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RESEARCH MEMORANDUM

INTERNAL FILM COOLING EXPERIMENTS IN 4-INCH DUCT WITH GAS TEMPERATURES TO 2000° F

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NATIONAL ADVISORY COMMITTEE
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INTERNAL-FILM-COOLING EXPERIMENTS IN 4-INCH DUCT WITH GAS TEMPERATURES TO 2000° F

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SUMMARY

The cooling effectiveness of water films on the inner surface of a well-insulated, 4-inch-diameter duct was investigated with air flowing through the duct at temperatures from 600° to 2000° F and Reynolds numbers from 400,000 to 1,400,000. Water flows from 1 to 6 percent of the gas flows were used and a few additional experiments were conducted with ethylene glycol as a coolant.

Liquid coolant films were established and maintained along the duct wall in co-current flow with the hot gas. During vaporization of the liquid, the coolants kept the duct wall below the boiling temperature of the liquid but little additional cooling of the duct was obtained from the coolant vapor.

Film cooling with an inert liquid having a high heat of vaporization was an effective cooling method. For example, at a gas temperature of 1600° F and a Reynolds number of 470,000, film cooling with a water flow of 465 pounds per hour (3.3 percent of the gas flow) kept the duct wall below 240° F for a distance of 5 diameters.

The relation between liquid-cooled length and coolant flow for given gas-stream conditions was nonlinear; the effectiveness of a given amount of coolant introduced at a single axial position of the duct decreased with increased coolant flow. The decrease in cooling effectiveness as coolant flow increased may make it desirable to limit the amount of coolant introduced at any single axial position and to introduce it at several axial positions. The relation between liquid-cooled length and coolant flow was unaffected by the use of various methods of coolant injection; the trend of the curves was unaffected by the use of coolants having different physical properties. The duct area cooled by a given coolant flow increased with a decrease in roughness of the duct surface and with elimination of projections from the duct wall.

Although a complete correlation of the variables was not obtained, the data were generalized by the use of heat-transfer analogy in a form that facilitates the estimation of the quantity of coolant required to film-cool a duct for a desired length when the temperature and the flow rate of the hot gas are known. 1356

INTRODUCTION

In combustion processes involving very high heat-flux densities as, for example, in rocket engines, and where conventional cooling methods are unfeasible, effective cooling can be accomplished by means of "sweat" cooling or internal-film cooling. In these methods, a layer of coolant is maintained on the inner surfaces of the combustor and acts as an effective shield between the walls and the hot gases. Investigations of sweat cooling with gases flowing through porous walls are reported in references 1 to 5, in which theoretical and experimental data indicate the amount of gaseous coolant required to maintain a porous wall at a given temperature for known hot-gas-stream conditions.

The use of certain liquids as internal coolants is desirable because their large heat-absorption capacity (including the heat of vaporization) can be utilized. The disadvantage of forcing liquids through a porous surface in a sweating process is the danger of vapor lock of the coolant within the porous wall and subsequent excessive wall temperatures (reference 6). This disadvantage of liquid coolants can be overcome by introducing the coolant at discrete positions within the combustion chamber and the nozzle in such a manner that an effective coolant film is maintained on the inner surfaces between the points of coolant injection (film cooling).

Exploratory work on film cooling of rocket engines by the Jet Propulsion Laboratory of the California Institute of Technology and by the NACA Lewis laboratory showed that small quantities of coolant are effective in cooling the engine. Data on film cooling with water in a 1-inch tube with a hydrogen-oxygen flame are given in reference 7, but film-coolant flows greater than the gas flows were used therein for most of the experiments and only about one-half of the water vaporized in the tube. Information is needed on film cooling for applications where the coolant flow is small compared with the gas flow and where the quantity of coolant required can be predicted from the hot-gas-stream conditions.

An investigation of film cooling is therefore being conducted at the NACA Lewis laboratory and the results of preliminary experiments, made during 1949, are reported herein. These initial experiments were conducted to: (a) determine if a film of liquid coolant could be established and maintained on the inner walls of a straight duct in concurrent flow with a hot gas; (b) determine the effects of coolant flow, gas flow, and gas temperatures on film-cooled length; and (c) obtain a correlation that would allow prediction of the amount of coolant required to cool a duct for a given length with known hot-gas-stream conditions. The experiments were conducted in a 4-inch-diameter duct with gas temperatures from 600° to 2000° F and gas-stream Reynolds numbers from 400,000 to 1,400,000. For most of the experiments, the coolant was water injected at a single axial position on the duct at flow rates from 1 to 6 percent of the gas flow; the remaining runs were made with ethylene glycol as a coolant. The length of duct cooled by the liquid was determined by means of thermocouples on the duct surface.

The gas temperatures investigated were limited to the range of 600° to 2000° F for simplification of apparatus and instrumentation. Although these temperatures are too low for practical use in rocket-cooling applications, they are adequate for the general purpose of establishing the relation between the coolant flow and the gas-stream conditions. Film-cooling investigations at low temperatures should therefore provide a foundation for additional investigations at higher temperatures, in which more complex apparatus and instrumentation are required.

APPARATUS

Flow System

The flow system (fig. 1) consists essentially of three parts: (1) source of hot gas at a uniform temperature, (2) test section, and (3) exhaust system. The source of hot gas consisted of the air supply at a pressure of 40 pounds per square inch gage, a surge chamber, a jet-engine combustor, a mixing section with orifice- and target-mixing baffles, and a calming chamber 12 inches in diameter. The test section was a 4-inch-diameter, 1/8-inch-wall duct with a 10-diameter approach section, a coolant injector, and a 12-diameter film-cooled test section. In the exhaust section was a series of water sprays to quench the hot gases; this section was connected to the laboratory exhaust system. All hot portions of the system, which were made of Inconel, were well insulated to minimize heat losses. The assembly was supported at the surge chamber and by roller stands located upstream and downstream of the test section. An expansion bellows was installed downstream of the exhaust-quenching sprays.

Coolant-Injection System

The coolant-injection system consisted of supply reservoir, filters, positive-displacement pump, adjustable pressure regulators (which controlled flow), rotameters, and coolant injectors.

The injectors for introducing the internal film coolants onto the inner surface of the test section are shown in figure 2. The jet-type injector (fig. 2(a)) was a brass ring having 60 holes 0.013 inch in diameter equally spaced around the circumference at an angle of 45° to the axis of the ring. A stainless-steel splash ring was fitted into the brass ring and a housing, which provided a supply annulus for the coolant, fitted over the brass ring. The coolant, supplied to the annulus about the brass ring, flowed through the holes in the ring and formed small jets, which hit the splash ring. This splash ring directed the flow of the coolant downstream and against the surface of the duct. The spacing between the splash plate and the injector surface did not control the coolant flow; the function of the splash plate was only to direct the flow. The splash ring blocked the gas-flow area approximately 5 percent. Two modifications of the jet-type injector shown in figure 2(a) were used for some of the experiments. The first modification consisted in enlarging the metering holes to decrease by a factor of 2 the velocity of the coolant jets impinging on the splash plate. The second modification was to alter the flow pattern of the coolant emerging from the injector by increasing the splash-plate clearance by a factor of 2.

The porous-surface-type injector (fig. 2(b)) consisted of a copper ring with 90 slots milled into the inner surface about the circumference. The slots were approximately 1.5 inches long and the thickness of the walls separating them was approximately 0.020 inch. Holes, 0.013 inch in diameter, were drilled through the ring into each of the slots. A liner of a porous wire cloth fitted inside the copper ring over the slots. A housing, which provided a supply annulus for the coolant, fitted over the copper ring. The coolant supplied to the annulus about the ring flowed through the small holes into the slots and then through the porous cloth liner onto the surface. The small holes metered the flow into each slot, which resulted in a uniform distribution of the

flow about the circumference. The slots spread the coolant flow over a large area, thereby reducing the flow velocity. Because the liner was very porous, it did not restrict the flow, but provided a surface onto which the coolant flowed at low velocity. The air
1356 flow over the surface of the injector carried the coolant downstream along the inner surface of the duct.

Film-Cooled Test Sections

Two film-cooled test sections and three types of duct surface were used during the investigation. Both test sections were 1/8-inch-wall Inconel ducts, 4 inches in diameter and 46 inches long. The first duct-surface type, hereinafter designated rough-surface duct, was a rolled section with a longitudinal weld; although the inside surface was finished after welding, some surface roughness and waviness remained. Other surface disturbances in this section were from instrumentation and consisted of: (a) a slot and a projecting pressure probe at each of three positions, 6, 23, and 40 inches downstream of the injector; and (b) four static-pressure taps spaced 90° apart at approximately the same duct-length positions. The second type, hereinafter designated smooth-surface duct, was a seamless duct honed to obtain a smooth inside surface; there were no slots, projections, holes, or other surface disturbances in this section. The third type of duct surface used was the rough-surface duct further roughened from products formed after the use of ethylene glycol as a coolant. This inadvertent roughening is referred to as "glycol-roughened" surface for convenience. All ducts were insulated with 3 inches of high-temperature Fiberglas insulation.

Coolants

Water was used as the coolant for most of the experiments. A short series of runs was made with ethylene glycol as the coolant to determine the effect of a change in physical properties of the coolant on film-cooling effectiveness. The ethylene glycol used contained 3-percent rust inhibitor and 3-percent water. The pressures in the test section for the range of conditions investigated varied between 20 and 30 pounds per square inch absolute, which resulted in a variation in the boiling temperatures of the coolants of approximately 20° F. The average boiling temperatures were 240° F for water and 360° F for the ethylene glycol. A comparison of the physical properties of the liquid coolants at their respective boiling temperatures is given in the following table:

	Water (240°F)	Ethylene glycol (360°F)
Viscosity, centipoises	0.24	0.57
Density, gram/cc	.95	.99
Specific heat, Btu/(lb)(°F)	1.00	.78
Thermal conductivity,		
Btu/(hr)(sq ft)(°F/ft)	.43	.26
Heat of vaporization, Btu/lb	952	344 (387°F)
Surface tension (at 77° F), dynes/cm . . .	72	54

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Instrumentation

Flow rate. - Air flow was measured within ±250 pounds per hour (1 to 2 percent of the air flows) by means of an orifice conforming to standard A.S.T.M. specifications and a differential water manometer. The coolant flow was measured by means of rotameters within ±5 pounds per hour (1 to 4 percent of flows used).

Pressure. - Pressures were measured by means of static- and total-pressure probes in conjunction with mercury manometers. These probes were installed in the calming section, in the approach section before the injector, and at three locations in the rough-surface duct as shown by figure 1. The pressure measurements were useful in determining approximate gas densities and velocities in the test section but were not accurate enough for use in determining friction losses.

