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RESEARCH MEMORANDUM

INTERNAL-FILM-COOLING EXPERIMENTS WITH 2- AND 4-INCH SMOOTH-SURFACE TUBES AND GAS TEMPERATURES TO 2000° F

By George R. Kinney

Lewis Flight Propulsion Laboratory
Cleveland, Ohio

NATIONAL ADVISORY COMMITTEE
FOR AERONAUTICS
WASHINGTON
April 29, 1952

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INTERNAL-FILM-COOLING EXPERIMENTS WITH 2- AND 4-INCH SMOOTH-SURFACE TUBES AND GAS TEMPERATURES TO 2000° F

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SUMMARY

An investigation of liquid-film cooling was conducted in 2- and 4-inch-diameter straight tubes having honed inner surfaces with air flows at temperatures from 800° to 2000° F and diameter Reynolds numbers from 220,000 to 1,100,000. The film coolant, water, was injected at a single axial position on the tube at flow rates from 0.02 to 0.24 pound per second per foot of tube circumference (0.8 to 12 percent of the air flow).

Correlation was obtained for heat-transfer from the hot air to the liquid-cooling film with the 2- and 4-inch-diameter smooth-surface tubes and was the same as was found in a previous investigation with a 4-inch-diameter rough-surface tube. Heat-transfer coefficients were about 20 percent lower with the smooth-surface tubes than with the rough-surface tube.

Effectiveness of the coolant was decreased by coolant-film disturbances in a 2-inch-diameter tube, as was found in a previous investigation in a 4-inch-diameter tube. The approximate coolant flow above which reduced cooling effectiveness resulted was predicted by a method developed in a previous investigation.

All film-cooling data obtained with smooth-surface tubes were generalized by means of heat-transfer correlation and data obtained over a range of coolant flows. This generalization of data facilitates the estimation of the flow of coolant required to film cool a tube for a desired length, over a wide range of conditions, when the temperature and flow of the hot gas are known.

INTRODUCTION

Combustion chambers and ducts which are subjected to extremely high heat transfer from flowing hot gases may be difficult to cool by circulation of a fluid over the outer walls. Internal-film cooling, wherein a coolant film is formed between the hot gases and the duct wall to maintain the wall at low temperature, may be used in such cases. The effectiveness of liquid-film cooling in reducing heat transfer to rocket nozzles and combustion chambers is described in references 1 and 2. Investigations concerned with establishing liquid-cooling films for rocket engines are described in references 3 and 4; and reference 4 summarizes work on rocket heat transfer, sweat cooling, and film cooling.

A general investigation of internal-liquid-film cooling is being conducted at the NACA Lewis laboratory to obtain correlation of experimental data that will allow predictions of film-coolant requirements for specific cooling problems. One phase of the fundamental investigation involves film-cooling experiments with hot-air flows at temperatures to 2000° F to determine the cooling effectiveness of water films on the inner surfaces of straight tubes. Correlation was obtained for heat transfer between flowing hot air and liquid films over a range of air flow and temperature for constant coolant flows with a 4-inch-diameter straight tube (reference 5). The relation between liquid-cooled area and coolant flow for given gas-stream conditions was nonlinear; the effectiveness of a given amount of coolant decreased with increased coolant flow. Heat-transfer correlations obtained were therefore dependent upon coolant flow. Reference 6 describes a visual investigation of liquid films on the inner surfaces of straight tubes in co-current flow with air. This investigation showed that as liquid flow per circumferential length of tube increases the liquid film changes from (1) relatively smooth, to (2) slightly rough, to (3) increasingly rough. These changes were related to the decreased cooling effectiveness with increased coolant flow with film cooling. References 7 and 8 report related investigations of co-current liquid-gas flow; the relative flow rates investigated were not in the range encountered in film-cooling applications.

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Most of the data reported in reference 5 were obtained with a tube in which the cooling film was partly disturbed by instrumentation and the inner surface did not have a smooth finish. Data were also obtained with a smooth-surface tube in which the film was not disturbed by instrumentation, and it appeared that correlation similar to that with the rough-surface tube but with greater cooling effectiveness was obtained. The data obtained with this tube, however, were insufficient to check the correlation. The data were obtained with only 4-inch-diameter tubes.

The investigation of liquid-film cooling in straight tubes with hot-air flows was continued at the Lewis laboratory in order to: (1) check the heat-transfer correlation presented in reference 5 with additional data in 2- and 4-inch-diameter tubes, (2) determine if cooling-film disturbances which resulted in reduced cooling effectiveness in 4-inch-diameter tubes have the same effect in a smaller tube and if their

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occurrence can be predicted by a method developed in reference 6, and (3) generalize the film-cooling data by means of heat-transfer correlation and data obtained over a range of coolant flows.

The generalization of all the film-cooling data obtained with smooth-surface tubes is intended to facilitate the estimation over a wide range of conditions of the flow of coolant required to film cool a tube for a desired distance when the temperature and flow of the hot gas are known.

The experiments were conducted in 2- and 4-inch-diameter tubes having honed inner surfaces with hot-air flows at temperatures from 800° to 2000° F and Reynolds numbers from 220,000 to 1,100,000. The film coolant was water injected at a single axial position on the tube at flow rates from 0.02 to 0.24 pound per second per foot of tube circumference (0.8 to 12 percent of the air flow). The length of tube cooled by the water was determined by means of thermocouples on the tube outer surface.

SYMBOLS

The following symbols are used in this report:

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

a proportionality constant

C_p average specific heat at constant pressure (Btu/(lb)(°F))

D inside diameter of film-cooled tube (ft)

G mass velocity (lb/(sec)(sq ft))

H_v heat of vaporization of coolant (Btu/lb)

ΔH change in enthalpy of liquid coolant from entering condition through vaporization (Btu/lb)

h heat-transfer coefficient (Btu/(sec)(sq ft)(°F))

k thermal conductivity (Btu/(sec)(sq ft)(°F/ft))

L liquid-cooled length (ft)

Pr Prandtl number, $C_p\mu/k$

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Re	Reynolds number, DG/μ
St	Stanton number, h/C_pG
$T_{g,1}$	stagnation temperature of gas stream entering film-cooled tube (°F)
$T_{g,2}$	stagnation temperature of gas-coolant vapor mixture (°F)
$T_{g,a}$	arithmetic mean of $T_{g,1}$ and $T_{g,2}$ (°F)
T_i	temperature of coolant before injection (°F)
T_s	temperature of liquid-coolant surface (assumed at saturation temperature) (°F)
W	flow rate (lb/sec)
μ	viscosity (lb/(sec)(ft))

Subscripts:

g	gas stream
c	liquid coolant
v	coolant vapor

APPARATUS

Flow System

The flow system (fig. 1) is the same as that used in reference 5 except for the test sections. It consists essentially of three parts: (1) source of hot air at a uniform temperature, (2) test sections, and (3) expansion and exhaust system. The hot-air source 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. Test sections of 2- and 4-inch diameters were used. They consisted of an Inconel approach section 40 inches long, a coolant injector, and a film-cooled tube made of Inconel with a 1/8-inch wall, 48 inches long. The exhaust section contained a series of water sprays to quench hot air; this section was connected to the laboratory exhaust system. An expansion bellows installed downstream of the exhaust-quenching sprays allowed for expansion of the apparatus when using the 4-inch-diameter test section; the 2-inch-diameter film-cooled tube fitted into a packed housing to allow for expansion of the apparatus. The assembly was supported at the surge chamber, by roller stands located upstream and downstream of the test section, and by a ring stand at the liquid injector.

