of these vents have been provided
in a new type filler cap designed by the Bell Aircraft Corporation.
Both of these vents are closed off when filling is completed in order
to prevent circulatory flow of coolant from the filler cap into the
tank during operation and air from being drawn into the engine blocks
after a shutdown.

The effectiveness of the air-vapor separators was tested by
noting the removal of air and vapor during the warm-up period and
by introducing air into the main coolant stream at both block out-
lets immediately upstream of the sight glasses. The main-line sep-
arators of system A removed the large air and vapor bubbles in less
than 1 minute and completely removed small bubbles in fine suspen-
sion in about 5 minutes. The shunt-line separators required about
10 to 15 minutes to remove all the air and vapor. The internal air-
vapor separator of the expansion tank of system B did not give as
rapid separation as the main-line separators used in system A but
did provide quicker separation than the shunt-line separators.

For both systems small bubbles of air or vapor were observed
at all times in the return line from the expansion tank to the pump
inlet, which indicates a continual pickup of vapor from the expan-
sion tank into the system. The addition of the horizontal baffle
to the expansion tank of system A decreased the amount of vapor
pickup from the tank but did not completely eliminate this condi-
tion. Vapor was also present in the static line of the system B
tank, which indicates that the liquid in the tank was fairly well
agitated and mixed with the vapor. This condition was also noted
during blow-off of excess pressure in the tank. Although the blow-
off valve was connected to the upper section of the tank, a con-
siderable quantity of liquid was discharged at every blow-off. In
order to ameliorate this condition of vapor pickup, the Linde Air
Products Company have recently added a shield over the end of the

vapor-separator vent line in such a way that the stream issuing
from it will not agitate the liquid in the tank. It should be
noted that the outlet of this vent line should be below the level
of the liquid to prevent air from being drawn into the system after
a shutdown.

A slight leakage, which was encountered through the pressure-
relief valve at all times during the tests of system A, made it dif-
ficult to maintain high expansion tank pressures. The addition of
the screen under the pressure-relief valve of the system B tank,
which was installed in an attempt to eliminate the leakage, became
so clogged with solid material that it prevented the pressure-relief
valve from operating properly and was consequently removed.

SUMMARY OF RESULTS

The results of the engine tests described herein on two pres-
surized shunt-type cooling systems using a 30-70 glycol-water mix-
ture as coolant indicate that the main difference in the performance
of the two cooling systems is that, for any fixed expansion-tank
pressure, the pump-inlet pressure obtained in system B (using Linde
expansion tank) is about 6 pounds per square inch higher than that
obtained in system A (using unpressurized-type expansion tank). This
characteristic of the expansion tank used in system B is considered
to be its principal advantage inasmuch as the higher pump-inlet pres-
sures result in a high pressure proximity at the pump inlet and attend-
ant stable operation of the pump at relatively low expansion-tank pres-
sures and maximum system pressures. In current practice the highest
expansion-tank pressure that can be reliably maintained is the coolant-
vapor pressure corresponding to the block-outlet temperature.

The data obtained in these tests show that, when a coolant-vapor
pressure of 26 pounds per square inch absolute (corresponding to a
block-outlet temperature of 250° F) existed in the expansion tank,
a coolant-flow rate of approximately 200 gallons per minute was
obtained in system A and a coolant-flow rate of about 255 gallons
per minute in system B. Although both of these flow rates are ade-
quate to cool liquid-cooled aircraft engines at present rated powers,
at this condition in system A the pump is operating in a dangerous
and unstable region. The expansion-tank pressure curve for system A
shows that a drop of 1/2 pound per square inch below the value cor-
responding to the vapor pressure in the expansion tank would result
in a large decrease in coolant-flow rate, if not in a complete break-
down of the flow. This condition would also be present if the flow-
measuring venturi were not included in the system because, as noted
previously herein, the pump-inlet pressure is not influenced by the
pressure drop through the venturi.

When coolant-vapor pressure existed in the expansion tank of system B, however, the pump was operating in a stable region and, although a slight amount of cavitation occurred, no harmful effects are expected. Higher pump-inlet pressures might be obtained for the same expansion-tank pressure by the use of a pump-inlet venturi or other boost arrangement. These devices are considered neither necessary nor desirable for present purposes when 30-70 glycol-water mixtures are used, in view of the negligible improvement in cooling resulting from the small possible increase in flow rate and the attendant higher system pressure levels. In this connection, it should also be noted that pressures higher than those necessary to give stable operation of the pump are undesirable owing to the difficulty of maintaining tight piping connections and engine-jacket seals.

Aircraft Engine Research Laboratory,
 National Advisory Committee for Aeronautics,
  Cleveland, Ohio.

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2. Stamm, C. A., and McCravey, W. E.: Improvements in Cooling Systems. SAE Jour. (Trans.), vol. 52, no. 7, July 1944, pp. 34-35.
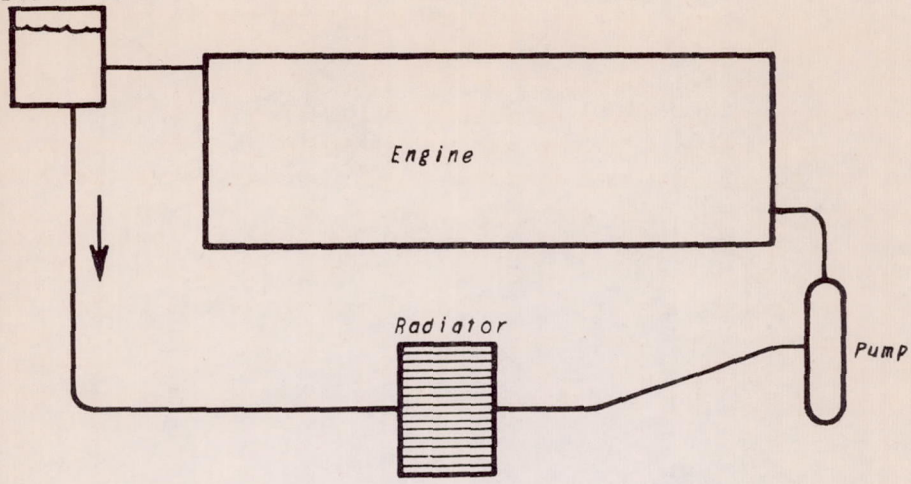
(a)   Series-type circuit.
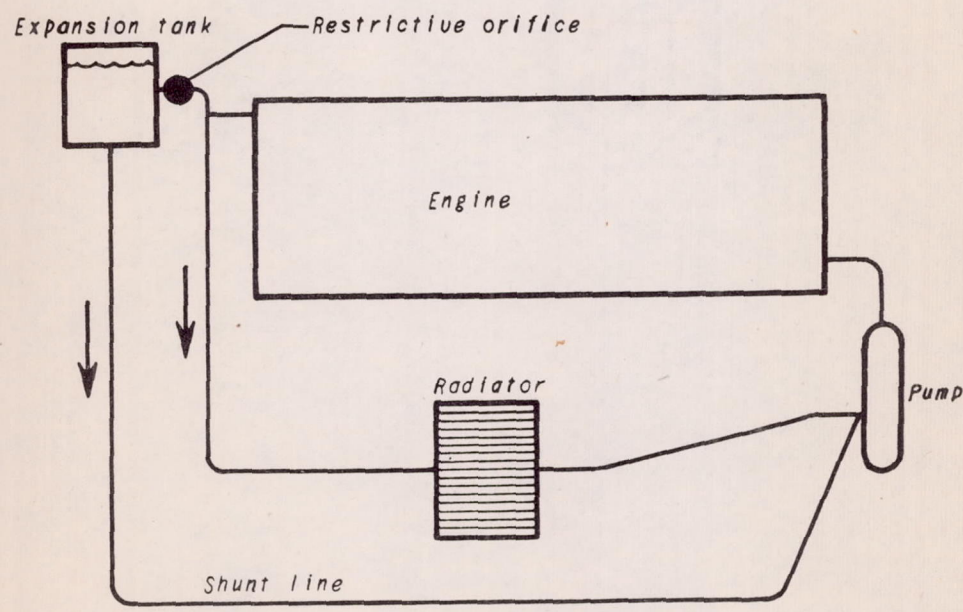
(b)   Shunt-type circuit.

Figure 1. - Schematic diagram of basic types of cooling
      circuit.