Gas temperature. - Gas temperatures were measured by means of chromel-alumel thermocouples and a self-balancing potentiometer. Rakes containing twenty thermocouples were located in the calming section and a rake with four thermocouples was placed in the approach section 3 inches upstream of the coolant injector. The estimated accuracy of gas-temperature measurements, exclusive of errors induced by uncertainties in recovery factor, varied from ±12° F at 600° F to ±25° F at 2000° F. These values were determined from a consideration of instrument accuracy, wire calibration, radiation and conduction heat losses, and fluctuations of the gas-stream temperature during runs. The recovery factor for the thermocouples located in the approach section (where the Mach numbers were from 0.6 to 0.7) was of the order of 0.9; this recovery factor was obtained from a comparison of the thermocouple readings in the approach section with those in the calming section (where Mach numbers were below 0.1). Because the recovery factor was high, the total temperatures measured in the approach section were used directly and were not corrected by use of the recovery factor. Wall temperatures in the film-cooled duct were measured by means of chromel-alumel thermocouples and a recording potentiometer. The thermocouples, which were welded to the outer surface of the duct, were spaced along the length of the duct at eight positions around the circumference (fig. 3). The thermocouple leads were kept along the wall surface for approximately 1 inch to minimize heat loss through the leads. Wall temperatures were measured by these thermocouples within $\pm 10^{\circ}$ F.

PROCEDURE

Operating conditions. - The operating procedure consisted in setting the coolant flow, air flow, and air temperature at desired values, allowing sufficient time for conditions to stabilize, and recording the data. The operating conditions covered the following ranges:

Gas temperature, °F	600-2000
Gas flow, lb/hr	12,500-25,600
Coolant flow, lb/hr	117-800

These conditions give the following values:

Reynolds number	400,000-1,400,000
Mach number	0.6-0.7
Prandtl number	0.63-0.68
Gas mass velocity, lb/(hr)(sq ft)	143,000-293,000
Gas velocity, ft/sec	900-1800
Coolant flow rate/gas flow rate, percent . . .	1-6

Liquid-cooled length. - Cooling effectiveness was determined by plotting the wall temperatures of the test section as a function of distance from the point of coolant injection. A typical plot from a run with the smooth-surface duct is shown in figure 3. The duct-wall temperature remains below the boiling temperature of the coolant (water) for approximately 19 inches downstream of the point of coolant injection, rises to the boiling temperature, and then rises rapidly as the coolant vapor is dispersed into the main gas stream. The wall temperature then approaches a value near the stagnation temperature of the gas coolant-vapor mixture. The liquid-cooled length L (fig. 3) is determined from an average of the distances for which the wall remained below the boiling temperature of the coolant at the eight circumferential positions. For the smooth-surface duct, the agreement of the liquid-cooled lengths for the eight circumferential positions was within about 10 to 15 percent; at very low coolant flows greater variation occurred. For the rough-surface duct, the roughness and the waviness of the inner surface and the instrumentation holes and projections caused disturbances of the liquid film and resulted in more scatter of the wall temperatures at the various circumferential positions. The liquid-cooled length for most of the positions agreed within about 20 percent; however, a few of the positions differed from the average by as much as 50 percent. 1356

Effect of gravity on liquid film. - Annular flow of a liquid on the inner walls of a duct with gas flowing through the duct at high velocity can be established (reference 8). The gas velocities in the experiments reported herein were sufficiently high to maintain the annular coolant flow established by the coolant injector. The effect of gravity would be to decrease the proportionate film thickness in the upper portions of the horizontal duct for a decrease in gas flow. The effect of gravity, however, although noticeable for some runs, was of insufficient magnitude to affect the trends observed in film cooling.

Reproducibility of results. - At various times throughout the investigation, runs were made at the same conditions to determine the reproducibility of results. For ten runs at constant conditions, the variation in liquid-cooled length was 8 percent; for seven runs at another set of conditions, the variation in liquid-cooled length was 6 percent; for three runs at still another set of conditions, the variation in liquid-cooled length was 14 percent. These results indicate reproducibility within 14 percent and that most results are reproducible within a smaller variation.

RESULTS AND DISCUSSION

Effect of Gas Temperature and Mass Velocity on Liquid-Cooled Length

Liquid-cooled length is shown in figure 4(a) as a function of gas temperature with gas mass velocity as a parameter and coolant flow constant at 464 pounds per hour. The data, obtained with the rough-surface duct, show a large decrease in liquid-cooled length with increase in temperature in the range of 800° to about 1400° F; from 1400° to 2000° F the rate of decrease in liquid-cooled length with increased temperature is comparatively small. These curves follow the expected trend from infinite liquid-cooled length for no temperature difference between gas stream and coolant to an asymptotic approach to zero liquid-cooled length as the temperature difference between gas stream and coolant increases to large values. Increased gas mass velocity results in decreased liquid-cooled length. This effect is more clearly shown in figure 4(b), where the same data are plotted as a function of gas mass velocity with temperature as a parameter. The cooled length decreases as the mass velocity increases and the slope of the curves increases with decreasing gas temperature.

Effect of Coolant Flow on Liquid-Cooled Length

Liquid-cooled length in the rough-surface duct is plotted in figure 5(a) as a function of coolant flow at two gas temperatures and two gas flows at each temperature. For each of the different gas-stream conditions, the relation between the liquid-cooled length and the coolant flow is nonlinear over the range of coolant flows investigated. The slope of each of the curves is greatest at low coolant flows and decreases with increased coolant flow. The rate of change of slope decreases with coolant flow; at the high flows the slope is essentially constant. Similar data for the smooth-surface duct (fig. 5(b)) show the same characteristic trends as those obtained with the rough-surface duct (fig. 5(a)).

A comparison of the data obtained with the smooth- and rough-surface ducts at the same hot-gas-stream conditions (gas temperature, 1400° F and mass velocity, approximately 163,000 lb/(hr)(sq ft)) shows that the liquid-cooled length is greater for the smooth-surface duct than for the rough-surface duct and that the difference between the curves increases with increasing coolant flow. For example, the liquid-cooled length for the smooth-surface duct is about 20 percent greater than that for the rough-surface ducts at a coolant flow of 400 pounds per hour.

The data of figure 5 show that film cooling with an inert liquid having a high heat of vaporization is an effective cooling method. For example, at a gas temperature of 1600° F and a Reynolds number of 470,000, water at the rate of 465 pounds per hour (or in the proportion of 3.3 percent of the gas flow) kept the wall temperature of the smooth section below the boiling temperature of the water (liquid-cooled length) for 5 diameters (fig. 5(b)). In contrast, cooling the same duct length with a water jacket on the outer surface of the duct and assuming an allowable temperature rise of 150° F would require about 1440 pounds per hour of cooling water. If in film cooling all the water is assumed vaporized from the surface, this particular comparison indicates that the effective heat-transfer coefficient for the film-cooling process is about 2.5 times greater than that obtained with conventional cooling.

Further Investigations on Effect of Coolant Flow on Liquid-Cooled Length

Additional experiments were made to determine if a change in coolant properties or coolant-injection method would affect the trends shown by the curves of figure 5. The results of experiments with ethylene glycol as a coolant with the glycol-roughened surface are shown in figure 6. Also shown are data with water in the same duct but at different gas-stream conditions. The data show the same characteristic trends as those of figure 5, even though the glycol had a greater viscosity and smaller heat of vaporization than water. The data of figure 6 do not permit a comparison of the cooling effectiveness of glycol and water because of the different gas-stream conditions, but the data do show the effect of surface roughness on cooling effectiveness. The increased surface roughness caused by the glycol deposits lowered the cooling effectiveness, as can be seen by comparing the data for water from figure 6 with comparable conditions of figure 5(a).

The results of experiments made to determine if coolant-injection methods affected the trends of the curves in figure 5 are shown by figure 7. Liquid-cooled length is shown in figure 7(a) as a function of coolant flow with the rough-surface section; the standard jet injector and two modified jet injectors (fig. 2(a)) were used. Although the modified jet injectors decreased the emerging velocity of the coolant jets by a factor of 2 and altered the flow pattern from the injector, the trends of the curves are the same and no measurable differences in cooling effectiveness were obtained. A comparison of results obtained with the standard jet-type injector and the porous-surface injector, which had a very low coolant emergence velocity, is shown in figure 7(b). The results, obtained with the glycol-roughened surface, show that the trends of the curves are the same as those of figure 5 and no measurable difference in cooling effectiveness was obtained. These comparisons of coolant-injection methods showed the cooling effectiveness to be independent of the velocity at which the coolant was uniformly admitted about the circumference of the duct for the conditions investigated.

Discussion of Results

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The nonlinearity of the plots of liquid-cooled length as a function of coolant flow may be caused by the combined effects of increased loss of liquid coolant from the wall and an increase of exposed area of the liquid per unit duct area with increased coolant flow. Both these factors would be influenced by disturbances of the liquid surface caused, for example, by the interaction effect of gas and liquid, by a change from laminar to turbulent flow of the liquid, and by eruptions of the liquid surface by rapid boiling; the surface disturbances may have increased appreciably with coolant flow.

An example of a possible interaction effect between gas and liquid is the formation of waves as reported in reference 8; however, the ratios of liquid flow to gas flow investigated were not in the same range as those of the experiments reported herein. Boiling may cause considerable disturbance of the liquid film and the magnitude of the disturbance may be affected by the thickness of the film and therefore by coolant flow.

The experiments showed that the decrease in cooling effectiveness per unit coolant flow was not overcome by changes in method of coolant injection, by the use of a very smooth duct, or by a coolant of different physical properties. For maximum coolant economy, it therefore appears desirable to limit the quantity of coolant introduced at any single axial position along the duct and to introduce the coolant at several axial positions.

SYMBOLS

The following symbols are used in this report:

A_D cross-sectional area of film-cooled duct, (sq ft)

A_L liquid-film-cooled surface area, (sq ft)

$c_{p,c}$ average specific heat at constant pressure of liquid coolant, Btu/(lb)/($^{\circ}$F)

$c_{p,g}$ specific heat at constant pressure of gas entering film-cooled duct, Btu/(lb)/($^{\circ}$F)

D diameter of film-cooled duct, (ft)

G mass velocity of hot gas, (lb/hr/sq ft)

H_v heat of vaporization of coolant, Btu/(lb)

ΔH_v change in enthalpy of liquid from entering condition through vaporization, Btu/(lb) 1356

h heat-transfer coefficient, Btu/(hr)(sq ft)($^{\circ}$F)

k thermal conductivity, Btu/(hr)(sq ft)($^{\circ}$F/ft)

L liquid-film-cooled length, (ft)

Pr Prandtl number, $\frac{c_p \mu}{k}$

Re Reynolds number, $\frac{DG}{\mu}$

St Stanton number, $\frac{h}{c_p G} = \frac{h}{\rho V c_p}$

T_g stagnation temperature of gas stream entering film-cooled duct, ($^{\circ}$F)

T_i temperature of coolant before injection, ($^{\circ}$F)

T_s temperature of coolant liquid surface (saturation temperature), ($^{\circ}$F)

W_c coolant flow, (lb/hr)

W_g gas flow, (lb/hr)

μ viscosity, (lb)/(hr)(ft)

PRELIMINARY CORRELATION OF DATA

A complete correlation of the data was not obtained because of the nonlinearity of liquid-cooled length with coolant flow and the effect of surface roughness on cooling effectiveness. Preliminary correlations of the data were made, however, by analogy with conventional heat-transfer relations.