Coolant-Injection System

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The coolant-injection system was also the same as that described in reference 5. It consisted of a supply reservoir, filters, a positive-displacement pump, adjustable pressure regulators (which controlled flow), rotameters, and coolant injectors.

The two coolant injectors (fig. 2) were those described in reference 6; they were of similar construction. Each consisted of a metal ring with slots milled into the inner surface about the circumference. The 2- and 4-inch-diameter coolant injectors had 60 and 90 slots, respectively. Holes 0.013 inch in diameter were drilled through the ring into each of the slots. A housing, which provided a supply annulus for the coolant, fitted over the ring. The coolant supplied to the annulus 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, thus providing a uniform distribution of the flow about the circumference. The slots spread the coolant flow over a large area, thereby reducing the flow velocity. As 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 flow over the surface of the injector carried the coolant downstream along the inner surface of the film-cooled tube.

Film-Cooled Tubes

Two film-cooled tubes were used for the experiments, one 2-inch and the other 4-inch inside diameter. Both tubes were 1/8-inch-wall Inconel tubes, 48 inches long. The 4-inch tube was insulated with 3 inches of high-temperature Fiberglass insulation. The 2-inch tube was not insulated because of difficulty in assembly; the estimated heat loss from the tube was not large enough to affect the film-cooling results. The 4-inch-diameter tube was the smooth-surface tube described in reference 5; it was a seamless tube honed to obtain a smooth inside surface. The 2-inch-diameter tube was finished in the same manner.

Instrumentation

<u>Flow rate</u>. - Air flows were measured within 2 percent by means of an orifice conforming to standard A.S.T.M. specifications and a differential

water manometer. Coolant flows were measured within 1 to 4 percent by means of rotameters.

Pressure. - Static pressure in the air stream was measured in the approach section 3 inches upstream of the coolant injector by means of a mercury manometer. The measurements were useful in determining approximate air densities and velocities and boiling temperature of the film coolant.

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Temperature. - Air temperatures were measured by means of chromel-alumel thermocouples and a self-balancing potentiometer. Rakes containing 20 thermocouples were located in the calming section, and a rake containing four thermocouples for the 4-inch test section and one thermocouple for the 2-inch test section was placed in the approach section 3 inches upstream of the coolant injector. The estimated accuracy of air-temperature measurements varied from ±12° F at 800° 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 air-stream temperature during runs. The recovery factor for the thermocouples located in the approach section was found to be in the order of 0.9 (reference 5). Because the recovery factor was high, the total temperatures as measured by thermocouples in the approach section were used; the differences between the temperatures used and the temperatures obtained by correcting on the basis of recovery factor were less than 1 percent. Wall temperatures on the film-cooled tube were measured by means of chromel-alumel thermocouples and a recording potentiometer. The thermocouples, which were welded to the outer surface of the tube, were spaced along the length of the tube at eight positions around the circumference of the 4-inch tube (table I) and four positions around the circumference of the 2-inch tube (table II). Wall temperatures were measured by these thermocouples within ±10° F.

PROCEDURE

Operating Conditions

The operating procedure consisted in setting coolant flow, air flow, and air temperature at desired values, allowing sufficient time for conditions to stabilize, and recording the data. Because the combustion gases resulting from the burning of gasoline to heat the air did not amount to more than a few percentage of the air flow, negligible error was introduced by using the physical properties of air for the total gas flow. The operating conditions covered the following ranges:

Air temperature[a], °F	800-2000
Air flow, lb/sec	0.89-6.5
Coolant flow, lb/sec	0.01-0.21
Coolant-flow rate/air-flow rate, percent	0.8-12
Coolant flow per circumferential length, lb/(sec)(ft)	0.02-0.24

[a]In approach section 3 in. upstream of coolant injector.

These conditions give the following values in the approach section 3 inches upstream of the coolant injector. Values of specific heat for air were obtained from reference 9 and values of thermal conductivity and viscosity for air were obtained from reference 10:

Reynolds number	220,000-1,100,000
Mach number	0.5-0.7
Prandtl number	0.63-0.70
Air mass velocity, lb/(sec)(sq ft)	39.4-81.7
Air velocity, ft/sec	1000-1700

Liquid-Cooled Length

Cooling effectiveness was determined by plotting the wall temperatures of the film-cooled tube as a function of distance from the point of coolant injection. A typical plot from a run with the 2-inch tube is shown in figure 3. The tube-wall temperature remains below the boiling temperature of the coolant (water) for approximately 16.5 inches downstream of coolant injection, rises to the boiling temperature, and then rises rapidly, approaching 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 four circumferential positions. Similar plots are obtained for the 4-inch tube (shown in reference 5), and the liquid-cooled length is determined from an average of the eight circumferential positions. For the 4-inch tube, the agreement of the liquid-cooled length for the eight circumferential positions was within about 10 to 15 percent; at very low coolant flows greater variation occurred. For the 2-inch tube, even distribution of the coolant was more difficult to achieve; agreement of liquid-cooled lengths for the different positions was generally within about 25 percent, but in many cases one of the four positions would differ from the others by as much as 40 percent.

One reason for the spread of the data on the 2-inch tube was the relative effect of gravity, which tends to decrease the proportionate film thickness in the upper portions of the horizontal tube, thus resulting in shorter liquid-cooled lengths there. This effect was not noticeable for most of the experiments with the 4-inch tube and was slight for the other experiments. The effect of gravity in the 2-inch tube was noticeable in all the experiments, but the magnitude of the gravity effect could not be determined accurately for most of the experiments because consistent even distribution of the coolant into the tube was not obtained. The effect, however, resulted in the liquid-cooled lengths in the upper portion of the tube being from 5 to 20 percent shorter than those in the lower portion. For the run plotted in figure 3, the liquid-cooled lengths in the upper positions were about 15 percent shorter than those in the lower positions of the tube. No trends in the magnitude of the gravity effect were noticeable with the different gas and coolant flows investigated. Gravity effects were of insufficient magnitude to affect the trends observed in film cooling.

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Carbon Deposition on Film-Cooled Tube

Each time a run was made carbon was deposited around the inner surface of the tube immediately downstream of where liquid coolant ceased for a distance of about 1/2 inch. Carbon did not form on the surface covered by the liquid film nor further downstream where the surface became hotter. These deposits will disrupt a liquid film and thereby reduce cooling effectiveness; the rate of carbon deposition and the resulting effect were considerably greater in the 2-inch tube than in the 4-inch tube. For these reasons all the experiments in the 2-inch tube were made with the liquid-cooled length for each succeeding run less than on the previous run so that the deposits could not disrupt the film. After each period of running, it was necessary to remove and clean the tube of carbon deposits before further running at long liquid-cooled lengths could be made. Most of the experiments in the 4-inch tube were also made using this procedure; for the other experiments the carbon deposition was not great enough to reduce liquid-cooled lengths more than a few percent.

Reproducibility of Results

Consideration of 12 different runs with the 4-inch tube showed that reproducibility of liquid-cooled length at the same experimental conditions was within 5 percent; reproducibility with the 2-inch tube was within 5 percent for six runs, but variations of 15 percent were obtained with three others. Measurements of liquid-cooled length with the same experimental conditions to determine reproducibility could not be made without disassembly and cleaning of the film-cooled tube because of carbon deposits on the surface, which reduced cooling effectiveness. The reason for the large variations during some of the runs with the 2-inch tube is not known, but it is believed to be due to slight changes in alinement between the coolant injector and tube when reassembly was made, which in some cases would reduce cooling effectiveness.