Front
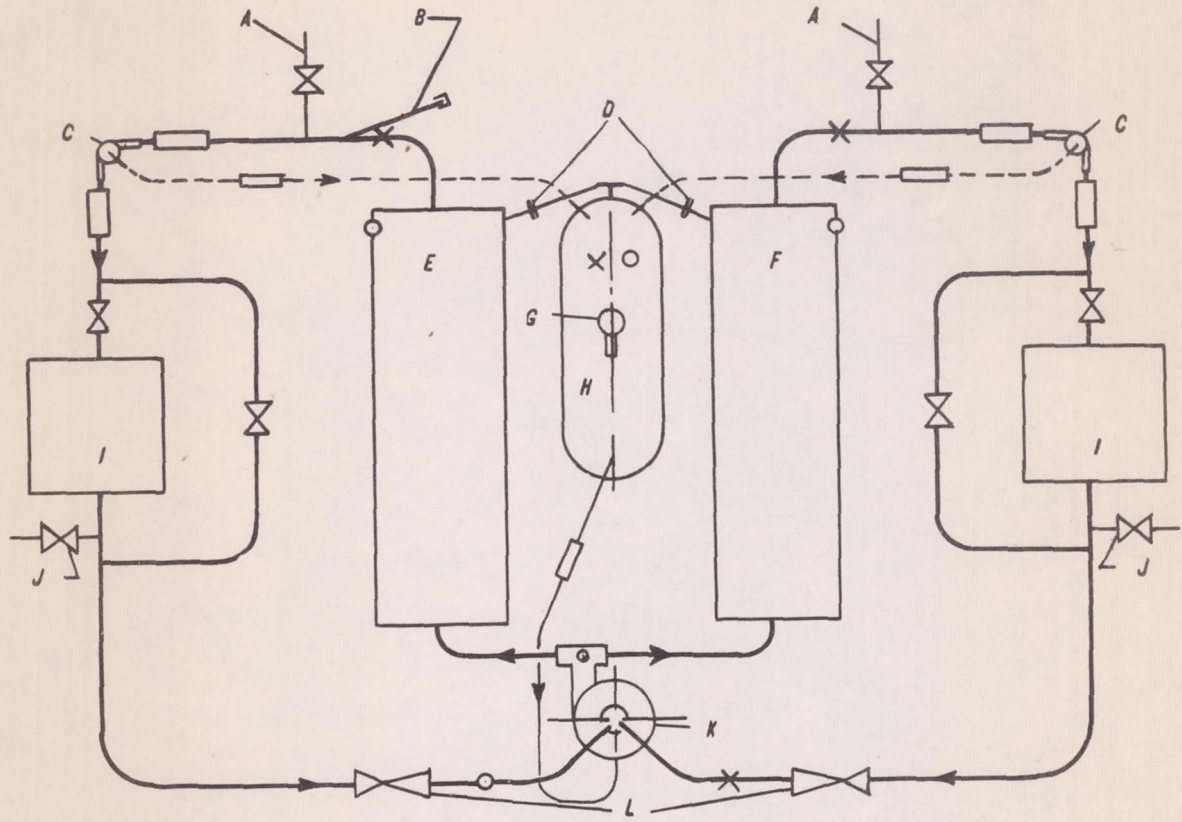
Cooling system
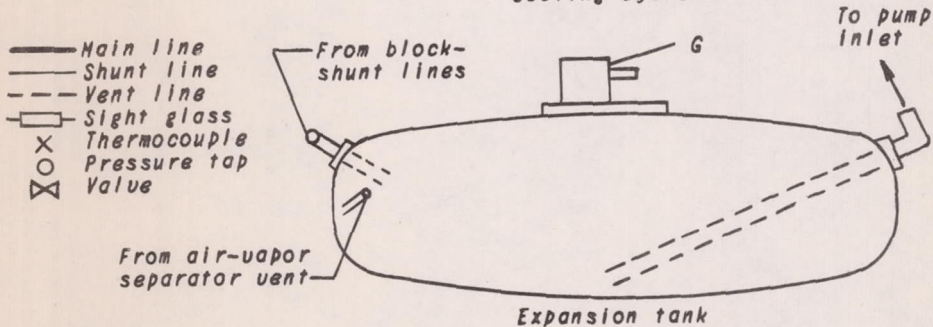
Main line
Shunt line
Vent line
Sight glass
Thermocouple
Pressure tap
Valve

From block-
shunt lines

To pump
inlet

From air-vapor
separator vent

Expansion tank

A Compressed air   E Left bank   I Cooler
B Filler line   F Right bank   J Bleed valve
C Air-vapor separators G Pressure-relief valves K Pump
D Orifices   H Expansion tank   L Venturis

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Figure 2. - Diagrammatic sketch of cooling system A showing
air-vapor separators installed in main coolant lines and
a cross-sectional view of the expansion tank.

Figure 3. - Diagrammatic sketch of cooling system A showing air-vapor separators installed in the shunt lines and a cross-sectional view of baffled expansion tank.

Vapor outlet

Inlet

NACA
C-5980
8-1-44

Outlet

0   1
INCH

(a)   In the main line.

Vapor outlet

NACA
C-5981
8-1-44

0   1
INCH

Outlet
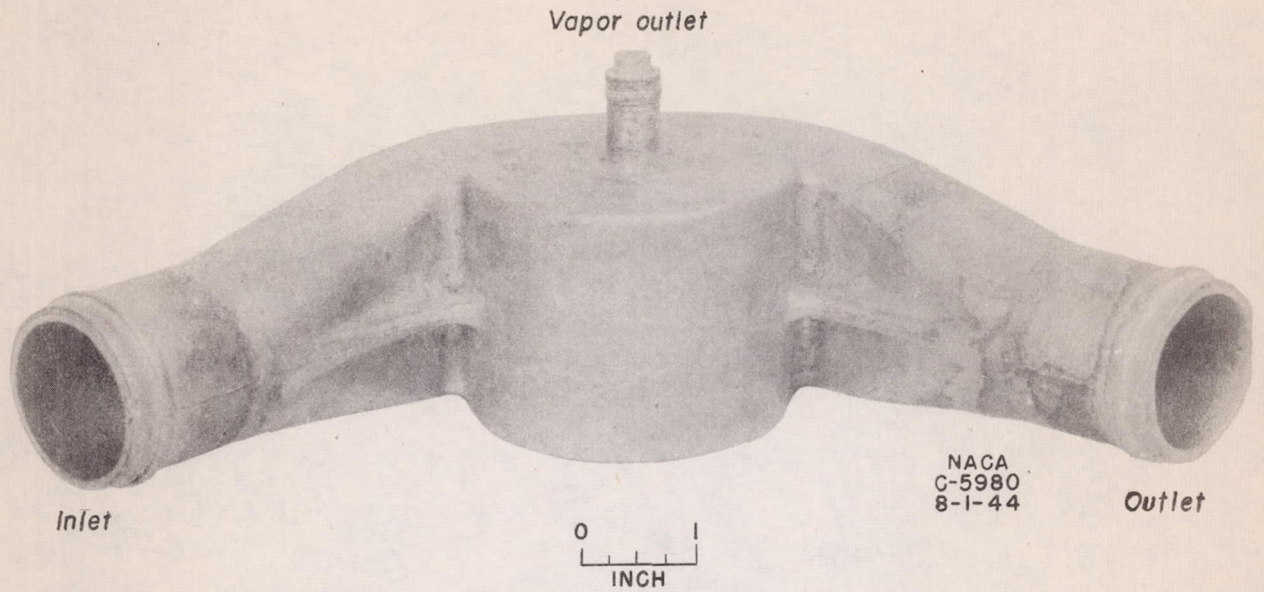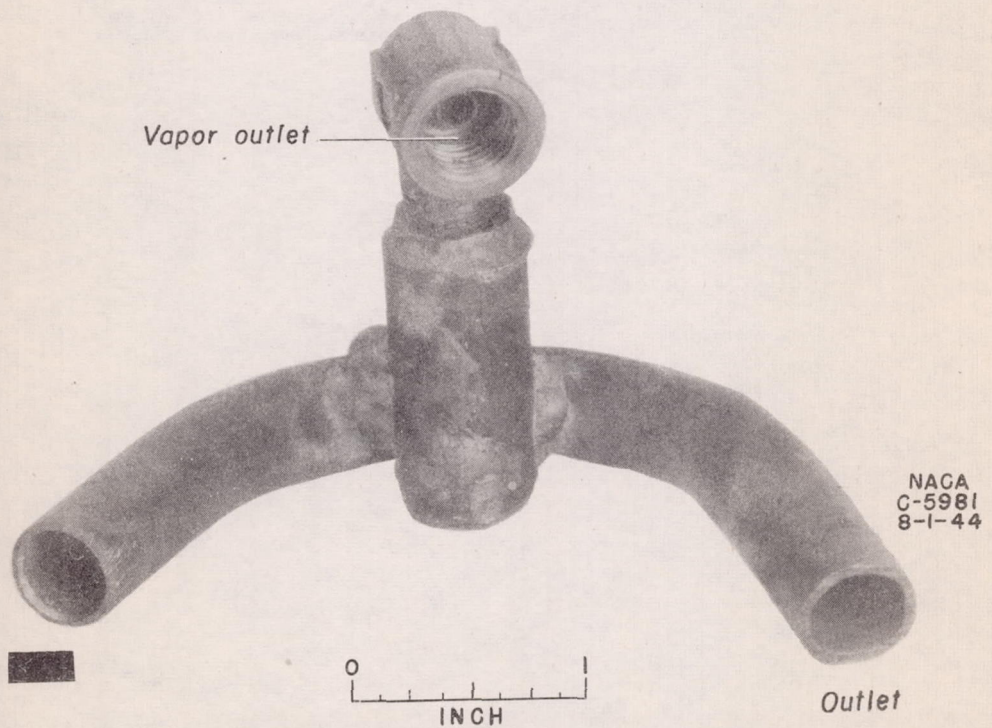
(b)   In the shunt line.