The liquid-cooled length was used to calculate a heat-transfer coefficient between the hot gas stream and the liquid-cooling film as follows:

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$$h = \frac{W_c \left[c_{p,c}(T_s - T_i) + H_v\right]}{(T_g - T_s)(\pi DL)}$$

A correlation was made in which this heat-transfer coefficient was used and in which all the coolant was assumed vaporized by heat transfer from the hot gas stream to the liquid film. The relative velocity between the gas stream and the coolant film was assumed large enough so that the heat transfer between them could be treated in the same manner as heat transfer between hot gases flowing in a duct and the duct walls. The correlation is made in the form of

$$St = f(Re,Pr)$$

The Stanton number for each run was calculated from the heat-transfer coefficient, the specific heat of air at the stagnation temperature of the gas stream, and the mass velocity of the gas stream. Prandtl number and Reynolds number for each run were calculated by using the physical properties of air at the stagnation temperature of the hot gas stream.

Figure 8 shows a log-log plot of Stanton number divided by Prandtl number raised to the -0.6 power against Reynolds number for results obtained from the rough-surface duct. (Data from the smooth-surface duct did not include a sufficient range of gas-stream conditions to make a similar correlation.) The Prandtl number exponent of -0.6 was assumed from conventional heat-transfer correlations because Prandtl number did not vary appreciably during the runs. The results shown include data obtained at gas-stream temperatures from 600° to 2000° F, mass velocities of the gas stream from 146,000 to 293,000 pounds per hour per square foot, and Reynolds numbers from 400,000 to 1,400,000. The three lines represent correlations for three coolant flows for which sufficient data were obtained at various gas-stream conditions. The coolant flows, 369, 464, and 654 pounds per hour, are each in the region where the rate of change of slope of the curves of figure 5(a) is small. Included on figure 8 is the line for conventional heat transfer to duct walls. The values of heat-transfer coefficients for the film cooling are considerably greater.

The data of figure 8 for a coolant flow rate of 464 pounds per hour were used to determine the Reynolds number function, because more data were obtained for this flow than for the other flows. The slope of the line shows that

$$\frac{St}{Pr^{-0.6}} \sim Re^{0.07}$$

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The data at coolant flows of 369 and 654 pounds per hour indicate approximately the same function of Reynolds number. For data obtained at a coolant flow of 464 pounds per hour, 45 out of 48 runs fell within ±8 percent of the correlation line. The other three points were 11, 14, and 17 percent from the lines. Reproducibility of the same order of magnitude as that obtained for liquid-cooled lengths at constant operating conditions is thus indicated.

The function of Reynolds number obtained for heat transfer to the coolant film different from that obtained for conventional heat transfer to duct walls $Re^{-0.2}$ is not surprising because of the differences in physical situation between the two cases. The coolant film has a velocity in the direction of flow of the hot gas and vaporization of the coolant film is occurring. Mixing of the coolant film with the hot gas stream, which is unaccounted for in the assumptions made in the heat-transfer analogy, may also affect the function of Reynolds number.

In order to assemble all the data, the heat-transfer relation

$$St = f(Re,Pr)$$

was solved in terms of coolant flow by substituting the relation for heat-transfer coefficient, previously defined, in Stanton number. When the function of Reynolds number determined from figure 8 is used, the relation becomes

$$W_c = \left(W_g\right)\left[\frac{c_{p,g}(T_g - T_s)}{\Delta H_v}\right]\left(\frac{A_L}{A_D}\right)Re^{0.07}\ Pr^{-0.6}$$

A plot of this function is shown in figure 9 with water in the rough- and smooth-surface ducts and also water and glycol in the rough section with glycol deposits on the surface. By means of these plots, the coolant flow required to maintain a liquid-cooled surface over a given area for known gas-stream temperature and flow conditions can be estimated. In figure 9(a) (water in rough-surface duct), the spread of the data is approximately the same as that obtained for reproducibility of liquid-cooled length at constant experimental conditions and the same as the spread of the data in the Stanton number correlations. From these data, little or no change of slope of the relation is apparent at coolant flows above about 350 pounds per hour.

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At coolant flows above about 400 pounds per hour (fig. 9(b), water in smooth section), the data show less scatter than those of figure 9(a); but the data are fewer and cover a more limited range of gas-stream conditions. At coolant flows below about 400 pounds per hour, the cooling parameter (ordinate of fig. 9(b)) decreases with increased gas temperature.

The heat-transfer analogy used for the preliminary correlations of the data (fig. 8) correlates the data at the higher coolant flows where little change of slope of the liquid-cooled length - coolant-flow relation occurs. At the low coolant flows, however, the data are not satisfactorily correlated as can be seen by the trends shown in figure 9(b). This effect of temperature at the low coolant flows is not so apparent in figure 9(a), possibly because of effects of the rough surface of the section and the limited amount of data taken.

Data obtained with ethylene glycol and water as coolants in the glycol-roughened duct are shown in figure 9(c). Although the data are limited, the results show that two coolants with considerably different physical properties can be correlated, which indicates that the effectiveness of inert coolants in film cooling is proportional to their heat-absorption capacity as liquids and their heat of vaporization. On this basis, the cooled length obtainable with water is about twice that obtainable with glycol. It is probable that coolant properties will affect film cooling at coolant-to gas-flow ratios at which maintaining annular coolant flow along the duct walls is critical.

The effect of surface roughness, previously mentioned, can also be seen by comparing figures 9(a), 9(b), and 9(c). A decrease in surface roughness and elimination of surface projections increase the liquid-cooled length at a given coolant flow for the same temperature and hot-gas flow. A projection from the wall disrupts the coolant film, promotes liquid loss from the wall, and in some cases may cause the formation and the growth of a hot region in the metal wall downstream of the projection.

SUMMARY OF RESULTS

An investigation was conducted on the cooling effectiveness of liquid films on the inner surface of a well-insulated, 4-inch-diameter duct with air flowing through the duct at temperatures from 600° to 2000° F and Reynolds numbers from 400,000 to 1,400,000. Water was investigated as the coolant at flow rates from 100 to 800 pounds per hour (1 to 6 percent of the gas flow); cooling effectiveness was determined by means of wall-temperature measurements. A few additional experiments were conducted with ethylene glycol as a coolant. The results of the experiments are summarized as follows:

1. The effectiveness of film cooling is illustrated by the following data: At a gas temperature of 1600° F and a Reynolds number of 470,000, film cooling with a water flow of 465 pounds per hour (3.3 percent of the gas flow) kept the duct-wall temperature below the boiling point of the water (240° F) for a distance of 5 diameters.

2. The relation between liquid-cooled length and coolant flow for given gas-stream conditions was nonlinear; the effectiveness of a given amount of coolant introduced at a single axial position of the duct decreased with an increase in coolant flow.

3. The relation between liquid-cooled length and coolant flow was unaffected by the use of various methods of coolant injection and the trend of the curves was unaffected by the use of coolants having different physical properties.

4. The duct area cooled by a given coolant flow increased with a decrease in roughness of the duct surface and from elimination of projections from the duct wall.

5. Although a complete correlation of the variables (coolant flow, gas flow, and gas temperature) was not obtained, the data for both water and glycol as coolants were generalized over part of the coolant-flow range investigated by the use of heat-transfer analogy in a form that facilitated the estimation of the quantity of coolant required to film-cool a duct for a desired length when the temperature and the flow of the hot gas were known.

CONCLUSIONS

1356 The following conclusions are drawn from the experiments reported herein where the inner surface of the walls of a duct containing flowing hot gases were film-cooled with water:

1. A liquid coolant film can be established and maintained along a duct wall in co-current flow with a hot gas, and the coolant will keep the duct wall below the boiling temperature of the
3 liquid while the liquid is vaporizing; after the liquid is vaporized, little additional cooling is obtained from the coolant vapor.

2. Cooling effectiveness, or the surface area cooled per unit coolant flow, decreases with coolant flow; it may thus be desirable to limit the amount of coolant introduced at any single axial position and to introduce it at several axial positions.

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Cleveland, Ohio.

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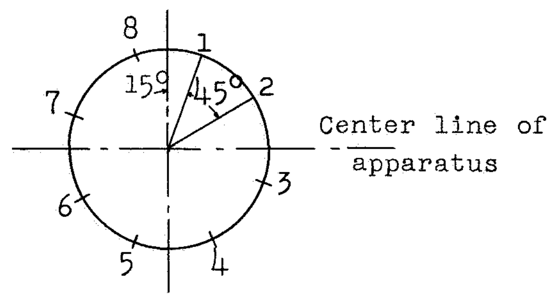
TABLE I - LOCATION OF THERMOCOUPLES ON FILM-COOLED TEST SECTIONS



End view of test section in upstream direction showing circumferential positions of thermocouples

Circumferential position	Location of thermocouples - distance downstream of coolant injection, in.	
	Rough-surface duct	Smooth-surface duct
1	[a]2-25,29,33,37,41,45	[a]3-25,29,33,37,41,45
2	13,21,29,45	5,9,13,17,21,25,29,45
3	2,5,9,13,21,29,45	5,9,13,17,21,25,29,33,37,41,45
4	13,21,29,45	5,9,13,17,21,25,29,45
5	2,5,9,13,21,29,45	5,9,13,17,21,25,29,33,37,41,45
6	13,21,29,45	5,9,13,17,21,25,29,45
7	2,5,9,13,21,29,45	5,9,13,17,21,25,29,33,37,41,45
8	13,21,29,45	5,9,13,17,21,25,29,45

[a]Thermocouples located at 1-in. intervals between these points.