RESULTS AND DISCUSSION

Effect of Coolant Flow per Circumferential Length on Liquid-Cooled Length

Liquid-cooled length in the 4-inch tube is plotted as a function of coolant flow per circumferential length in figure 4(a) for different constant gas-stream conditions. Data are shown from reference 5 at three different gas temperatures with approximately the same gas flow; also shown are additional data at different gas flows with the same gas temperature. For each of the different gas-stream conditions, the relation between liquid-cooled length and coolant flow per unit circumferential length is nonlinear and shows the same trends. The cooling effectiveness of a given amount of coolant is greatest at the low coolant flows, decreases appreciably in the range of coolant flow between 0.05 and 0.09 pound per second per foot, and does not change greatly at the higher flows.

Liquid-cooled length in the 2-inch tube is plotted as a function of coolant flow per circumferential length in figure 4(b) for different constant gas-stream conditions; data in the 4-inch smooth-surface tube from figure 4(a) are reproduced for comparison. Each of the curves shows the same trends, and the large decrease in cooling effectiveness occurs in the same region of coolant flow per circumferential length.

The decrease in cooling effectiveness encountered in film cooling was related to disturbances of the liquid-coolant film (reference 6), and analysis of the flow system gave an explanation for the formation of these disturbances. Conditions for the formation of film disturbances were found to be dependent chiefly on the liquid flow per circumferential length of tube and liquid viscosity; these were expressed by a flow-rate parameter $W_c/\pi D\mu_c$.

Values of the flow-rate parameter, determined for coolant flows representing medians over which marked changes in cooling effectiveness occur for the curves of figure 4, vary between 265 and 450.

Correlation of Heat Transfer from Hot Gas to Liquid Film for Constant Coolant Flow per Circumferential Length

Heat-transfer correlation was made in the manner described in reference 5, except that the gas temperatures used were an arithmetic mean between the air temperature at the tube entrance and the air-coolant-vapor equilibrium temperature instead of entrance-air temperatures. Differences between entrance-air temperatures and average temperatures in the film-cooled tube were not large for the experiments of reference 5, which were all conducted in a 4-inch tube. At the gas-coolant flow ratios for the experiments in the 2-inch tube, the differences were considerably greater and the necessity for using a mean gas temperature for the correlation was evident. Heat-transfer coefficients between the hot-gas stream and the liquid-cooling film were calculated as follows:

$$h = \frac{W_c\left[C_{p,c}(T_s-T_i) + H_v\right]}{(T_{g,a}-T_s)(\pi DL)} \tag{1}$$

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where

$$T_{g,a} = \frac{T_{g,1} + T_{g,2}}{2} \tag{2}$$

and $T_{g,2}$ is calculated from the following heat balance:

$$W_g\left[C_{p,g}(T_{g,1}-T_{g,2})\right] = W_c\left[H_v + C_{p,c}(T_s-T_i) + C_{p,v}(T_{g,2}-T_s)\right] \tag{3}$$

The correlation reported in reference 5 was obtained with a 4-inch tube in which the cooling film was partly disturbed by instrumentation and the inner surface did not have a smooth finish. The approximate relation obtained at three different coolant flows was

$$\frac{St}{Pr^{-0.6}} \sim (Re)^{0.07} \tag{4}$$

The Prandtl number exponent of -0.6 was assumed from conventional heat-transfer correlations because the Prandtl number did not vary appreciably for the runs.

Figure 5 shows log-log plots of Stanton number divided by Prandtl number raised to the -0.6 power against Reynolds number for results obtained in 2- and 4-inch smooth-surface tubes (the inner surfaces of these tubes were honed). The results shown include data obtained with the following conditions at the tube entrance: Air-stream temperatures from 800° to 2000° F, mass velocities of the air stream from 39.4 to 81.7 pounds per second per square foot, and Reynolds numbers from 220,000 to 1,100,000. Figure 5(a) shows data obtained at a coolant flow of 0.116 pound per second per foot of tube circumference in both 2- and 4-inch-diameter tubes. Data obtained in a 4-inch rough-surface tube (reference 5) are shown for comparison; the data were recalculated from average gas-stream temperatures in the film-cooled tube in place of entrance-air temperatures. The relation obtained with the smooth-surface tubes is approximately the same as that for the rough-surface tube (slopes were about 0.05 to 0.07), but heat-transfer coefficients are about 20 percent less with the smooth-surface tubes. A comparison of data obtained at two different coolant flows in the 2-inch smooth-surface tube is shown in figure 5(b). The relation appears to be the same with each of the coolant flows (approximately 0.07) and heat-transfer coefficients are higher at the high coolant flow; these trends are the same as obtained with the 4-inch rough-surface tube.

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Data scatter for these correlations is in the same order of magnitude as reproducibility of liquid-cooled length with constant test conditions. The greatest amount of scatter occurred between different periods of running and was probably caused by the effect of reassembly of the coolant injector after cleaning of the film-cooled tube as described previously in PROCEDURE.

Generalized Plot of Film-Cooling Data Based on Heat-Transfer Correlation at Constant Coolant Flow per Circumferential Length

A generalized plot of film-cooling data which was based on the heat-transfer correlation and data obtained at different coolant flows was presented in reference 5. Coolant flow was plotted against terms which included gas-stream and coolant variables, tube dimension, and film-cooled area. Investigation has shown (reference 6) that heat-transfer coefficients change with coolant flow per circumferential length of tube because of the occurrence of coolant-film disturbances; a generalized plot with coolant flow per circumferential length has therefore been made.

The equation for the heat-transfer correlation with the smooth-surface tubes (fig. 5(a)) is

$$\frac{h}{C_{p,g}G_g} = a\left(\frac{DG_g}{\mu_g}\right)^{0.05}\left(\frac{C_{p,g}\mu_g}{k_g}\right)^{-0.6} \tag{5}$$

where the proportionality constant a varies with coolant flow per circumferential length. Substituting the equation for heat-transfer coefficient (equation (1)) in equation (5) and rearranging yield:

$$\frac{W_c}{\pi D} \sim (L)(G_g) \frac{c_{p,g}(T_{g,a}-T_s)}{\Delta H} (Re)_g^{0.05} (Pr)_g^{-0.6} \tag{6}$$

This relation is plotted in figure 6 for all data obtained with smooth-surface tubes; coolant flow per circumferential length of tube is plotted against liquid-cooled length and gas and coolant variables. The curve shows the same trends as the curves of figure 4; cooling effectiveness per flow of coolant decreases appreciably in the range of water flow between 0.05 and 0.09 pound per second per foot. Because wave-like disturbances occur on a liquid-cooling film when the coolant flow per circumferential length of tube exceeds a certain value and the result is a decrease in film-cooled area per unit flow of coolant, it may be desirable in many practical applications to limit the coolant flow introduced at any one axial position and to introduce it at several axial positions over the area to be cooled in order to achieve the desired cooling with as little coolant flow as possible.

Figure 6 is convenient for determining the water flow required to maintain liquid-film cooling for a desired distance in a straight tube with fully developed turbulent hot-gas flows to 2000° F entering the tube. Figure 6 is also useful to estimate film-coolant requirements for conditions which have not been investigated experimentally and to act as a guide for correlating data obtained from actual applications.