Figure 4. - Vapor separators used in cooling system A.
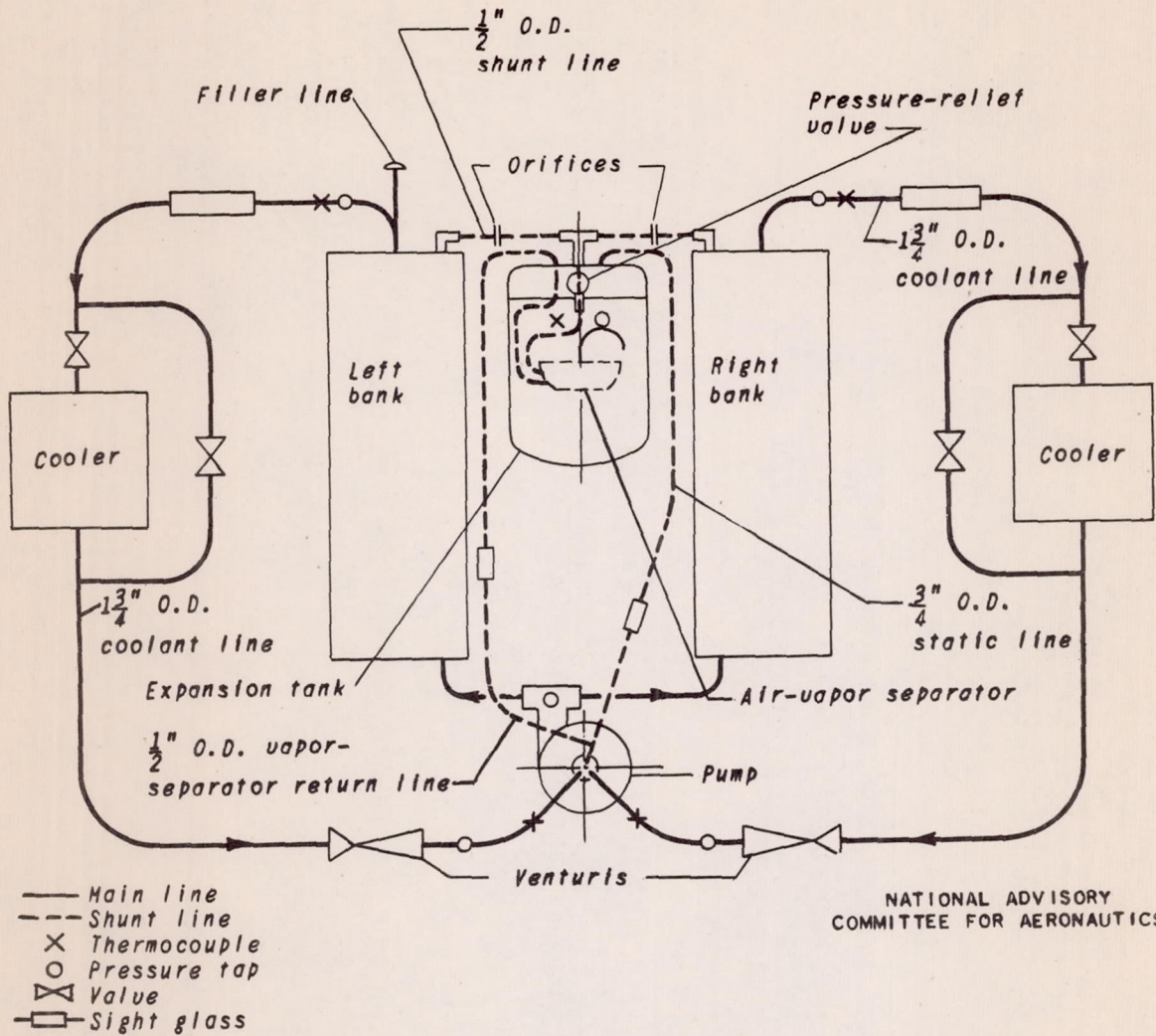
Figure 5. – Schematic diagram of cooling system B using Linde
   expansion tank.

½" O.D. shunt line from blocks

½" O.D. vapor separator return line

Pressure-relief valve

Screen

Air-vapor separator vent

Thermocouple boss

¾" O.D. static line to pump inlet

Pressure gage, compressed air, and blow-off connection (normally connected to filler line)

Air-vapor separator
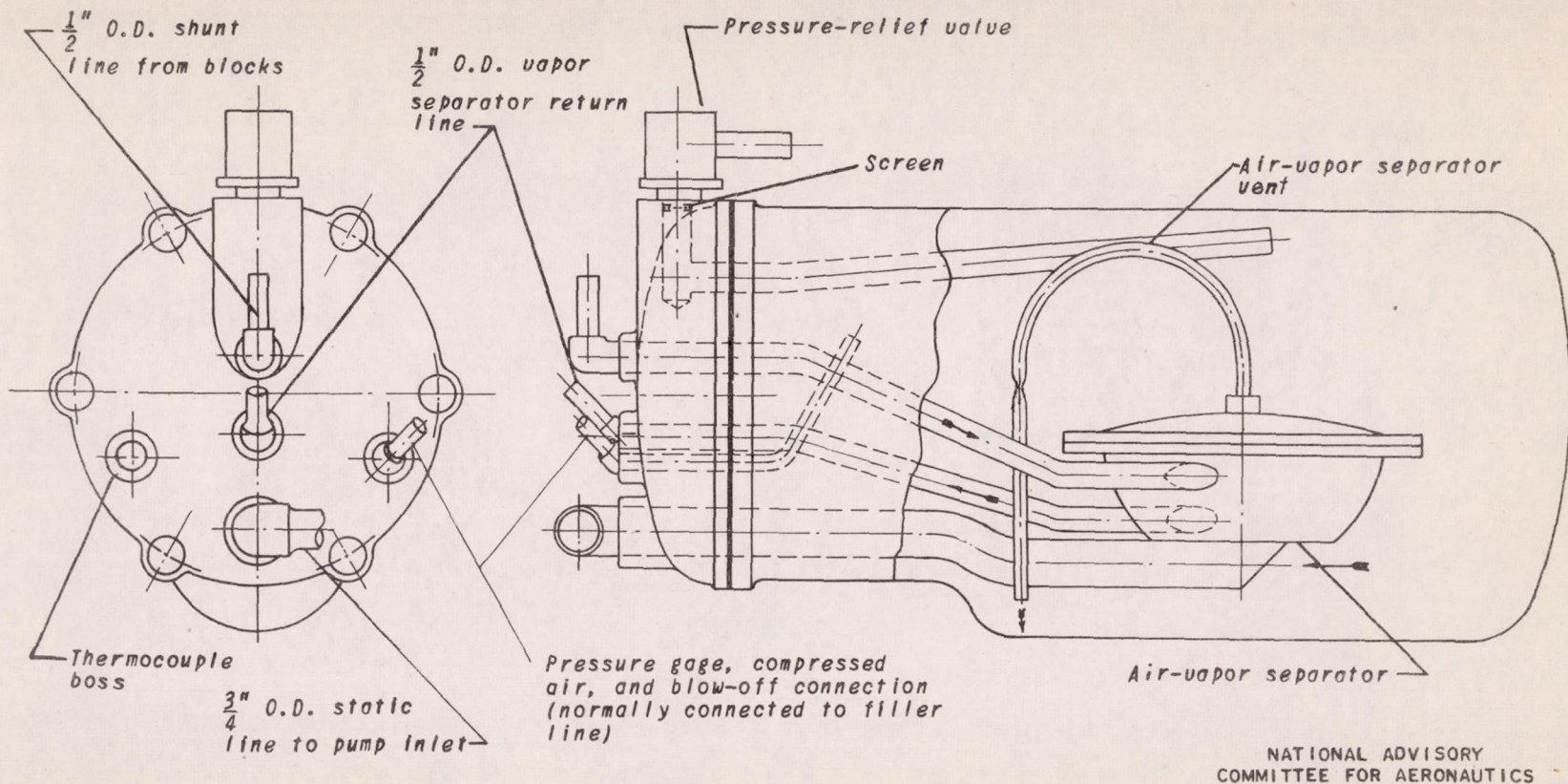
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Figure 6. - Details of Linde expansion tank used in cooling system B.