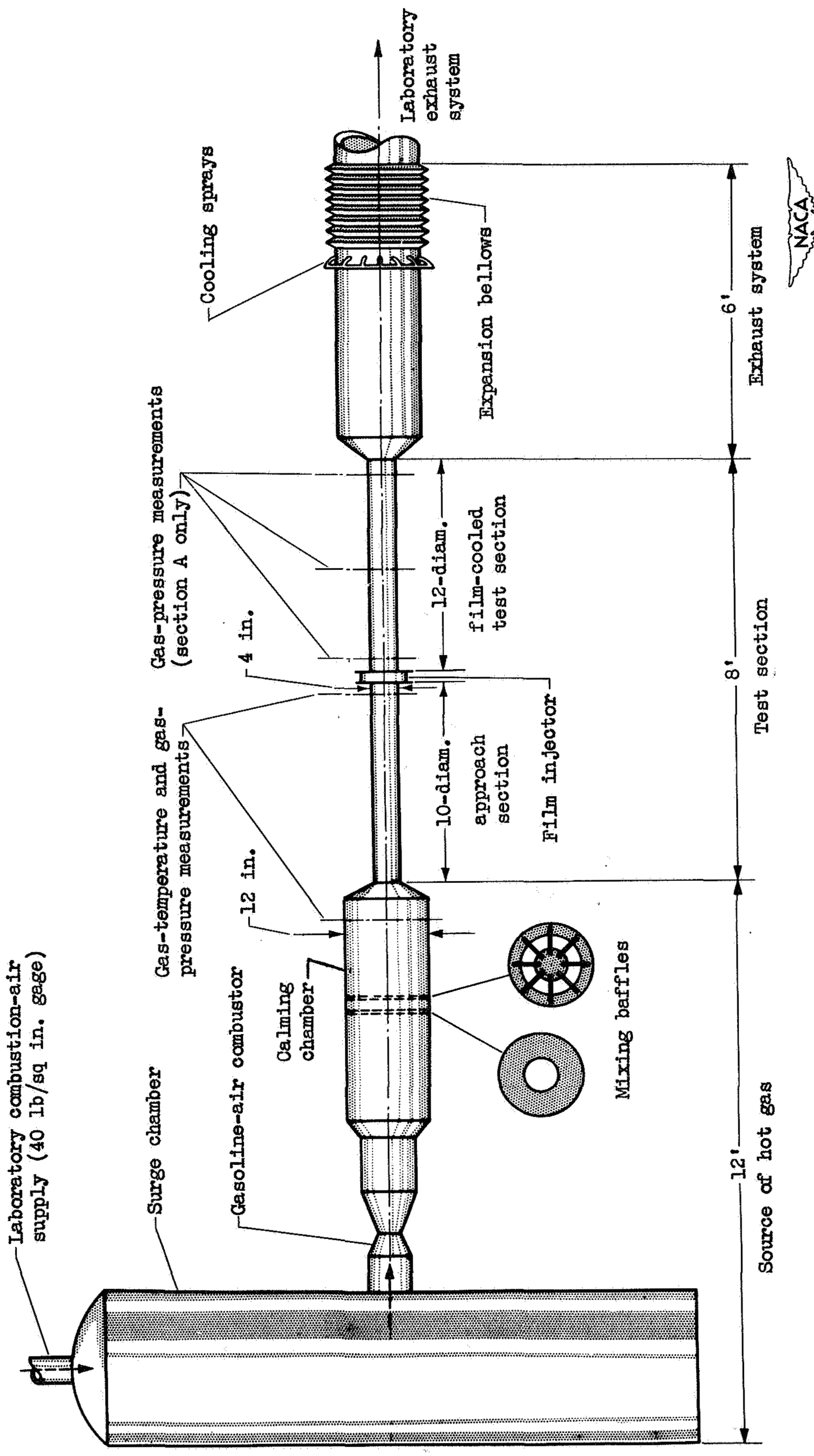


Figure 1. - Gas flow system.

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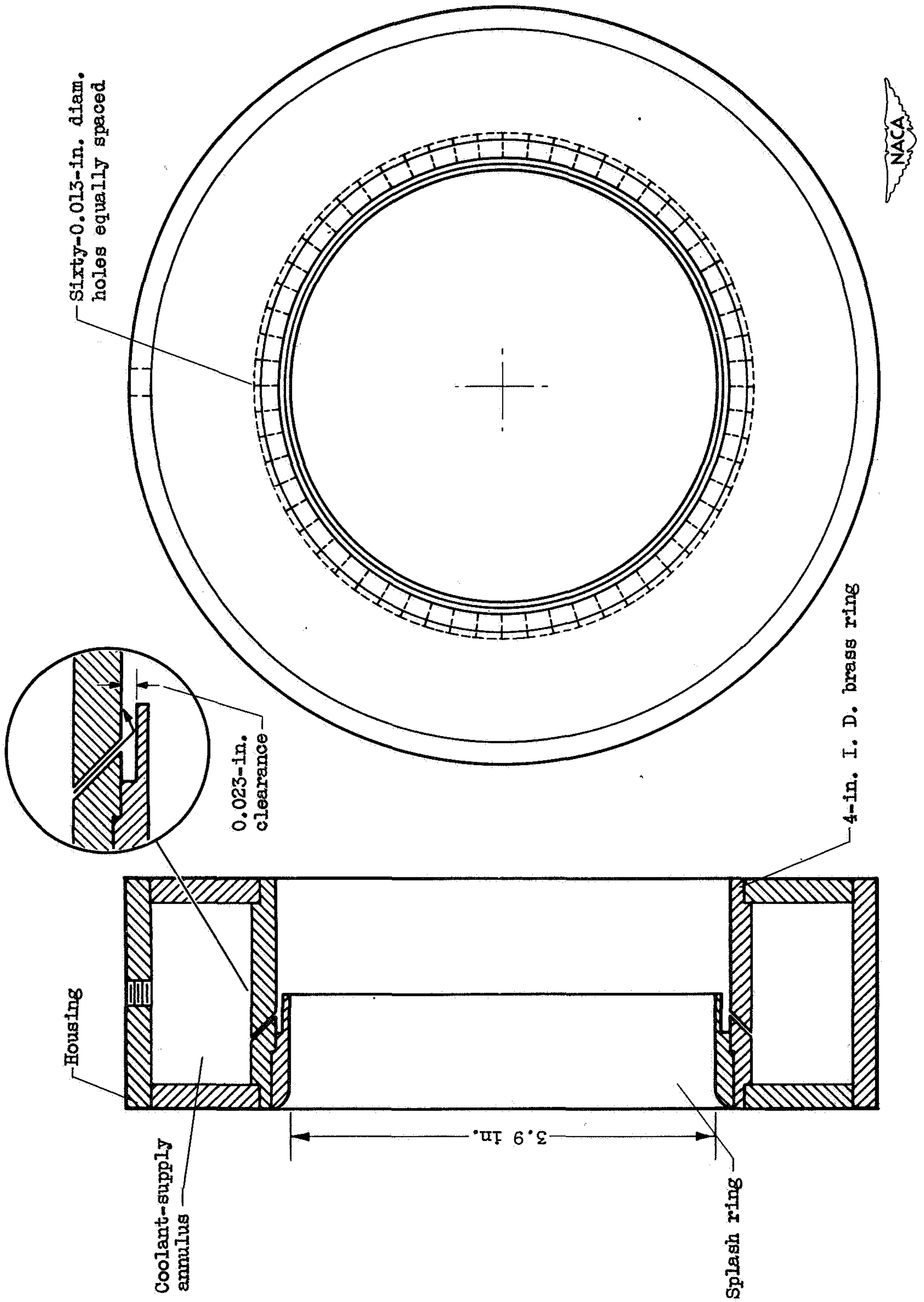
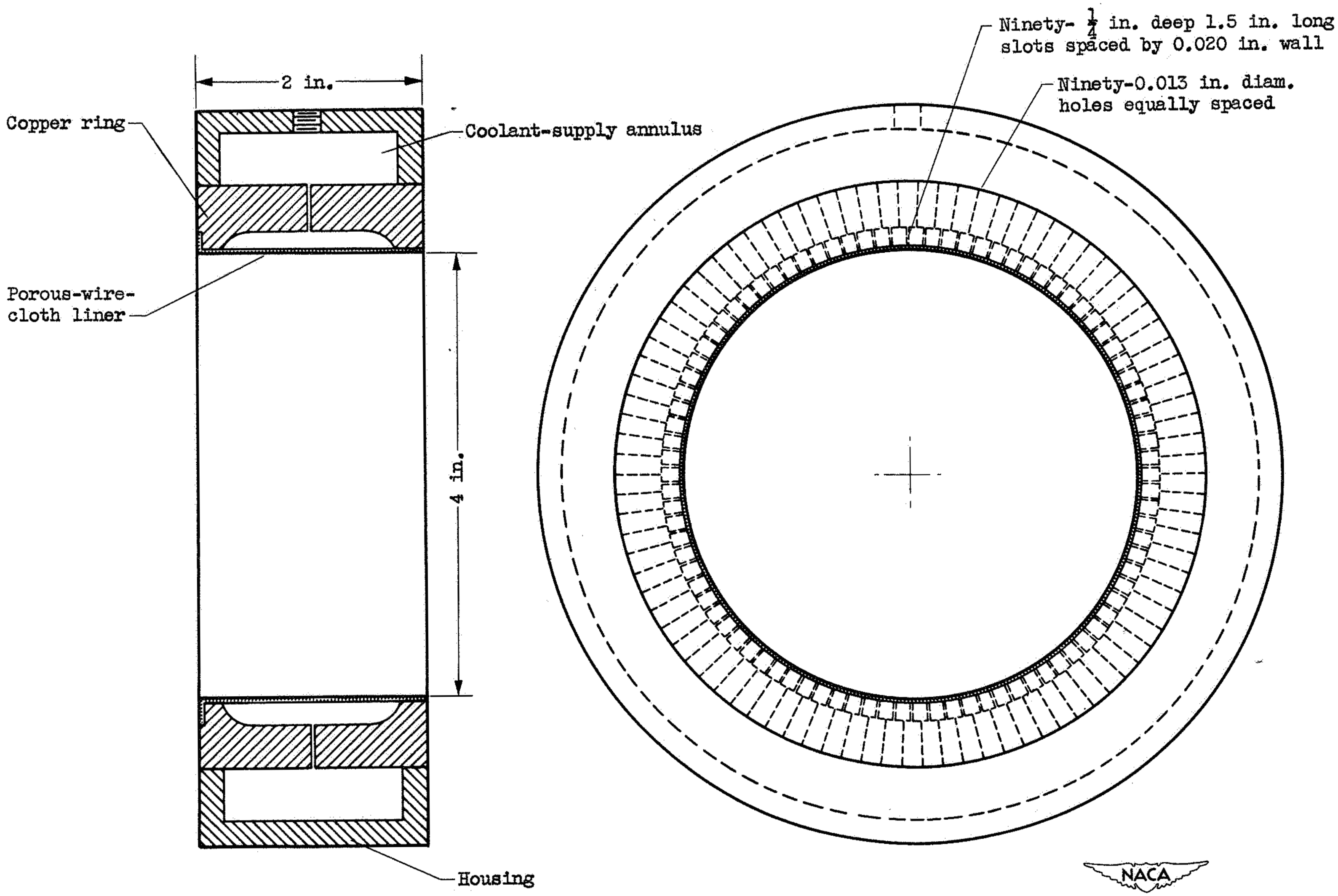


(a) Jet type.