Further research is required to determine if gas temperatures greater than 2000° F will alter results as predicted from figure 6 besides the effect of increasing heat transfer due to radiation. Coolant requirements obtained from figure 6 account for the heat transferred from the hot gas to the liquid film by forced convection alone and not by radiation to the liquid film or to the tube wall; additional coolant is required where the heat transferred by radiation is appreciable. If the amount of heat transferred to the tube wall by radiation were large enough to result in a wall temperature considerably above the saturation temperature of the coolant, it is possible that additional coolant would be lost because of disturbance of the liquid film caused by the formation of vapor bubbles under the film.

Results predicted from figure 6, when coolants having viscosities and surface tensions considerably different from those of water are used, are doubtful because of the effect of these properties on the flow conditions of the liquid film. Investigations described in reference 6 show that the changing flow conditions of the liquid film with increased values of liquid flow per tube circumferential length (evidenced by the occurrence of turbulent or disturbed flow) result in the variation in cooling effectiveness with increased coolant flow, which can be seen in

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figure 6. The investigation also showed that the values of coolant flow per circumferential length at which disturbed flow begins and the nature of the disturbed flow differ with the viscosity and surface tension of the liquid. Little is known of the extent to which changes in these properties can affect the film-cooling results, but experiments described in reference 5 indicate that for relatively small changes it is not extreme; no appreciable effect on the results was found when ethylene glycol was used as the coolant, and it had a viscosity about $2\frac{1}{2}$ times that of water at liquid-film temperatures and a lower surface tension.

Gas-flow conditions with many applications of film cooling are considerably different than those of the experiments reported herein, and large differences between the film cooling obtained in such cases and that obtained in the experiments are possible. For example, in a rocket engine combustion chamber the effects of the combustion process and the short length of the chamber will not allow conditions approximating fully developed turbulent velocity and temperature distributions as with the experiments.

Other conditions that are met in practical applications of film cooling and which need experimental investigation are the use of film coolants which will burn in the gas stream and changing contour of the flow duct.

An example is given in the appendix to illustrate the use of figure 6 for determining film-coolant requirements for convective heat transfer. The relation plotted is represented quite well by the following empirical equation which may be used for convenience in place of figure 6:

$$X = \frac{0.0013}{\left(\frac{1}{Y}\right) - 0.007}$$

where

$$X = \frac{W_c}{\pi D} \text{ (abscissa of fig. 6)}$$

and

$$Y = (L)(G_g)\left[\frac{c_{p,g}(T_{g,a}-T_s)}{\Delta H}\right](Re)_g^{0.05}(Pr)_g^{-0.6} \text{ (ordinate of fig. 6)}$$

In general, because a film coolant must be carried as added weight in flight, the application of internal-film cooling in flight-propulsion engines appears limited to conditions where propulsive energy is derived from the film coolant in addition to its cooling function and to conditions of high heat release where more conventional cooling methods are inadequate. The use of film cooling in some rocket engines is an example in which these conditions apply.

SUMMARY OF RESULTS

An investigation of liquid-film cooling was conducted in 2- and 4-inch-diameter straight tubes having honed inner surfaces with air flows at temperatures from 800° to 2000° F and diameter Reynolds numbers from 220,000 to 1,100,000. The film coolant was water which was injected at a single axial position on the tube at flow rates from 0.02 to 0.24 pound per second per foot of tube circumference (0.8 to 12 percent of the air flow). Cooling effectiveness was determined by means of wall-temperature measurements. The results of the experiments are summarized as follows:

1. Correlation was obtained for heat transfer from the hot air to the liquid-cooling film with 2- and 4-inch-diameter smooth-surface tubes and was the same as was found in a previous investigation with a 4-inch-diameter rough-surface tube; heat-transfer coefficients were about 20 percent lower with the smooth-surface tubes than with the rough-surface tube.

2. Effectiveness of the coolant was decreased by coolant-film disturbances in a 2-inch-diameter tube as was found in a previous investigation in a 4-inch-diameter tube; the approximate coolant flow above which reduced cooling effectiveness resulted was predicted by a method developed in a previous investigation.

3. All film-cooling data obtained with smooth-surface tubes were generalized by means of heat-transfer correlation and data obtained over a range of coolant flows; this generalization of data facilitates the estimation of the quantity of coolant required to film cool a tube for a desired length, over a wide range of conditions, when the temperature and flow of the hot gas are known.

National Advisory Committee for Aeronautics
Lewis Flight Propulsion Laboratory
Cleveland, Ohio

APPENDIX - EXAMPLE OF USE OF FIGURE 6 TO DETERMINE

FILM-COOLANT REQUIREMENTS

It is desired to determine the coolant flow required to film cool a 4-inch-diameter tube with water supplied at 100° F for a distance of 1 foot with air flowing through the tube at 3 pounds per second and entering at 2000° F and with the pressure in the tube at 300 pounds per square inch absolute. Properties for air were obtained from references 9 and 10, and properties for water and water vapor were obtained from reference 11.

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Because the average temperature $T_{g,a}$ cannot be calculated (equations (2) and (3)) without knowing the coolant flow, a first approximation of the coolant flow is made from figure 6 by calculating the ordinate of figure 6 with the air temperature at the entrance of the tube $T_{g,1}$, which in this case is 2000° F, in place of $T_{g,a}$ and with the physical properties of air at that temperature. The ordinate Y of figure 6 is then calculated from the following:

$$L = 1.0 \text{ ft}$$

$$G_g = \frac{W_g}{A_D} = \frac{3}{0.0873} = 34.4 \text{ lb/(sec)(sq ft)}$$

$$c_{p,g} = 0.283 \text{ Btu/(lb)(}^{\circ}\text{F)}$$

$$T_{g,1} - T_s = 2000 - 417 = 1583^{\circ} \text{ F}$$

$$\Delta H = c_{p,c}(T_s - T_1) + H_v = 1.01(417-100) + 809 = 1129 \text{ Btu/lb}$$

$$(Re)_g^{0.05} = \left(\frac{DG_g}{\mu_g}\right)^{0.05} = \left(\frac{34.4}{3 \times 34.5 \times 10^{-6}}\right)^{0.05} = 1.89$$

$$(Pr)_g^{-0.6} = (0.627)^{-0.6} = 1.33$$

$$Y = (1.0)(34.4)\left(\frac{0.283\ (1583)}{1129}\right)(1.89)(1.33) = 34.3$$

From figure 6, $W_c/\pi D = 0.057$ pound per second per foot and $W_c = 0.057 \times \pi/3 = 0.059$.

From equation (3) and with the use of this coolant flow $T_{g,2}$ is calculated. Successive approximations are sometimes necessary because specific-heat values are averages for the temperature gradients. For the first approximation, the value for $c_{p,g}$ is taken at $T_{g,1}$ and the value for $c_{p,v}$ is taken at an average between $T_{g,1}$ and T_s:

$$W_g\left[c_{p,g}(T_{g,1}-T_{g,2})\right] = W_c\left[H_v + c_{p,c}(T_s-T_i) + c_{p,v}(T_{g,2}-T_s)\right]$$

$$3\left[0.283(2000-T_{g,2})\right] = 0.059\left[809+1.01(417-100)+0.53(T_{g,2}-417)\right]$$

$$T_{g,2} = 1868^\circ \text{ F}$$

Further determinations will not appreciably affect the value of $T_{g,2}$ in this case because the changes in specific heats are small. $T_{g,a}$ is then calculated from equation (2)

$$T_{g,a} = \frac{2000+1868}{2} = 1934^\circ \text{ F}$$

The ordinate of figure 6 is then calculated using this value for average gas temperature $T_{g,a}$ and using the physical properties of air at this temperature

$$Y = (1.0)(34.4)\left(\frac{0.282(1517)}{1129}\right)(1.89)(1.33) = 32.8$$

From figure 6, $W_c/\pi D = 0.054$ pound per second per foot and $W_c = 0.054\times\pi/3 = 0.057$ pound per second. Further calculations do not improve the determination appreciably; so a coolant flow of 0.057 pound per second is the requirement.