Fig. 6

NACA
C-5979
8-1-44

0 ___ I
INCH
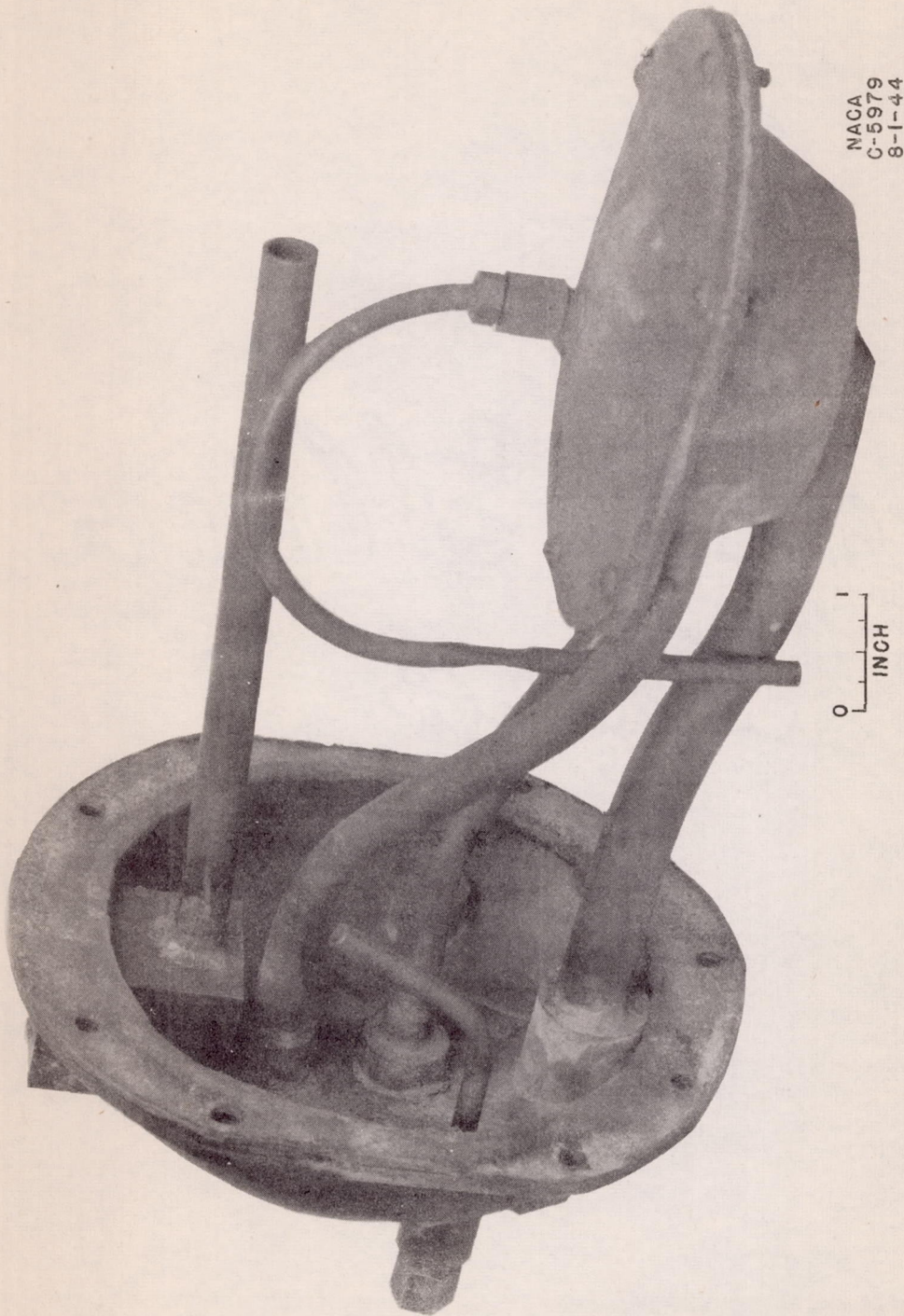
Figure 7. - Internal details of Linde expansion tank used in cooling system B.

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o   Main-line separators
+   Shunt-line separators
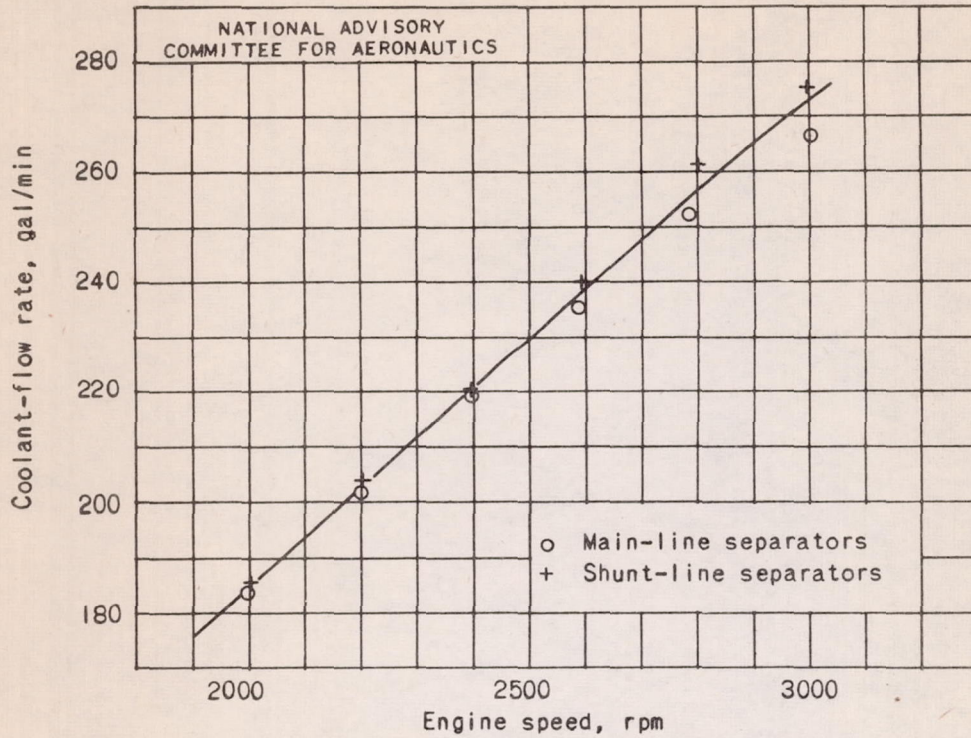
(a) Cooling system A Pressure proximity, 7 pounds per square inch.

Shunt-line orifice (in.)	Pressure proximity (lb/sq in.)
◇ 1/8	5
× 1/8	6
□ 3/16	6
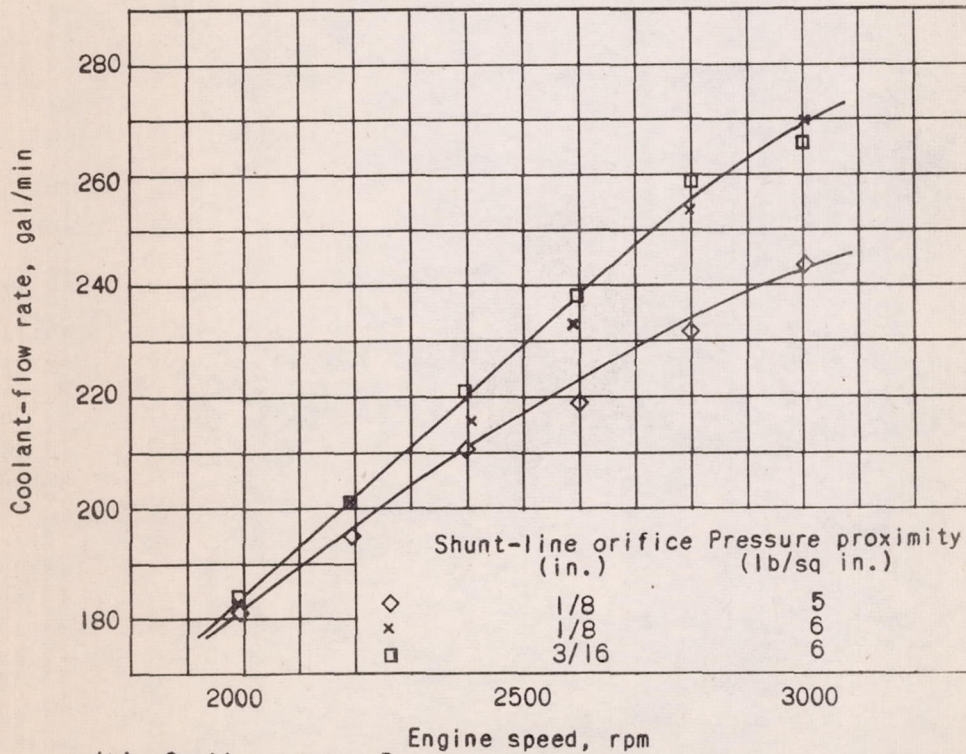
(b) Cooling system B.