Figure 2. - Coolant injectors.



(b) Porous-surface type.
Figure 2. - Concluded. Coolant injectors.

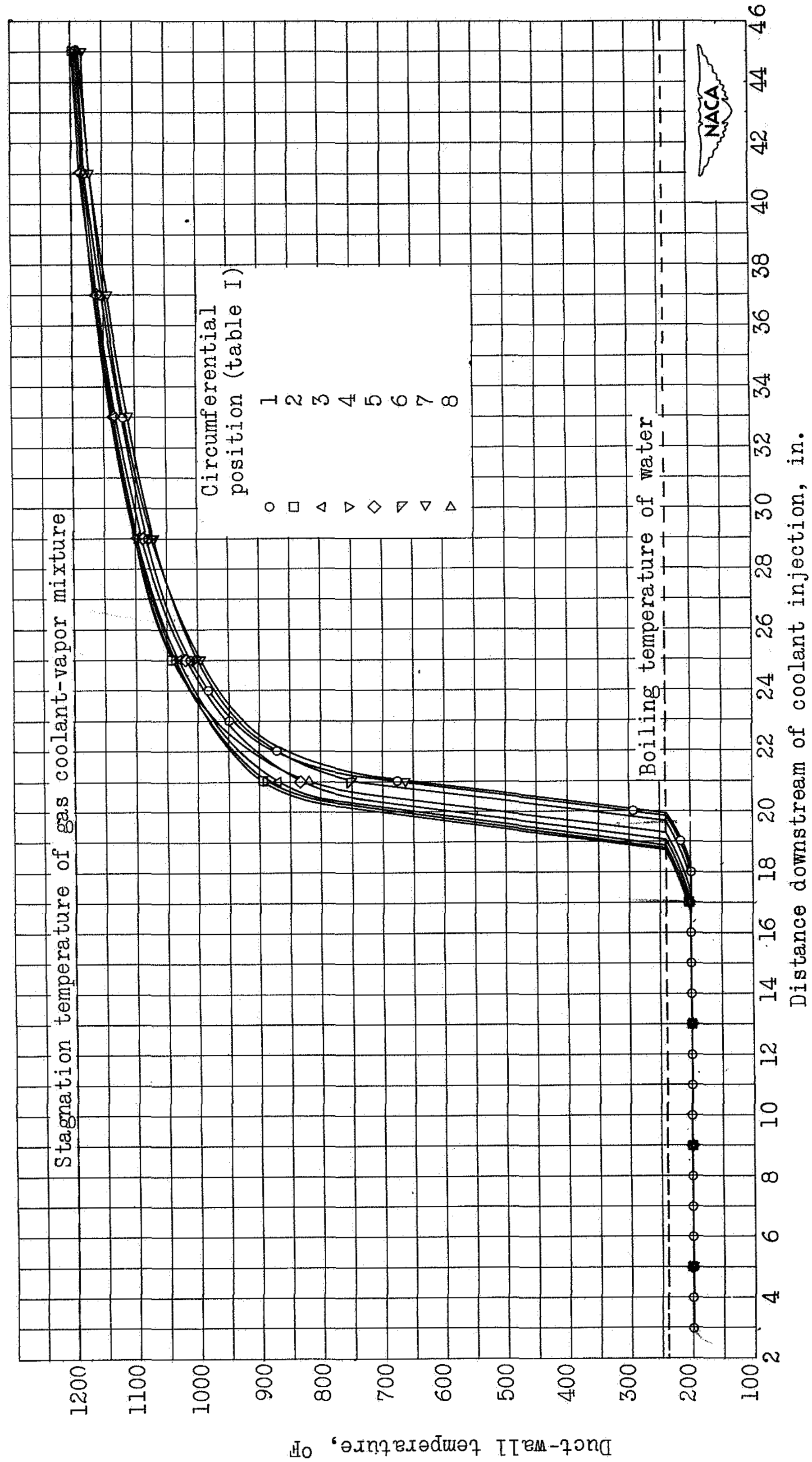
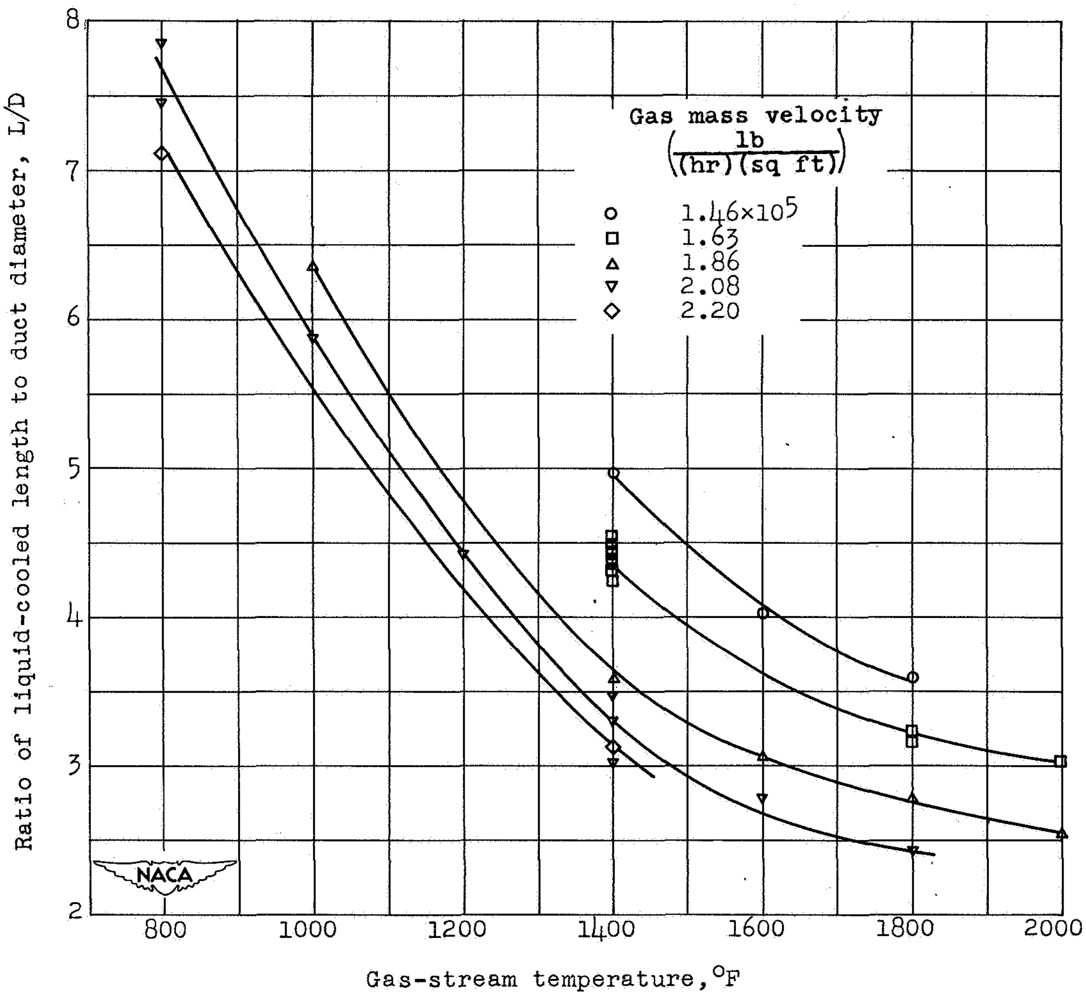


Figure 3. - Liquid-film-cooled length in smooth-surface duct. Gas temperature, 1400° F; gas mass velocity, 1.63×10^5 pounds per hour per square foot; ratio of coolant flow to gas flow, 2.6 percent.



(a) Effect of temperature at various gas mass velocities.

Figure 4. - Variation of liquid-cooled length with gas-stream temperature and mass velocity for water. Coolant flow, 464 pounds per hour; rough-surface duct.