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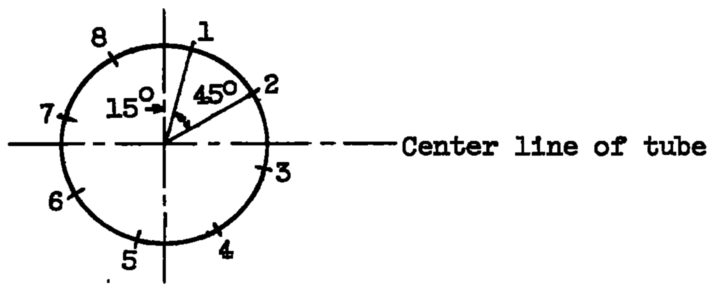
TABLE I - LOCATION OF THERMOCOUPLES ON 4-INCH-DIAMETER FILM-COOLED TUBE



End view of tube in upstream direction showing circumferential positions of thermocouples

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

[a]Thermocouples located at 1-in. intervals in this range.

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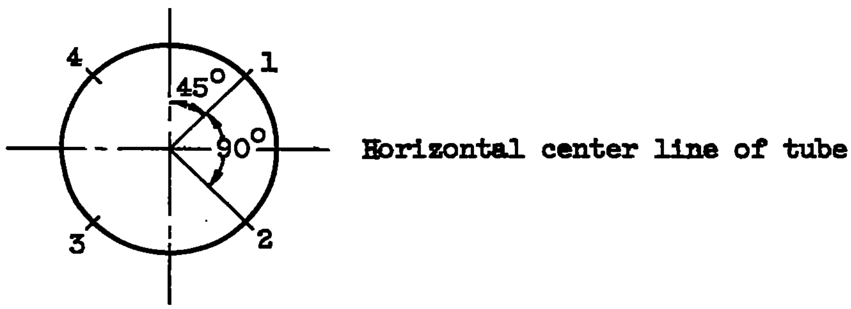
TABLE II - LOCATION OF THERMOCOUPLES ON 2-INCH-DIAMETER FILM-COOLED TUBE



End view of tube in downstream direction showing circumferential positions of thermocouples

Circumferential position	Location of thermocouples - distance downstream of coolant injection (in.)
1	3, 5 to 42[a]
2	5, 9 to 41[b]
3	5, 9 to 41[b]
4	5, 9 to 41[b]

[a]Thermocouples located at 1-in. intervals in this range.

[b]Thermocouples located at 2-in. intervals in this range.

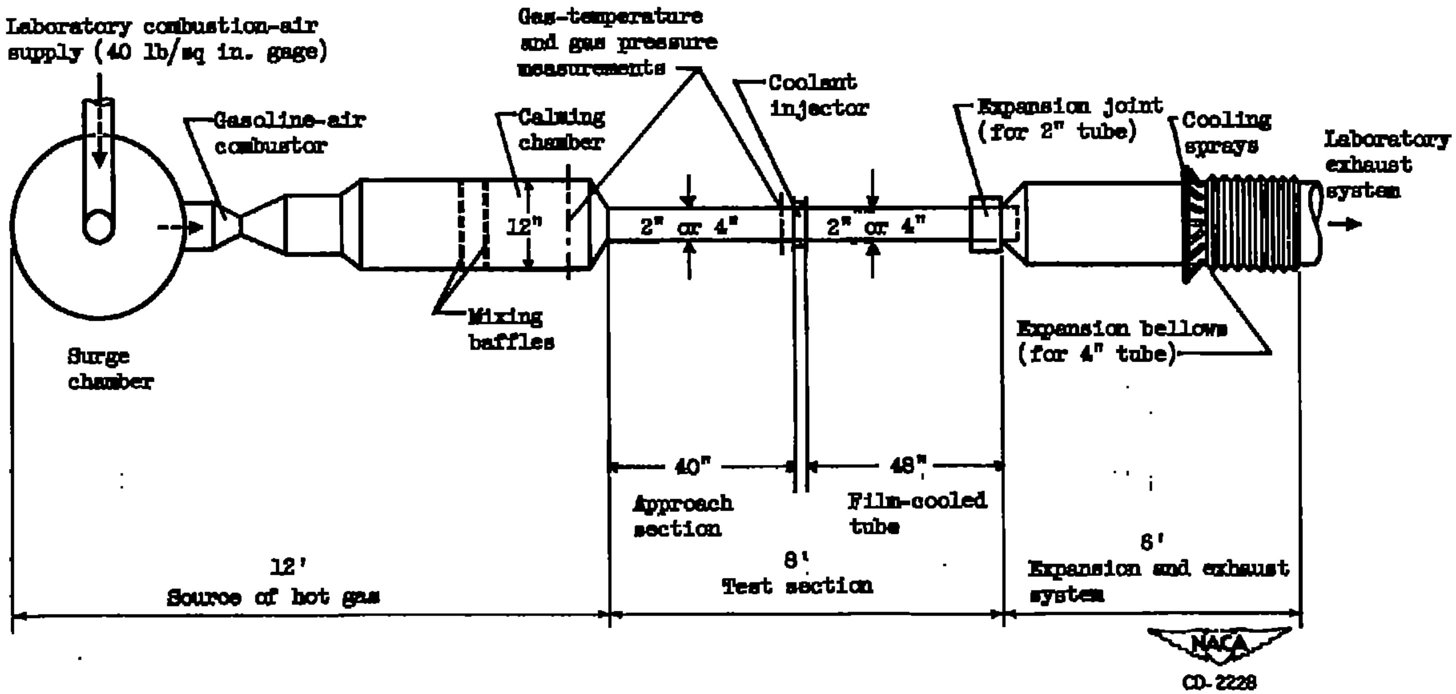


Figure 1. - Flow system.