Figure 8. - Variation of coolant-flow rate with engine speed. Coolant-out
temperature, 250° F; coolant, 30 percent AN-E-2 ethylene glycol plus 70
percent water; pump speed, 1.234 times engine speed.
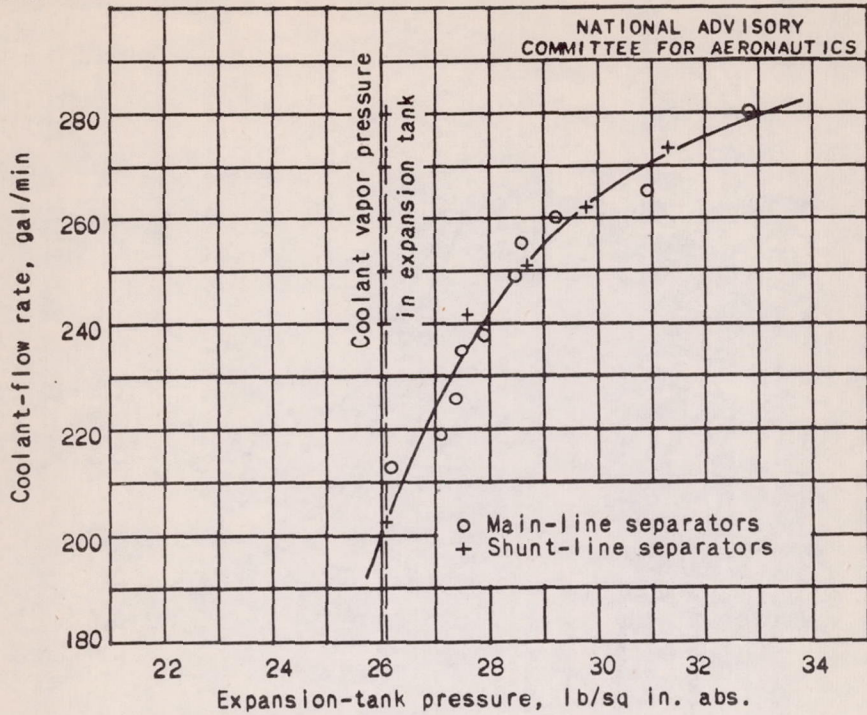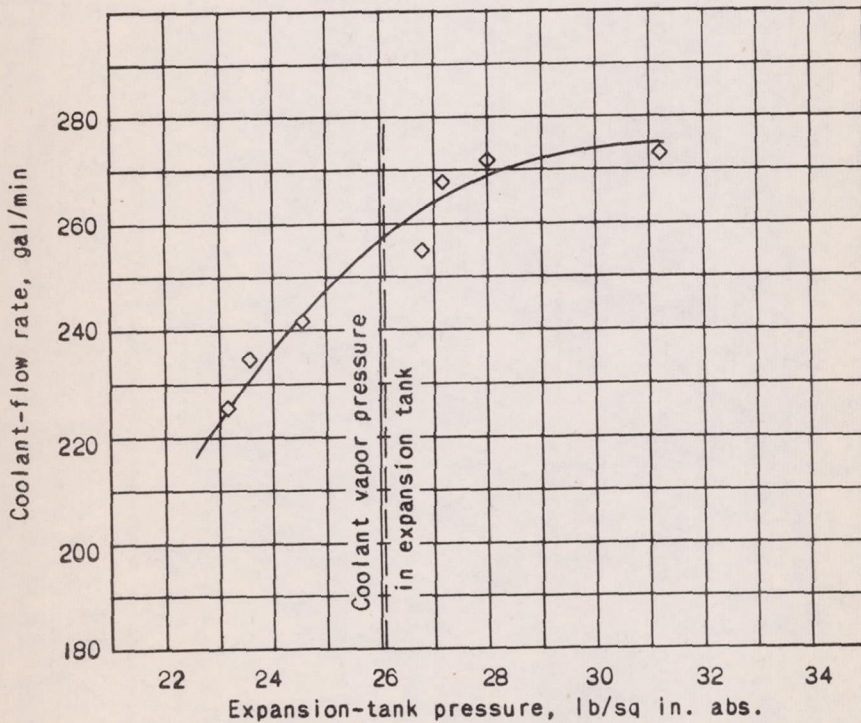
(a) Cooling system A.

(b) Cooling system B.

Figure 9. - Variation of coolant-flow rate with expansion-tank
    pressure. Engine speed, 3000 rpm; coolant-out temperature,
    250° F; coolant, 30 percent AN-E-2 ethylene glycol plus 70
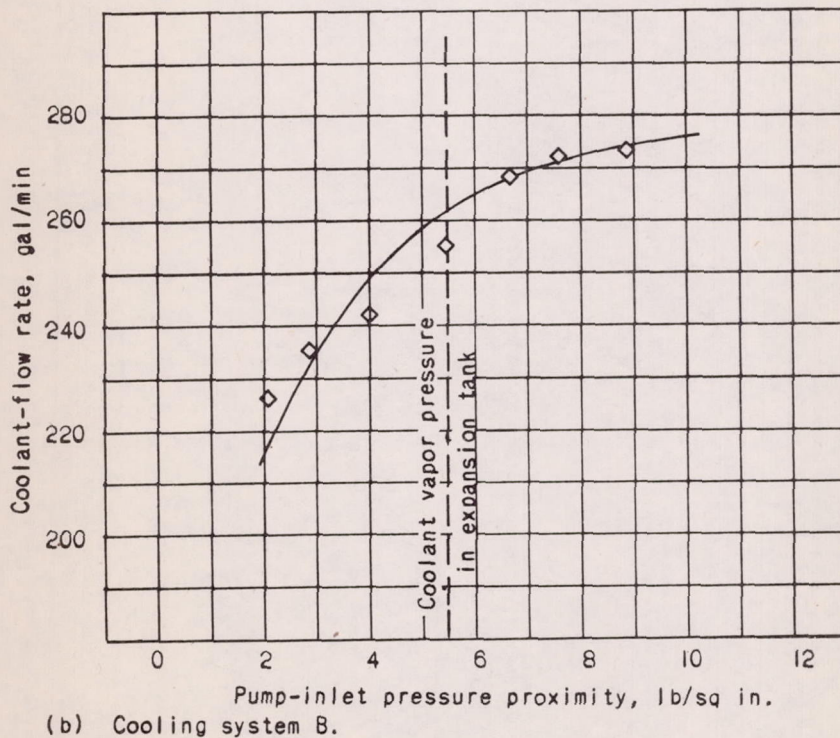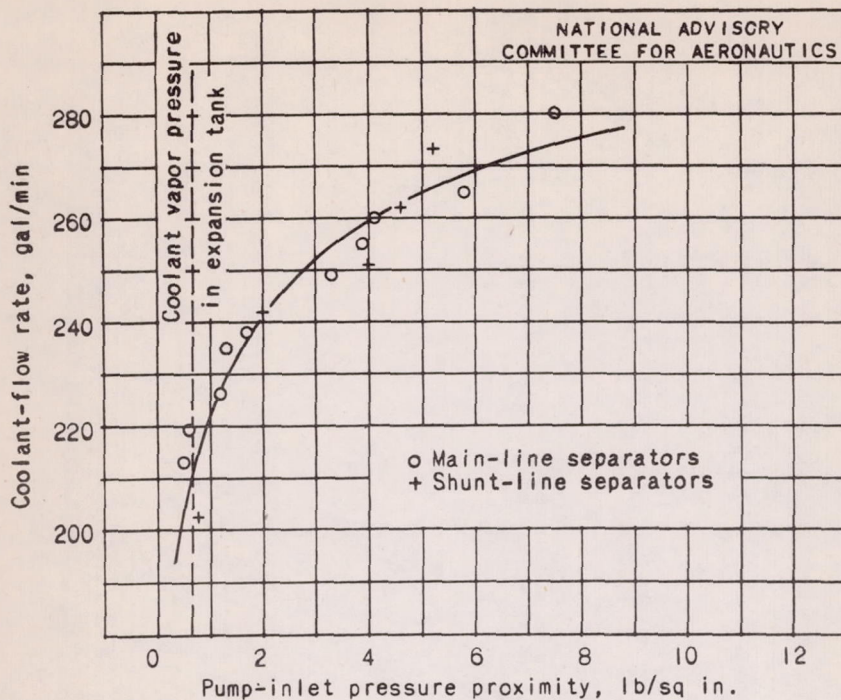   percent water; 1/8-inch shunt-line orifices.
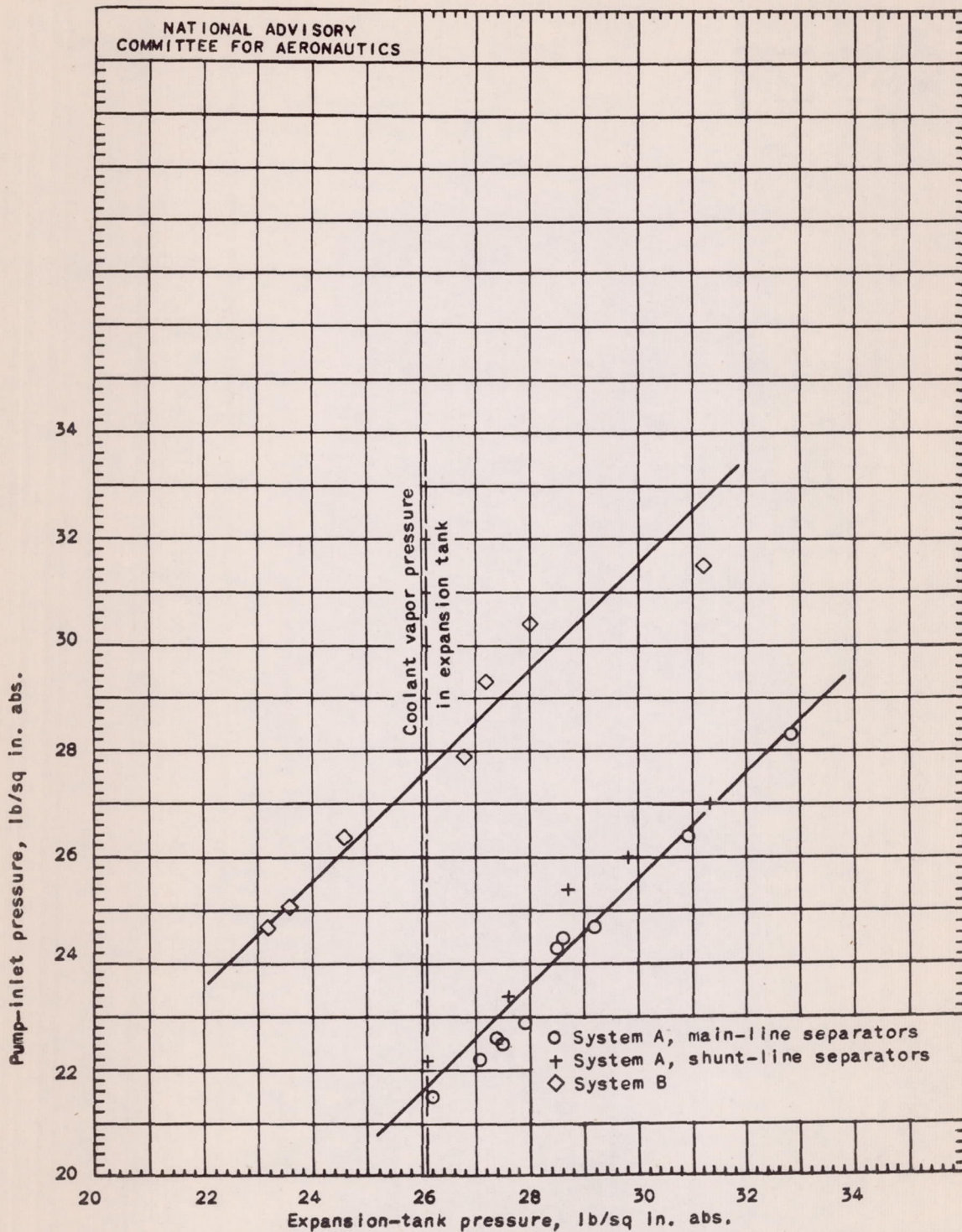
Figure 10. – Variation of pump-inlet pressure with expansion-tank pressure for both cooling systems A and B. Engine speed, 3000 rpm; coolant-out temperature, 250° F; coolant, 30 percent AN-E-2 ethylene glycol plus 70 percent water; 1/8-inch shunt-line orifices.