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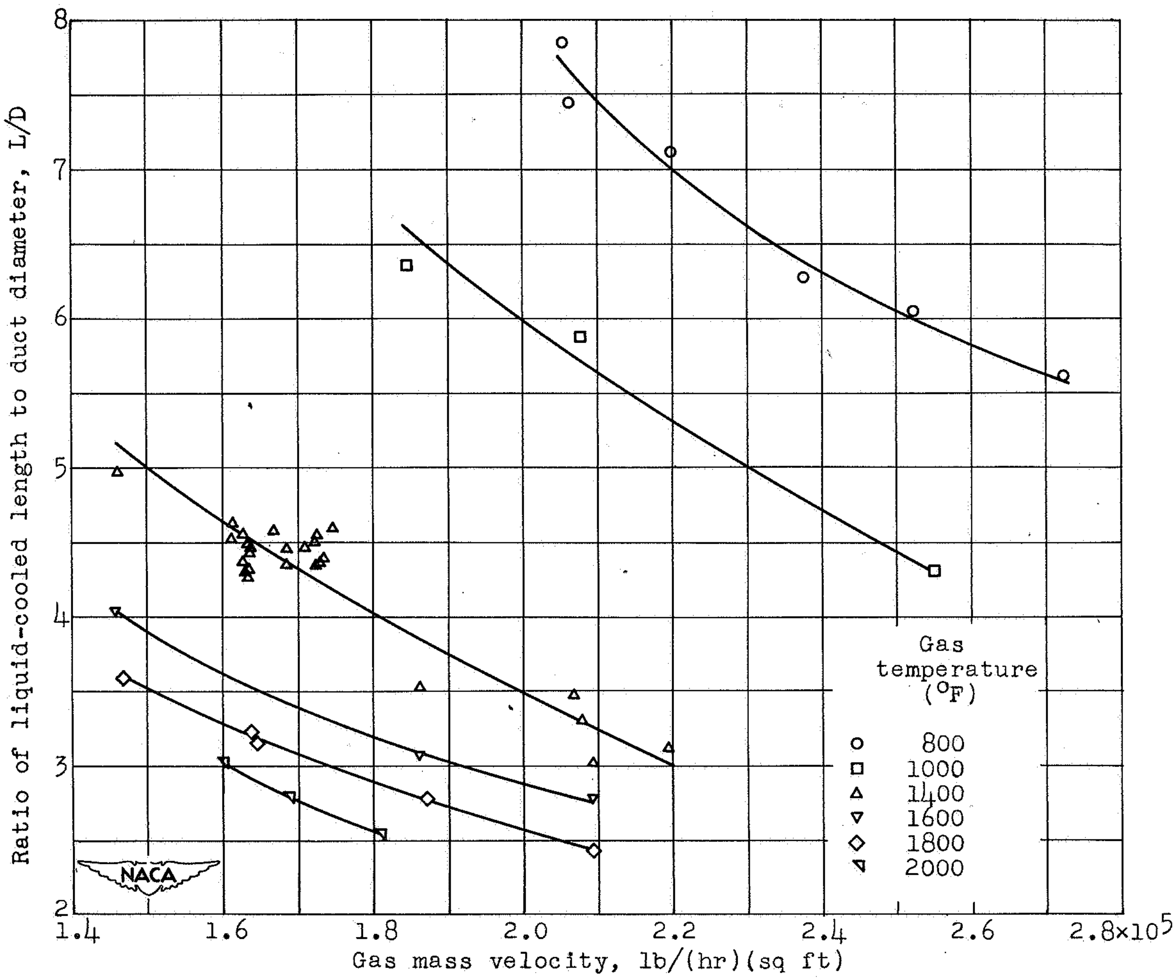
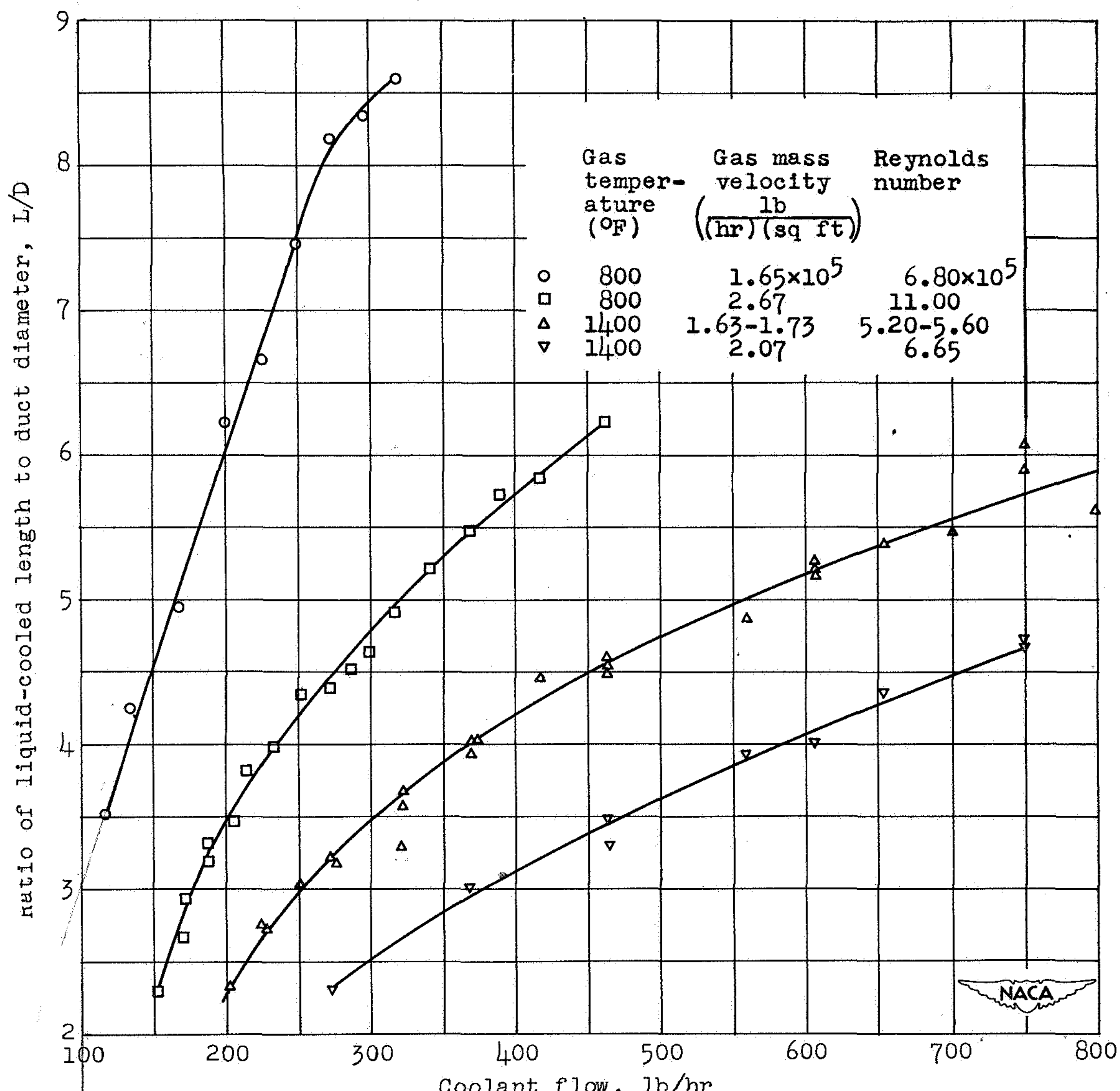


(b) Effect of gas mass velocity at various temperatures.

Figure 4. - Concluded. Variation of liquid-cooled length with gas-stream temperature and mass velocity for water. Coolant flow, 464 pounds per hour; rough-surface duct.



(a) Rough-surface duct.

Figure 5. - Variation of liquid-cooled length with coolant flow.

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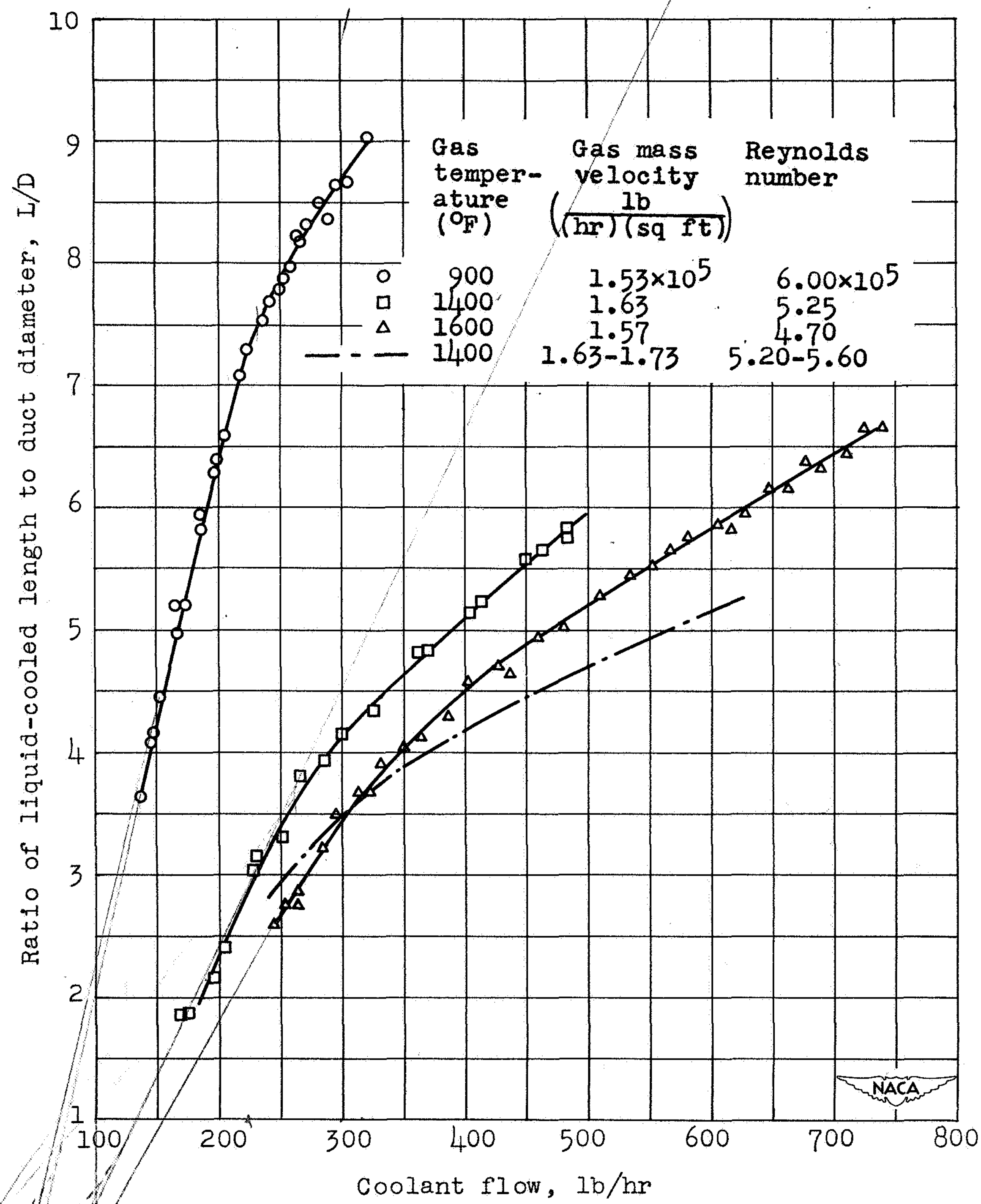

(b) Smooth-surface duct.

Figure 5. - Concluded. Variation of liquid-cooled length with coolant flow.