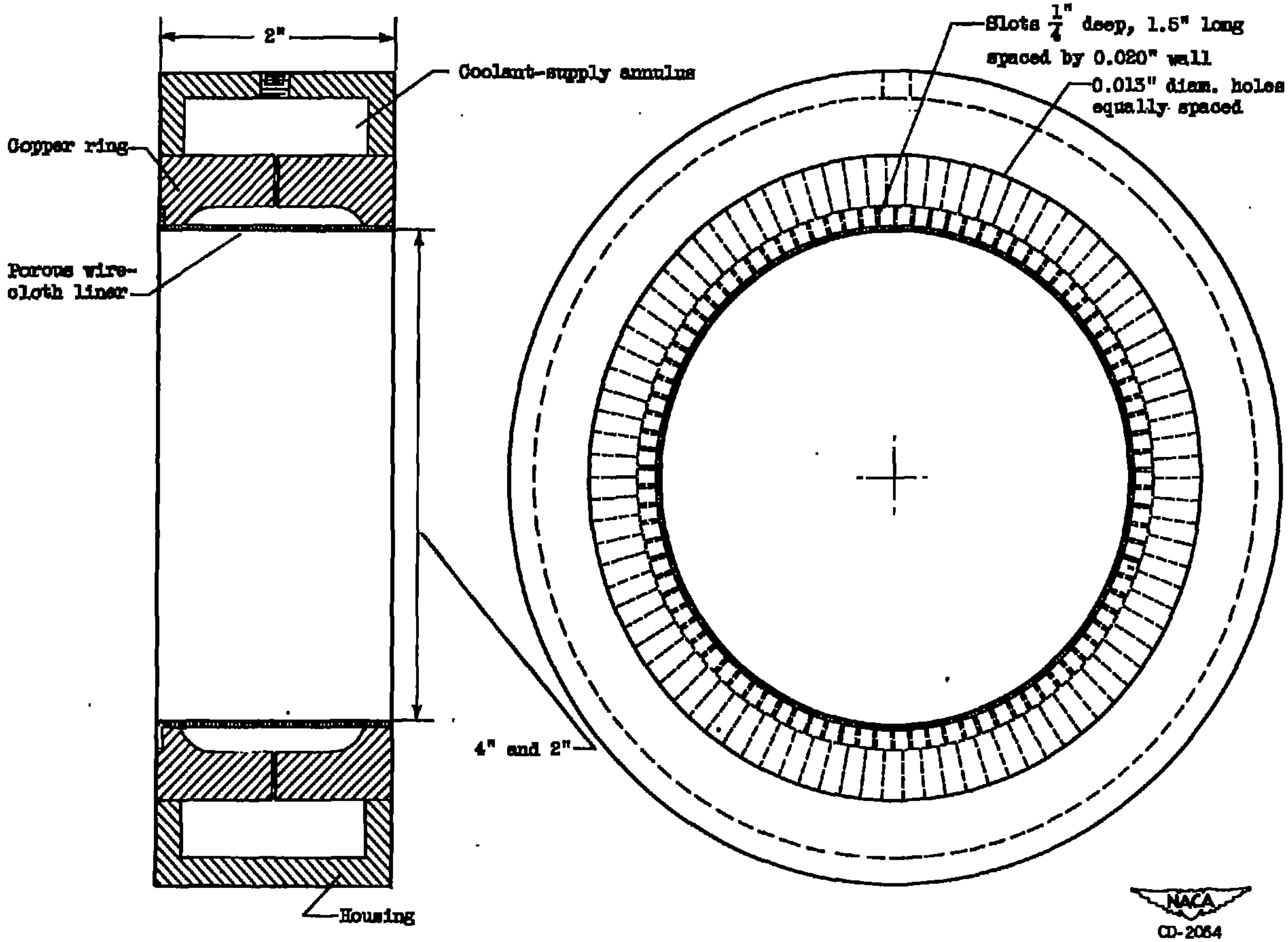


Figure 2. - Coolant injectors.

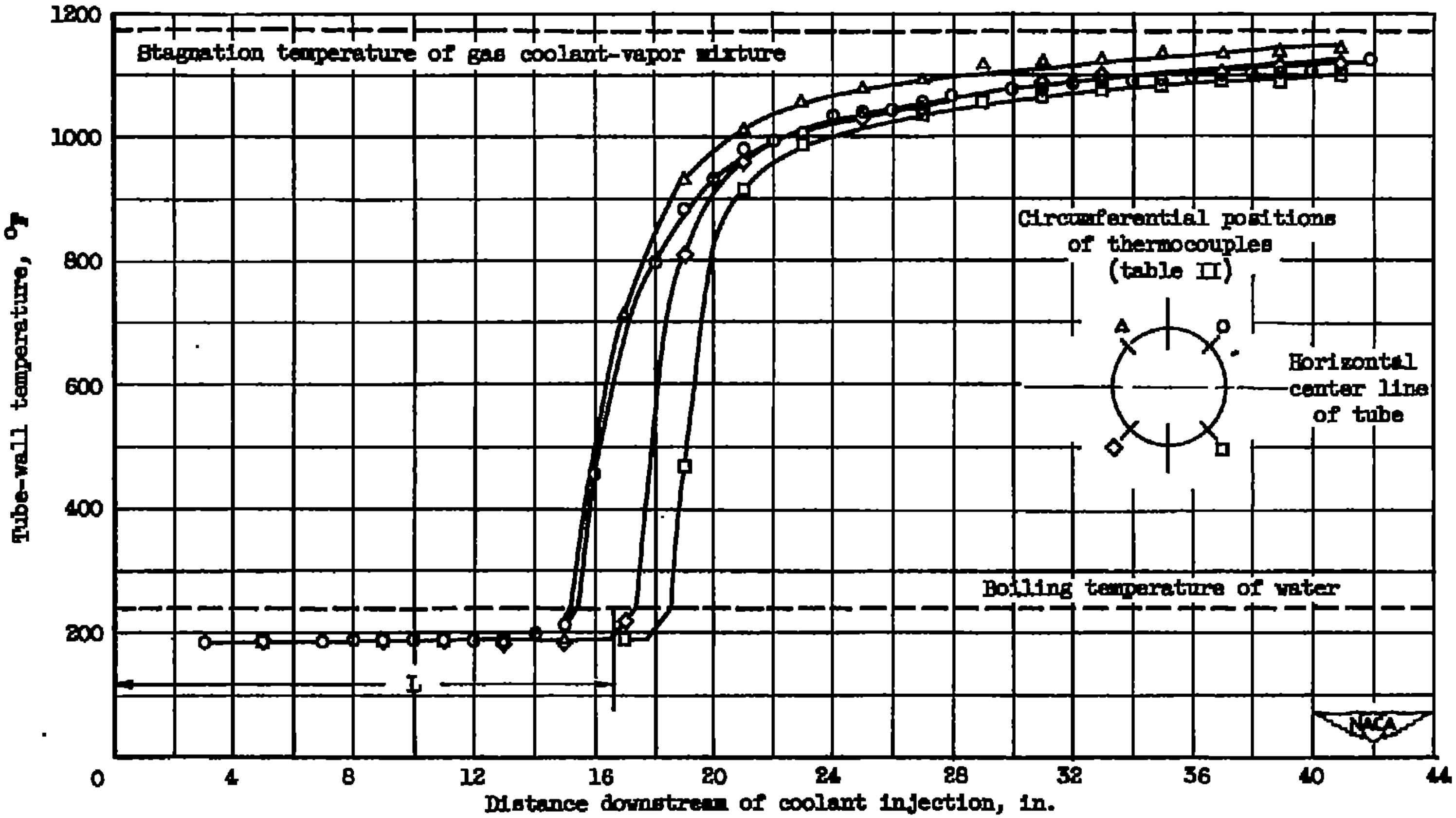


Figure 3. - Liquid-film-cooled length L in 2-inch-diameter smooth-surface tube. Gas temperature, 1600° F; gas mass velocity, 46.8 pounds per second per square foot; coolant flow per circumferential length, 0.080 pound per second per foot; ratio of coolant flow to gas flow, 4.1 percent.