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Coolant-flow rate, gal/min

280

260

240

220

200

Coolant vapor pressure in expansion tank

○ Main-line separators
+ Shunt-line separators

20   22   24   26   28   30   32   34
Pump-inlet pressure, lb/sq in. abs.

(a) Cooling system A.

Coolant-flow rate, gal/min

280

260

240

220

200

Coolant vapor pressure in expansion tank

20   22   24   26   28   30   32   34
Pump-inlet pressure, lb/sq in. abs.

(b) Cooling system B.

Figure II. - Variation of coolant-flow rate with pump-inlet pressure.
Engine speed, 3000 rpm; coolant-out temperature, 250° F; coolant,
30 percent AN-E-2 ethylene glycol plus 70 percent water; 1/8-inch
shunt-line orifices.

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o Main-line separators
+ Shunt-line separators

(a) Cooling system A.

(b) Cooling system B.

Figure 12. - Variation of coolant-flow rate with pump-inlet
    pressure proximity. Engine speed, 3000 rpm; coolant-out
    temperature, 250° F; coolant, 30 percent AN-E-2 ethylene
    glycol plus 70 percent water; 1/8-inch shunt-line orifices.
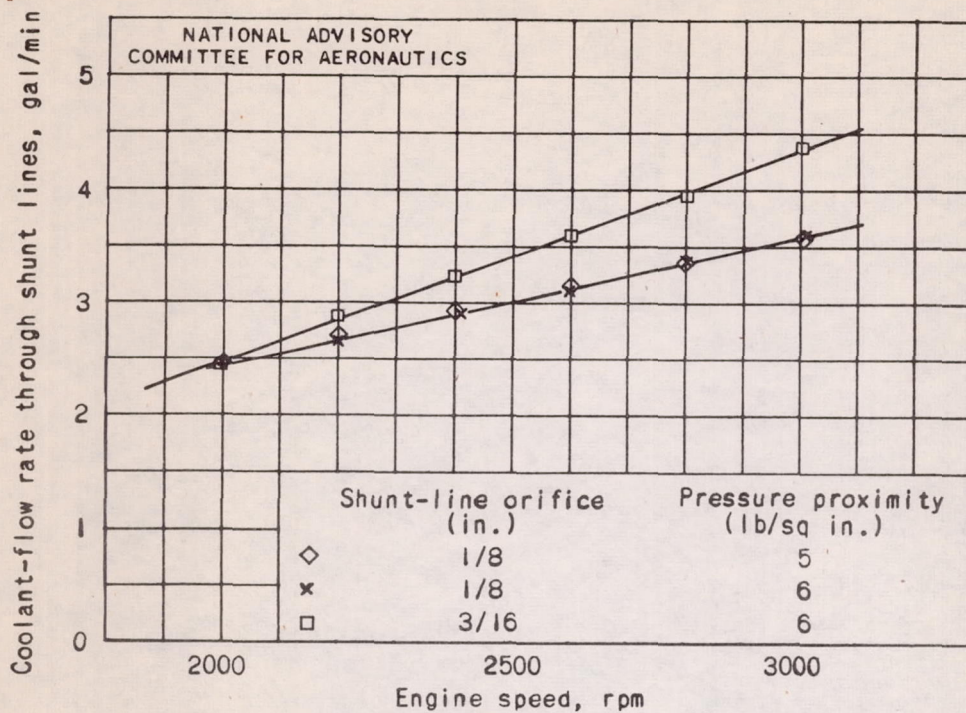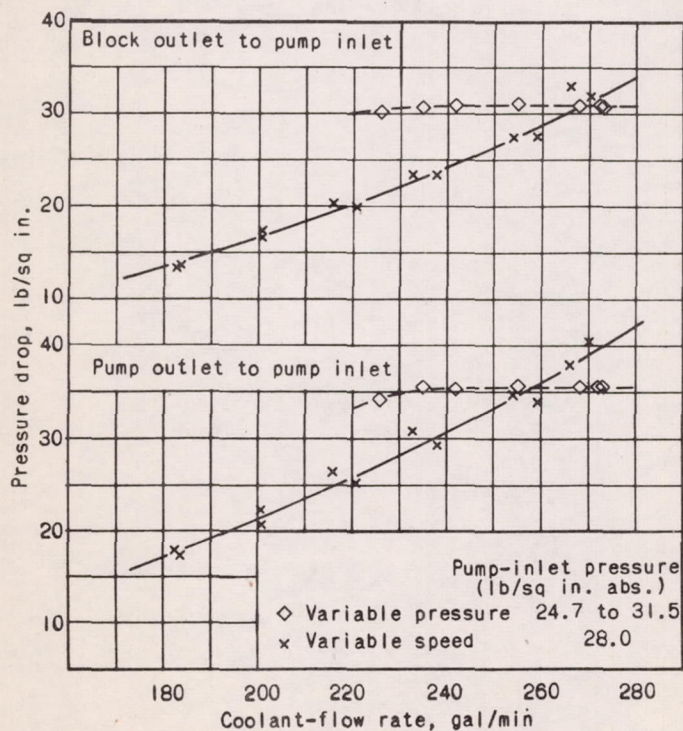
(a) Variation with engine speed.
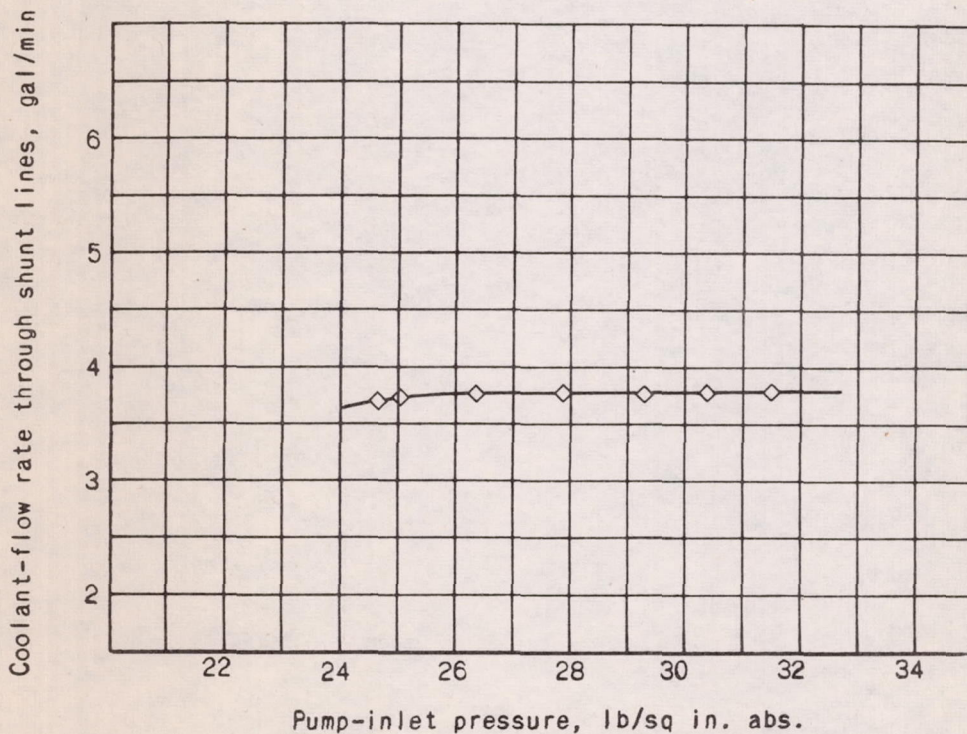
(b) Variation with pump-inlet pressure. Engine speed, 3000 rpm.