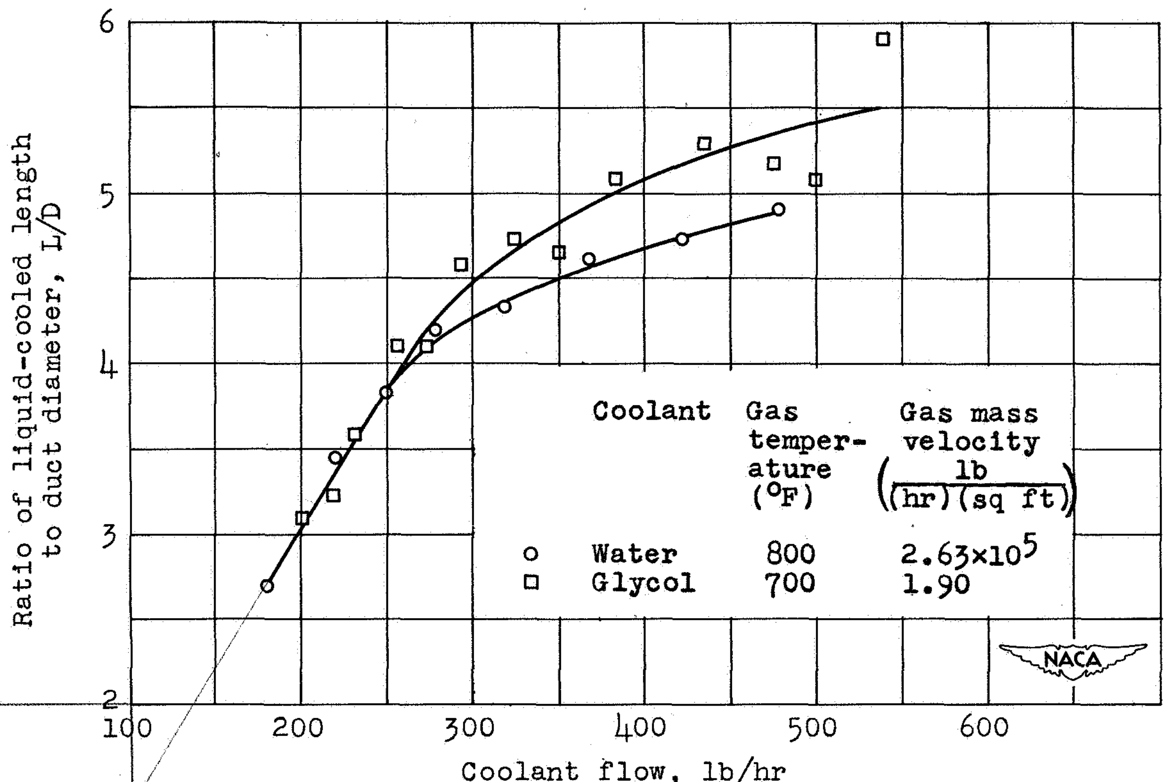

Figure 6. - Variation of liquid-cooled length with coolant flow. Coolants, water and ethylene glycol; glycol-roughened surface.

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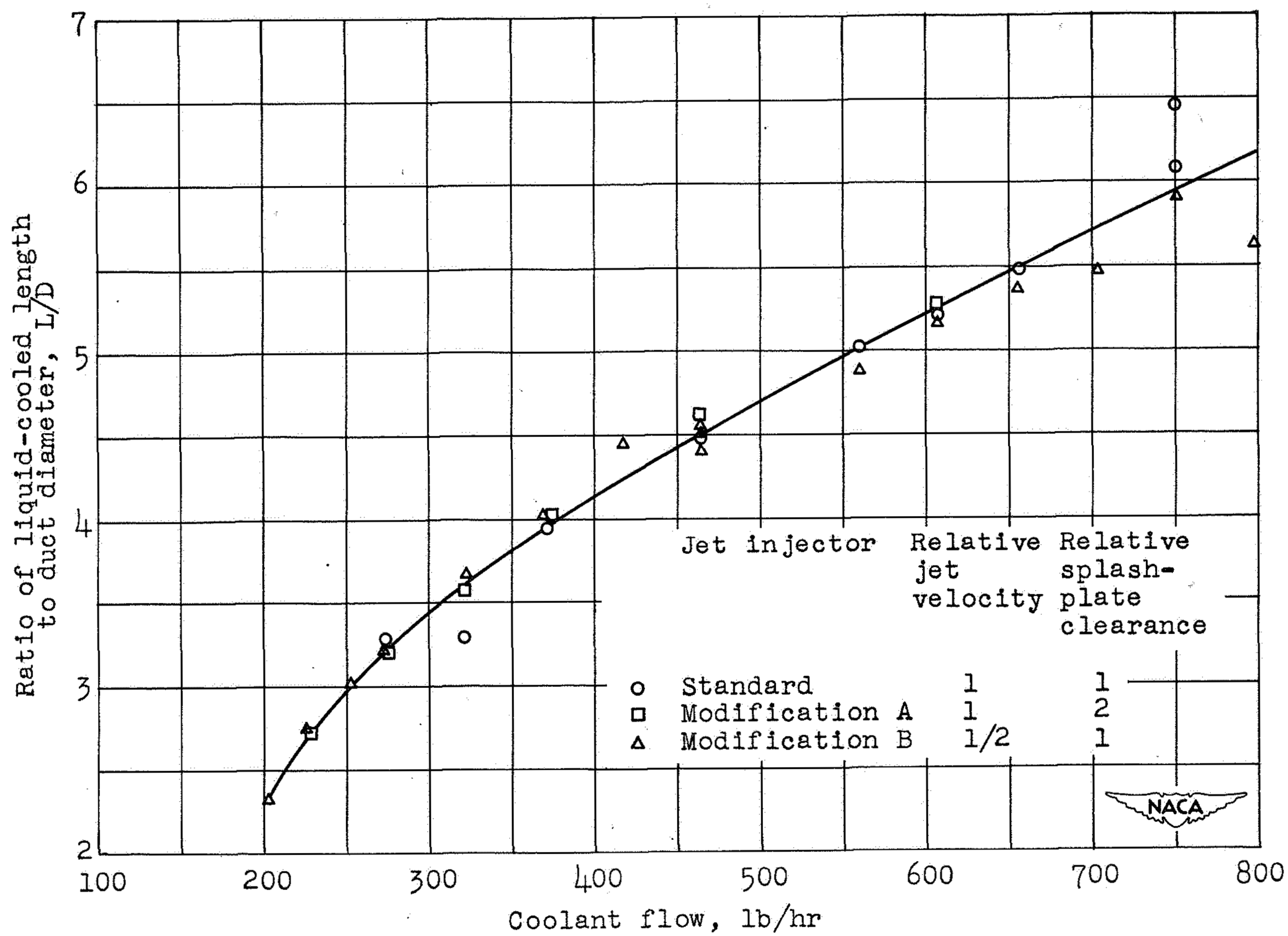
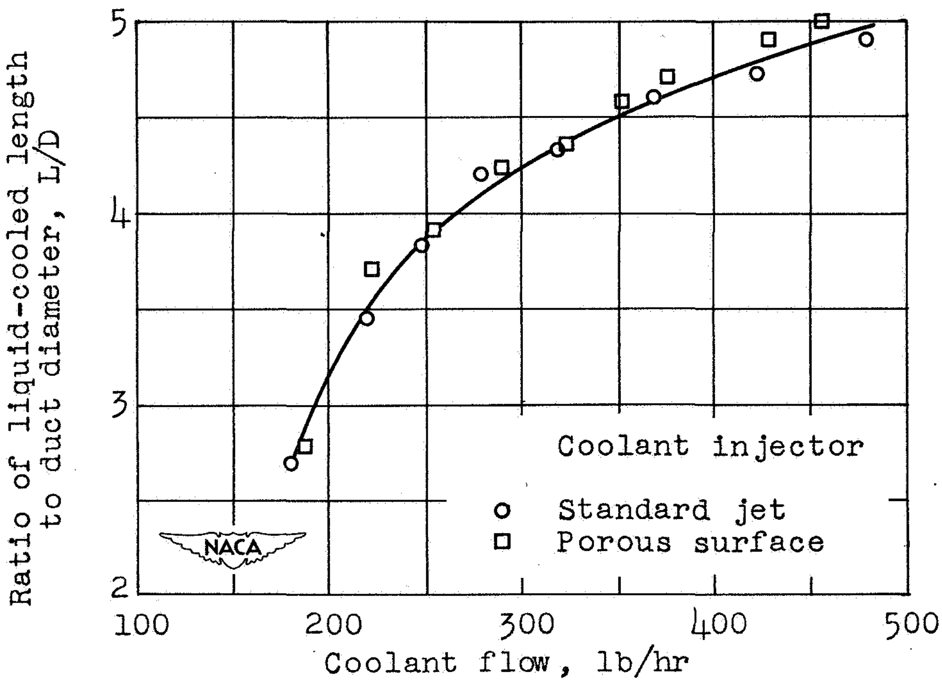


(a) Standard jet injector and two modifications; gas temperature, 1400° F; gas mass velocity, 1.63 to 1.73×10^5 pounds per hour per square foot; rough-surface duct.

Figure 7. - Variation of liquid-cooled length with coolant flow for various coolant injectors. Coolant, water.



(b) Standard jet and porous-surface injectors; gas temperature, 800° F; gas mass velocity, 2.58×10^5 pounds per hour per square foot; glycol-roughened surface.

Figure 7. - Concluded. Variation of liquid-cooled length with coolant flow for various coolant injectors. Coolant, water.