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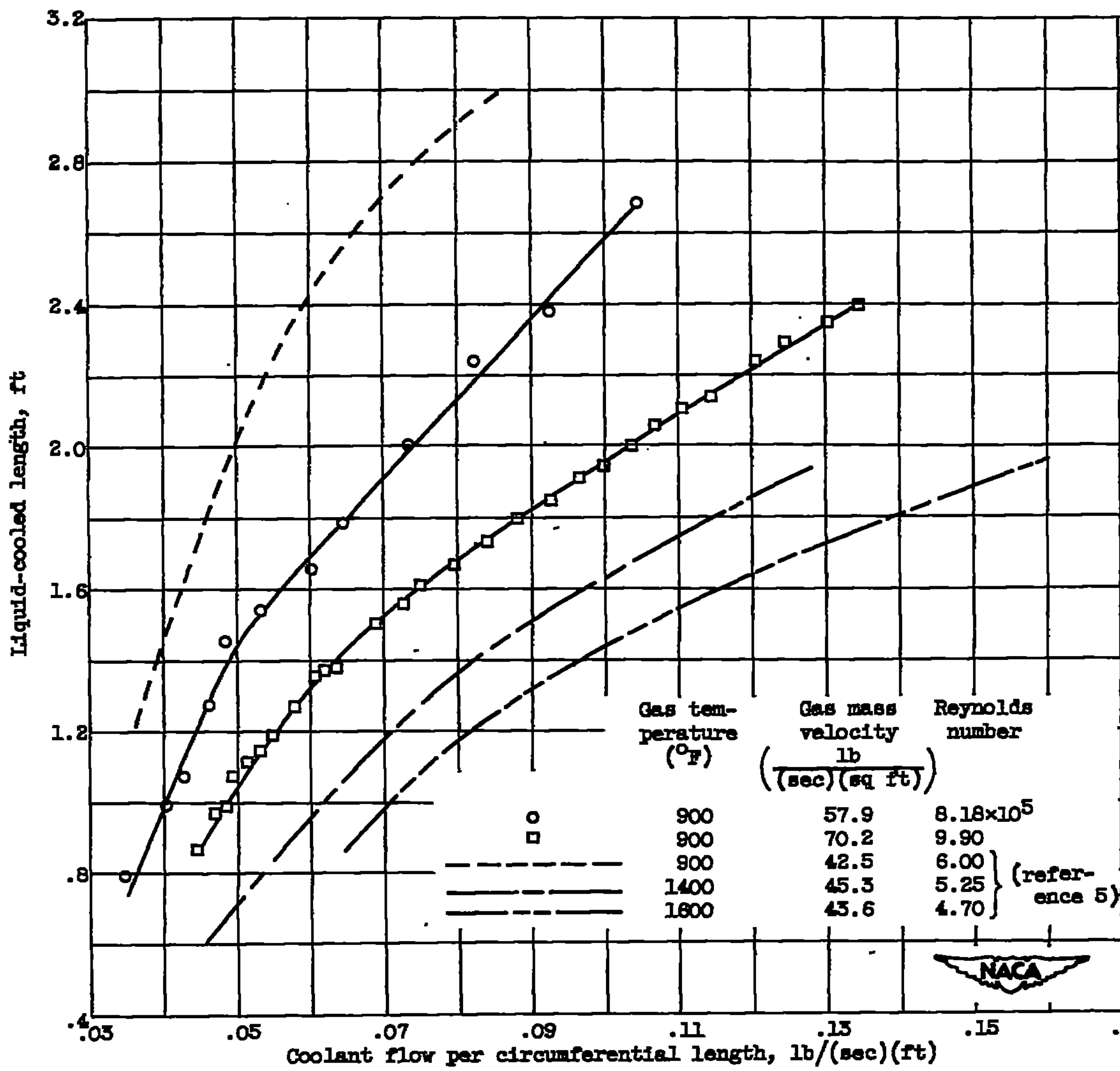
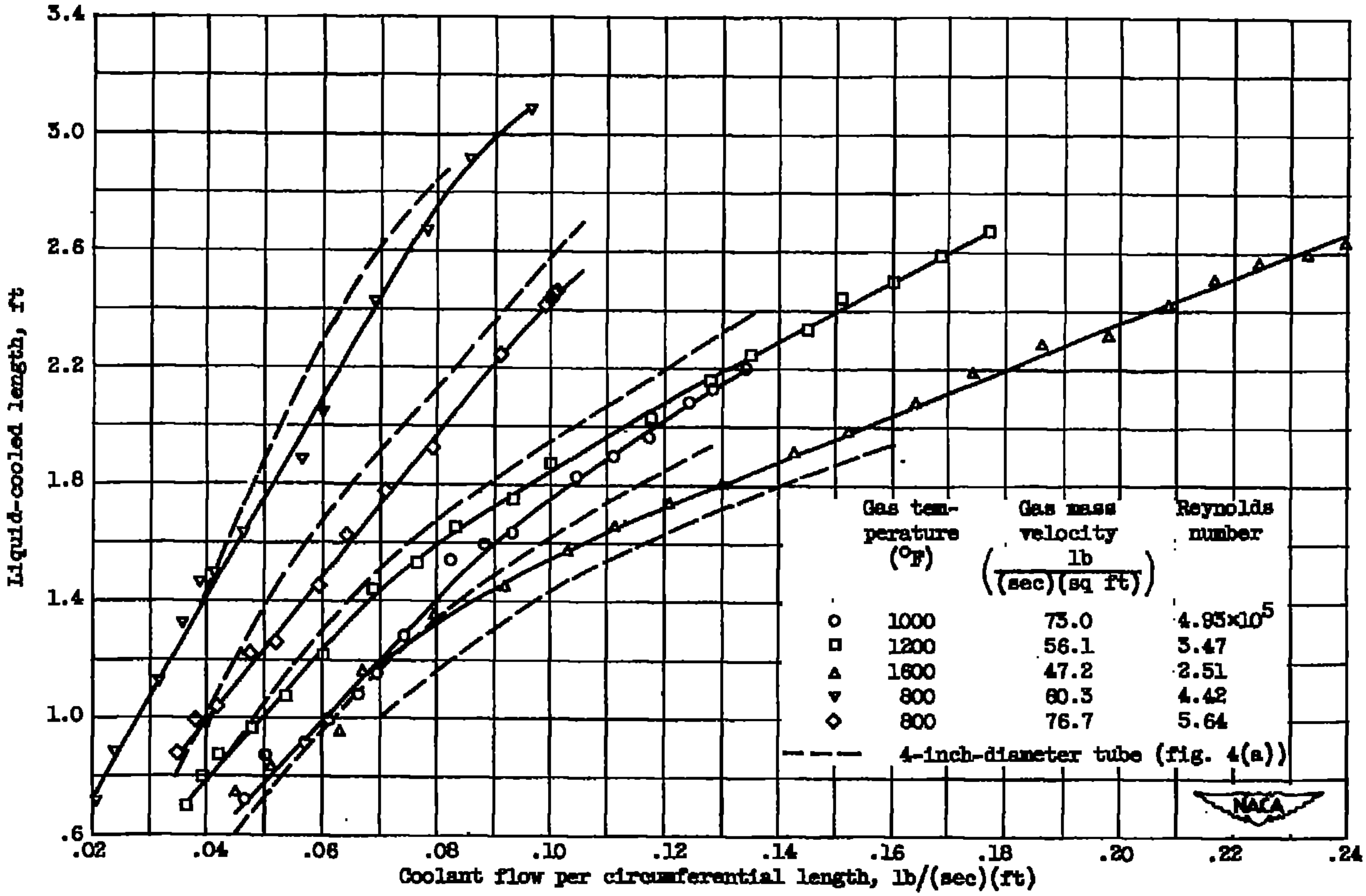