Figure 13. - Variation of coolant-flow rate through shunt lines. Coolant-
out temperature, 250° F; coolant, 30 percent AN-E-2 ethylene glycol
plus 70 percent water; cooling system B; pump speed, 1.234 times engine
speed.
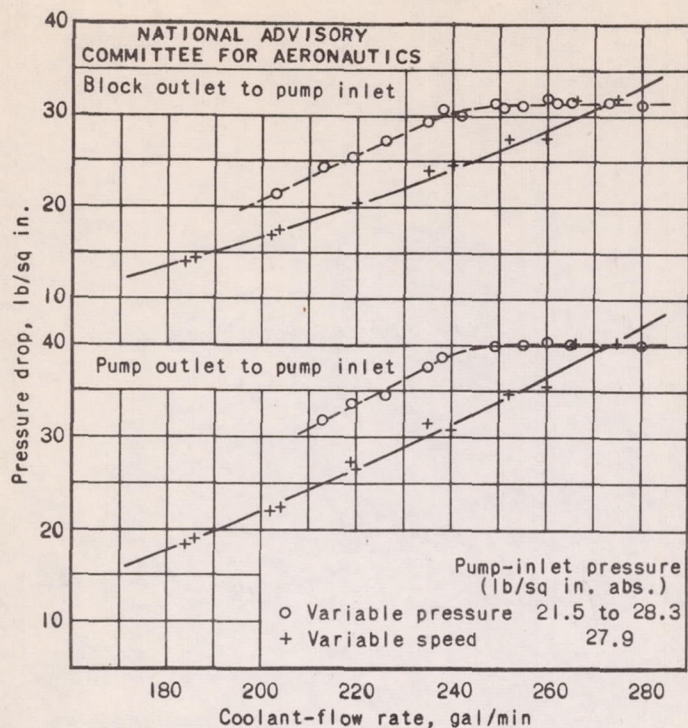
(a) Cooling system A.

(b) Cooling system B.

Figure 14. - Variation of pressure drop through entire
system and from block outlet to pump inlet with total
coolant-flow rate. Coolant-out temperature, 250° F;
coolant, 30 percent AN-E-2 ethylene glycol plus 70
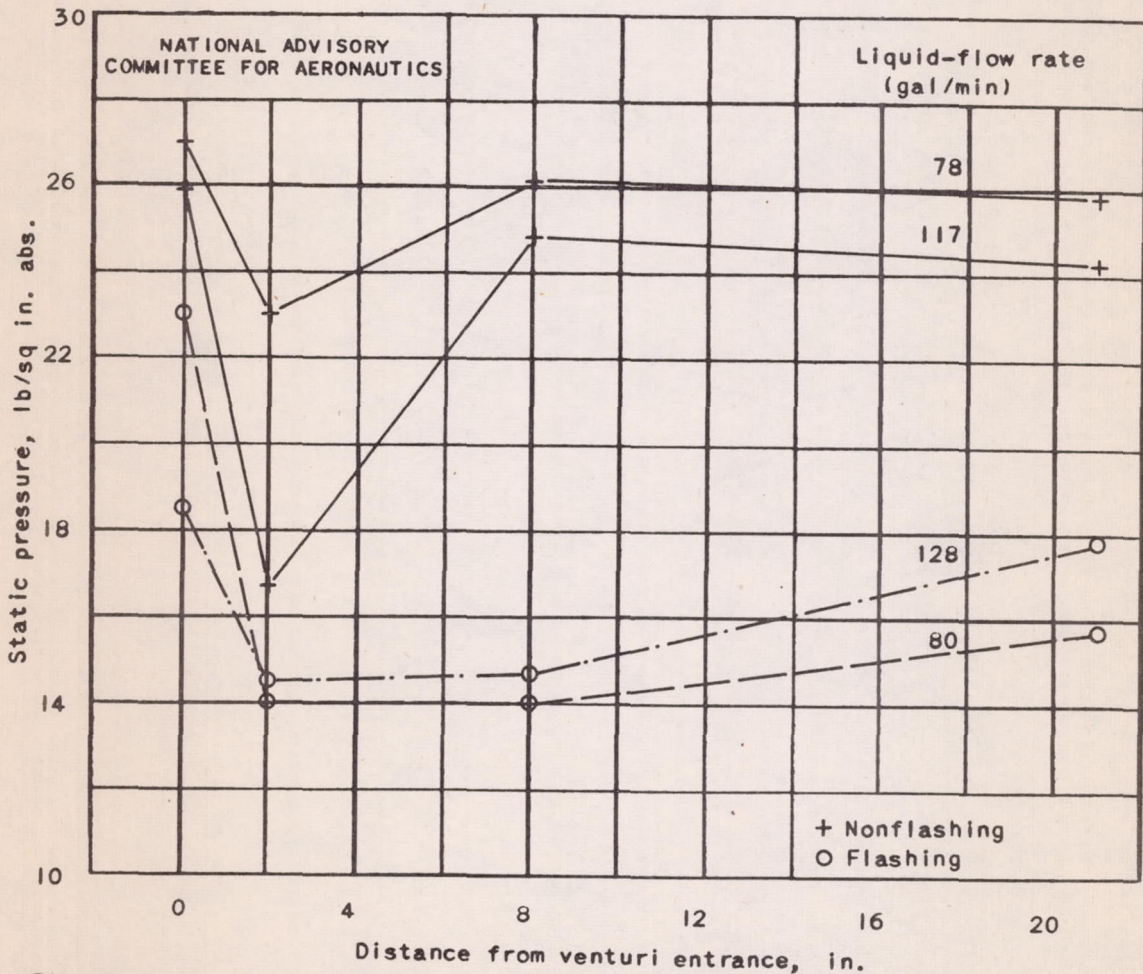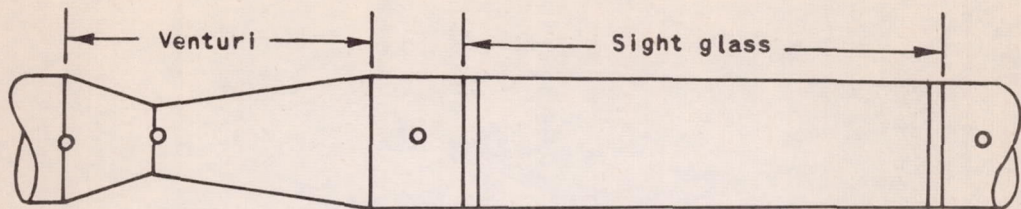percent water.