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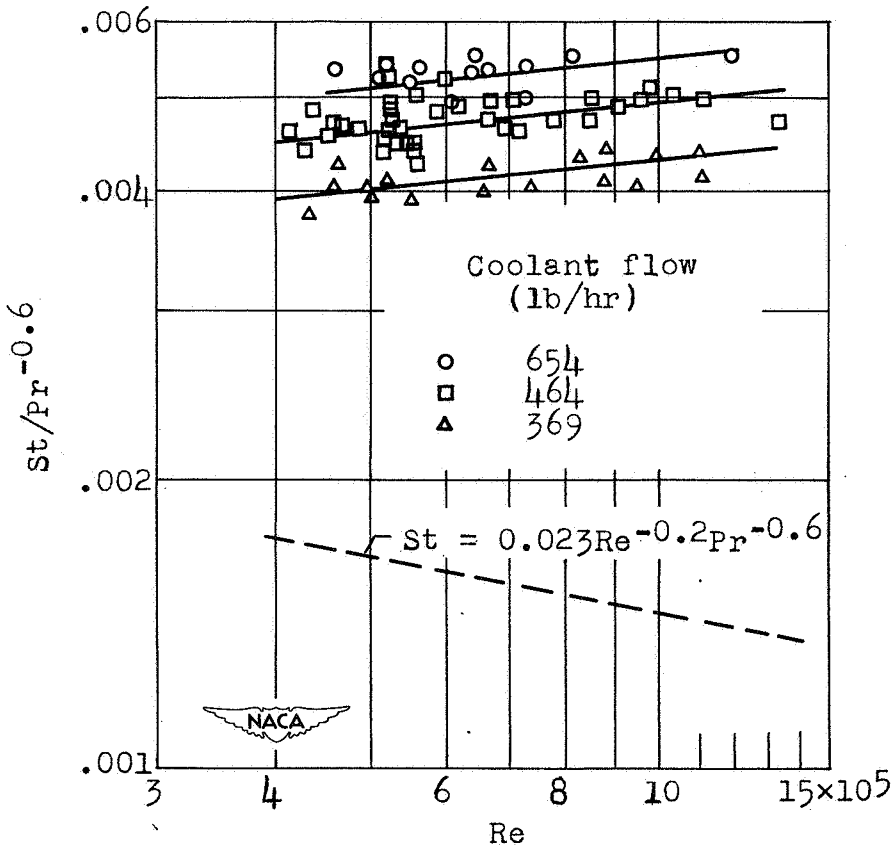
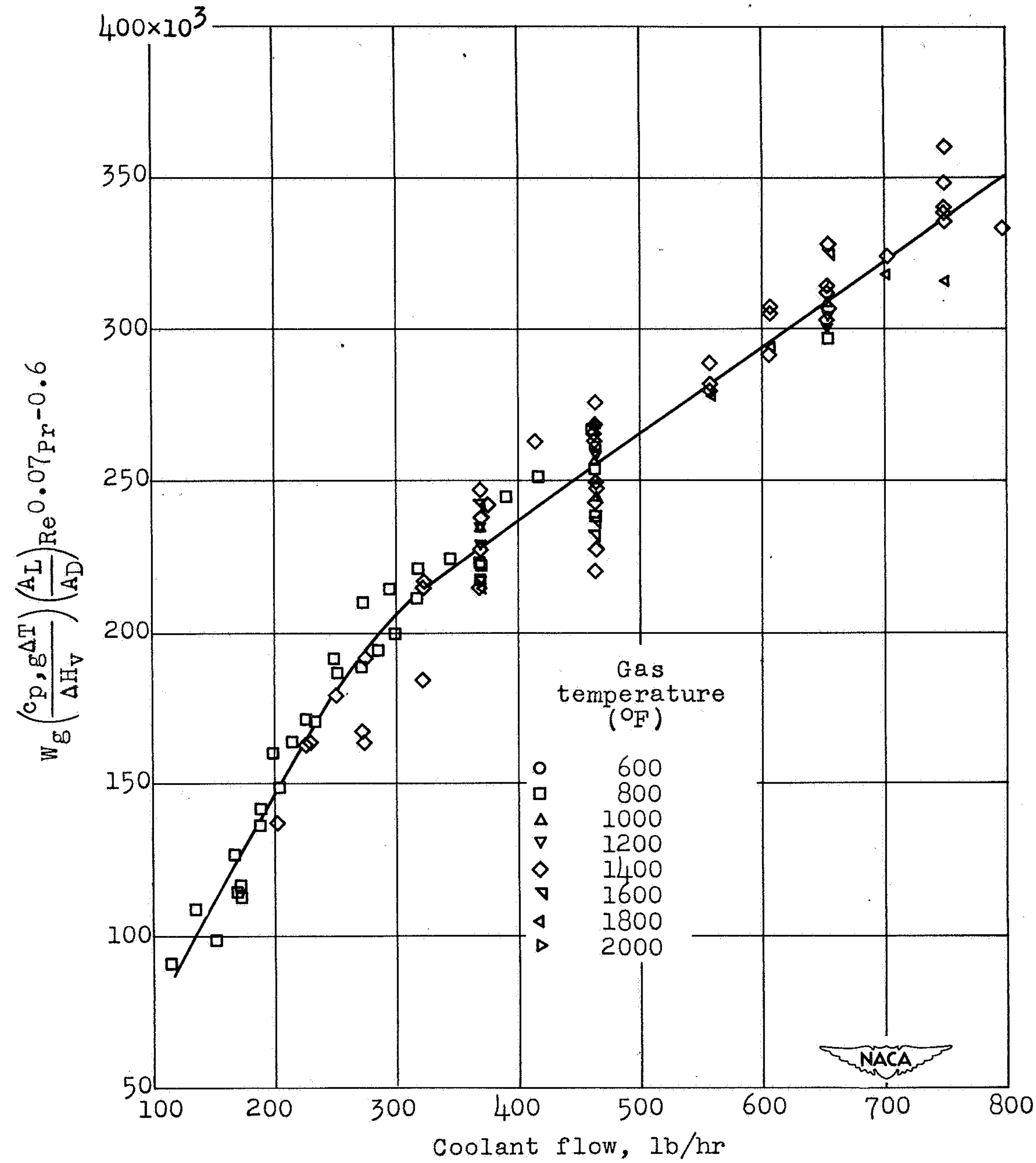


Figure 8. - Correlation of data at constant coolant flow by heat-transfer analogy. Gas temperatures, 600° to 2000° F; gas mass velocities, 1.43 to 2.93×10^5 pounds per hour per square foot; rough-surface duct.



(a) Coolant, water; rough-surface duct; gas mass velocities, 1.43 to 2.93×10^5 pounds per hour per square foot; Reynolds numbers, 4 to 14×10^5.

Figure 9. - Variation of cooling parameter with coolant flow from heat-transfer analogy.

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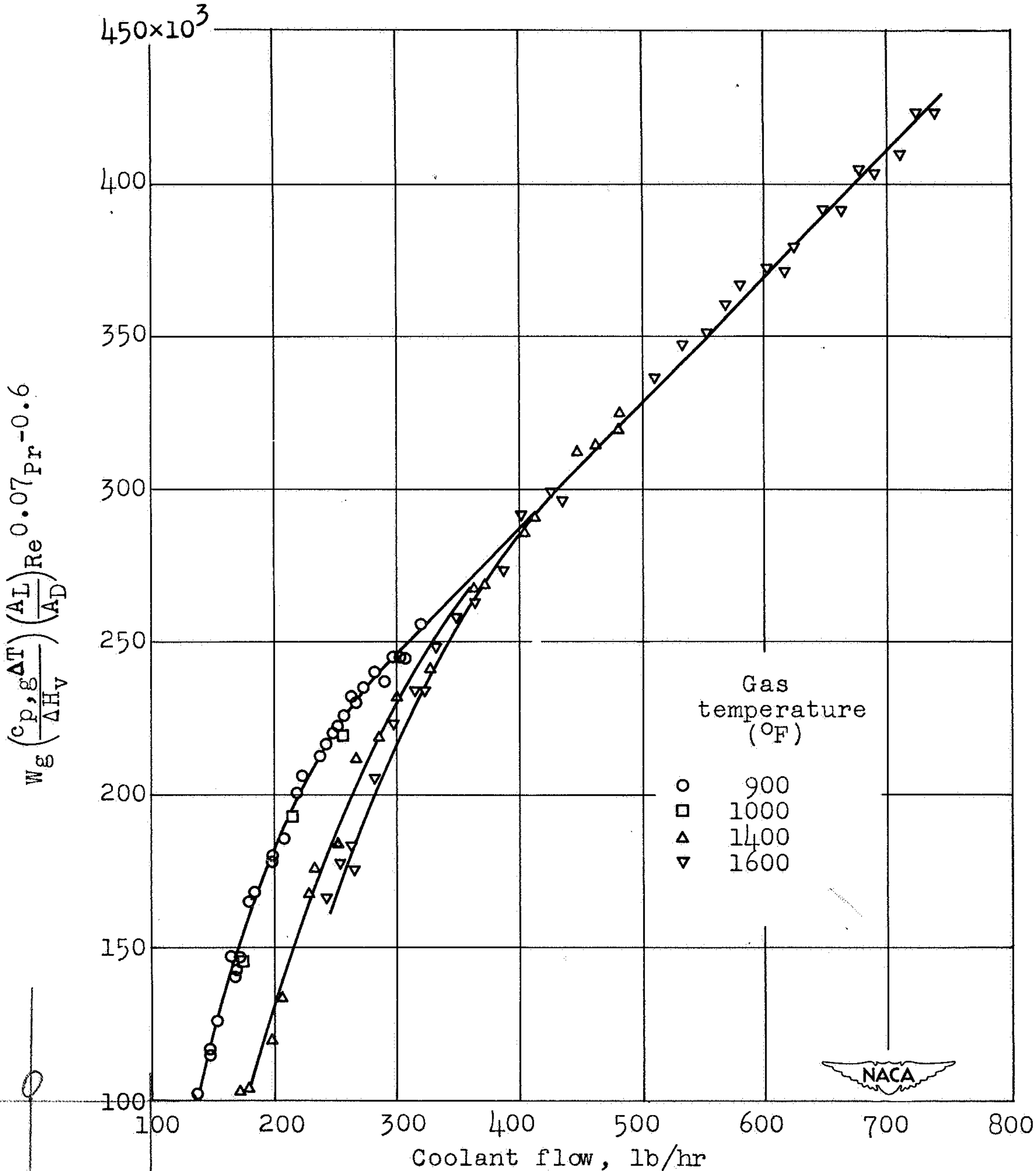
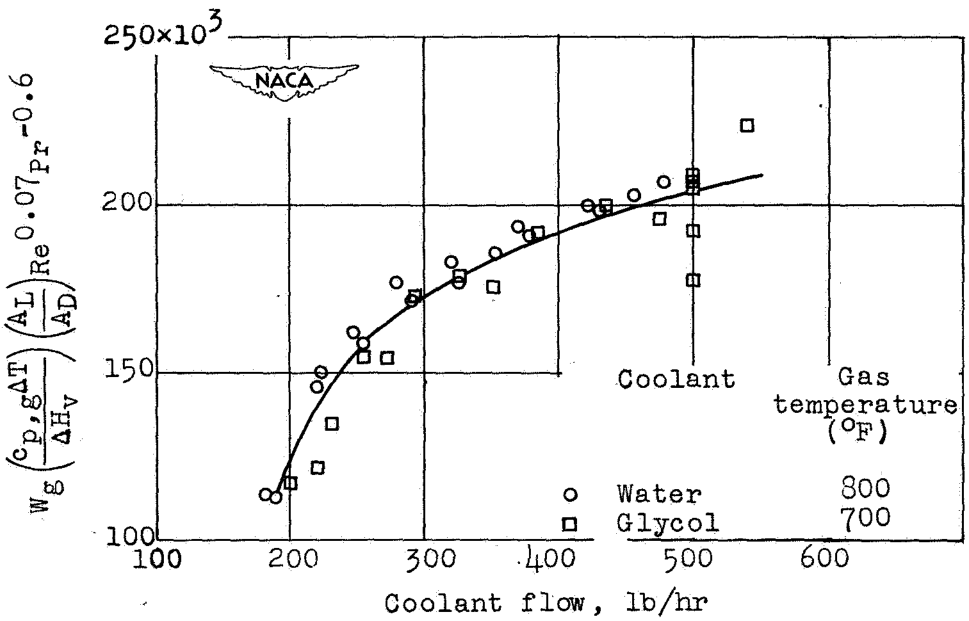


(b) Coolant, water; smooth-surface duct; gas mass velocities, 1.53 to 1.64×10^5 pounds per hour per square foot; Reynolds numbers, 4.7 to 6.07×10^5.

Figure 9. - Continued. Variation of cooling parameter with coolant flow from heat-transfer analogy.



(c) Coolants, water and glycol; rough-surface duct with glycol deposits; gas mass velocities, 1.43 to 2.75×10^5 pounds per hour per square foot; Reynolds numbers, 6.2 to 12×10^5.

Figure 9. - Concluded. Variation of cooling parameter with coolant flow from heat-transfer analogy.