Figure 15. - Pressure survey through venturi and sight glass for flashing and nonflashing runs. Fluid, 30 percent AN-E-2 ethylene glycol plus 70 percent water; fluid temperature, 219° F.
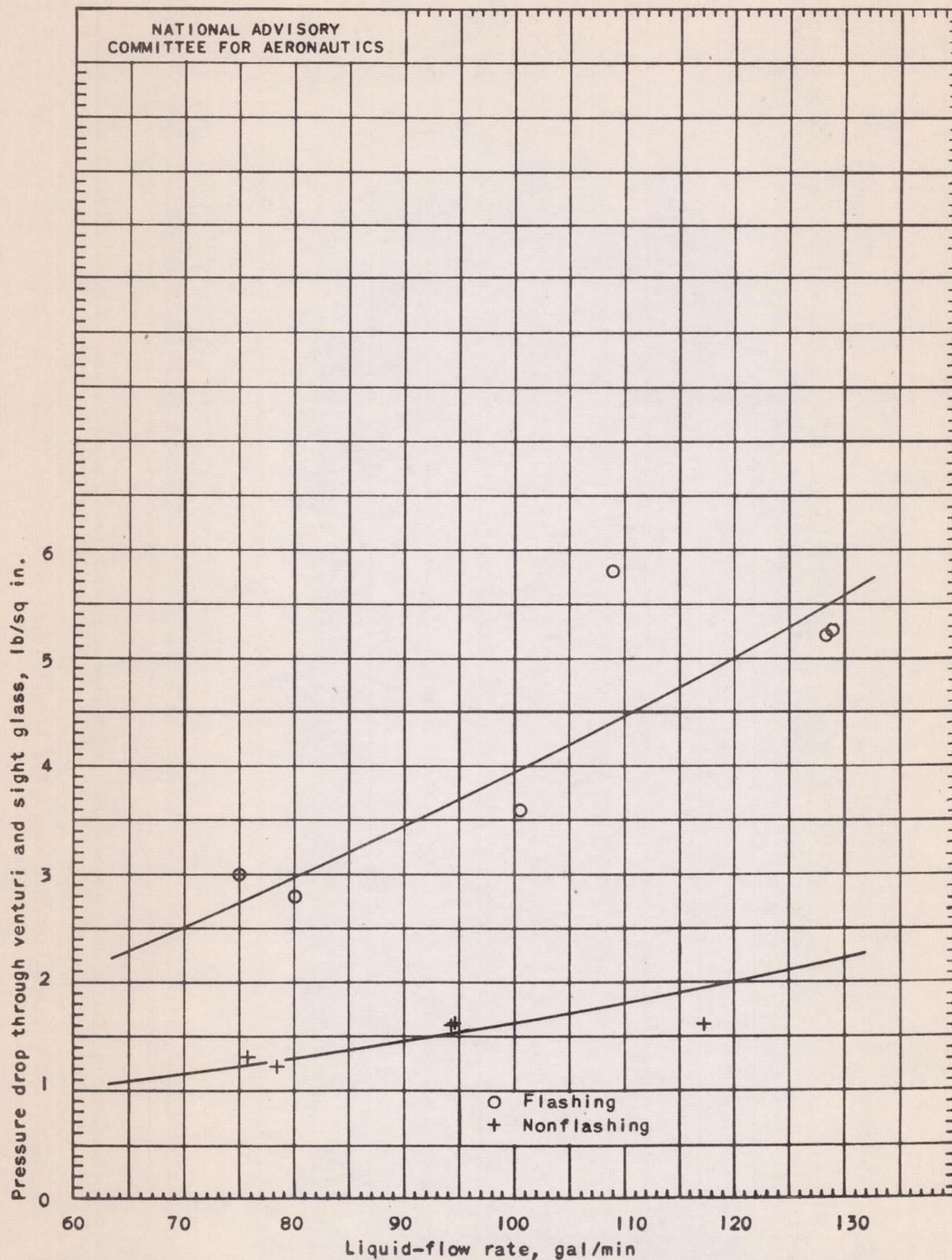
Figure 16. — Variation of over-all pressure drop through venturi and sight
glass with fluid-flow rate. Fluid, 30 percent AN-E-2 ethylene glycol plus
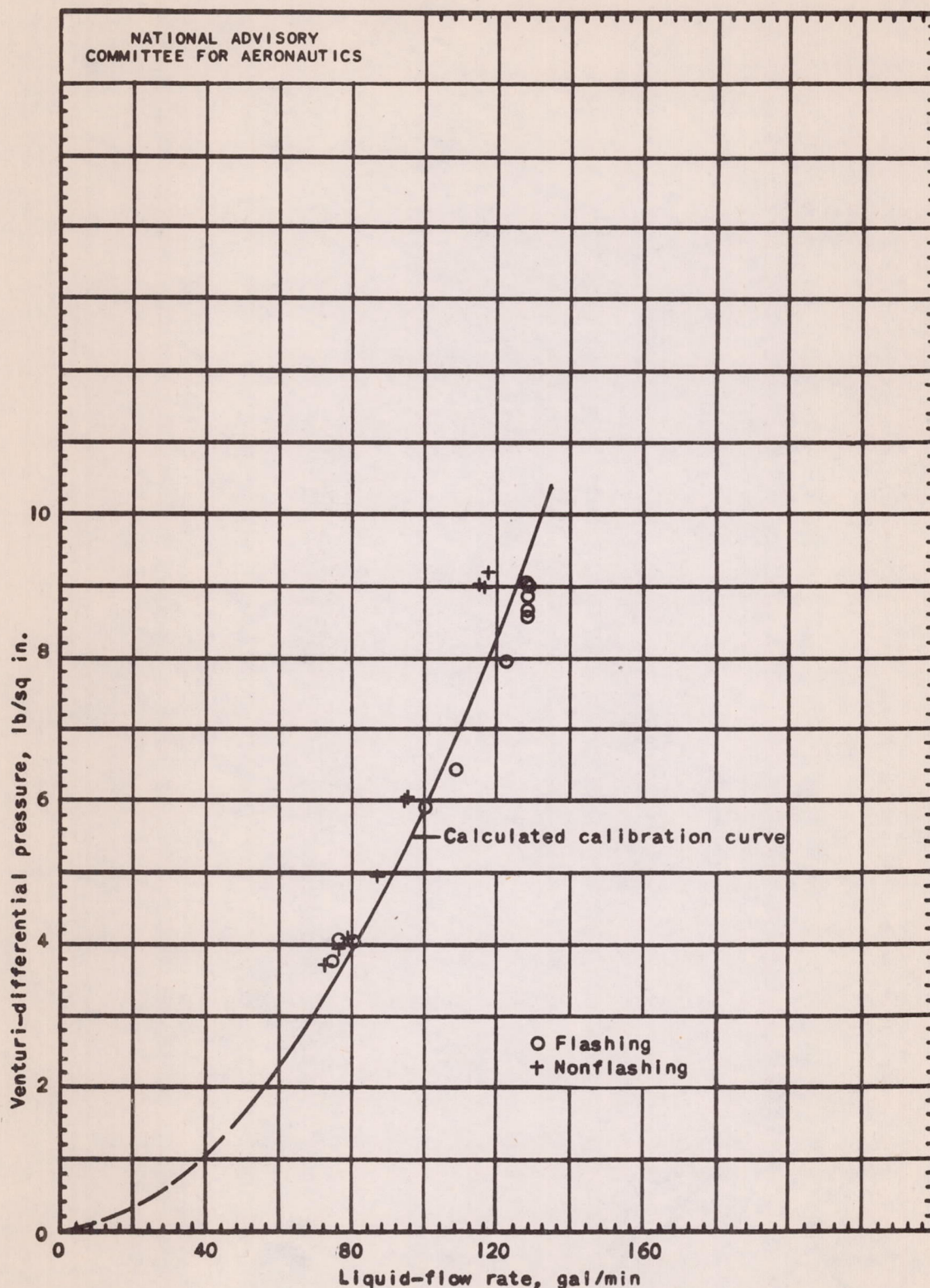70 percent water; fluid temperature, 219° F.

Figure 17. — Venturi calibration curve for flashing and nonflashing runs.
Fluid, 30 percent AN-E-2 ethylene glycol plus 70 percent water; fluid
temperature, 219° F.