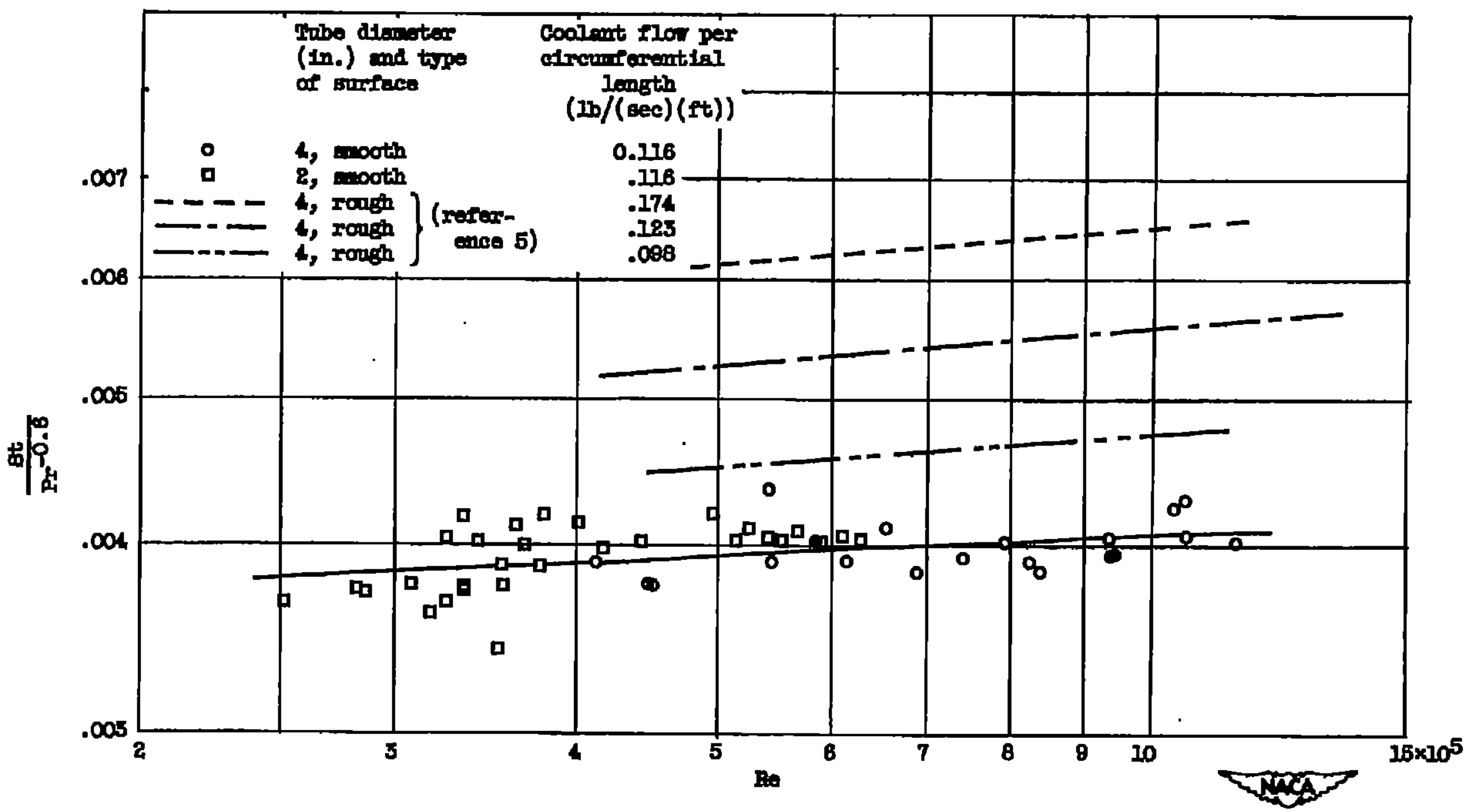
(a) 4-inch-diameter smooth-surface tube.

Figure 4. – Variation of liquid-cooled length with coolant flow per circumferential length.



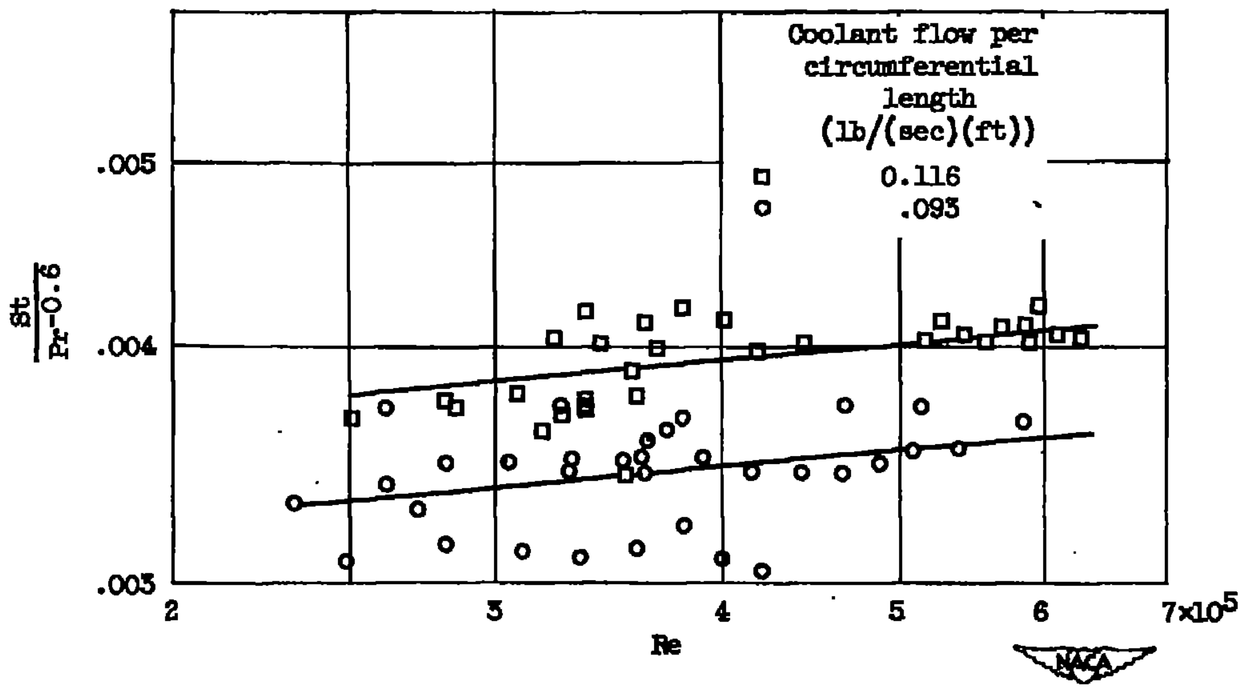
(b) 2-inch-diameter smooth-surface tube.

Figure 4. - Concluded. Variation of liquid-cooled length with coolant flow per circumferential length.



(a) 2- and 4-inch-diameter smooth-surface and 4-inch-diameter rough-surface tubes.

Figure 5. - Correlation of heat transfer from hot gas to liquid film at constant coolant flow per circumferential length. Gas temperatures, 800° to 2000° F; gas mass velocities, 39.4 to 81.7 pounds per second per square foot.



(b) 2-inch-diameter smooth-surface tube.

Figure 5. - Concluded. Correlation of heat transfer from hot gas to liquid film at constant coolant flow per circumferential length. Gas temperatures, 800° to 2000° F; gas mass velocities, 39.4 to 81.7 pounds per second per square foot.

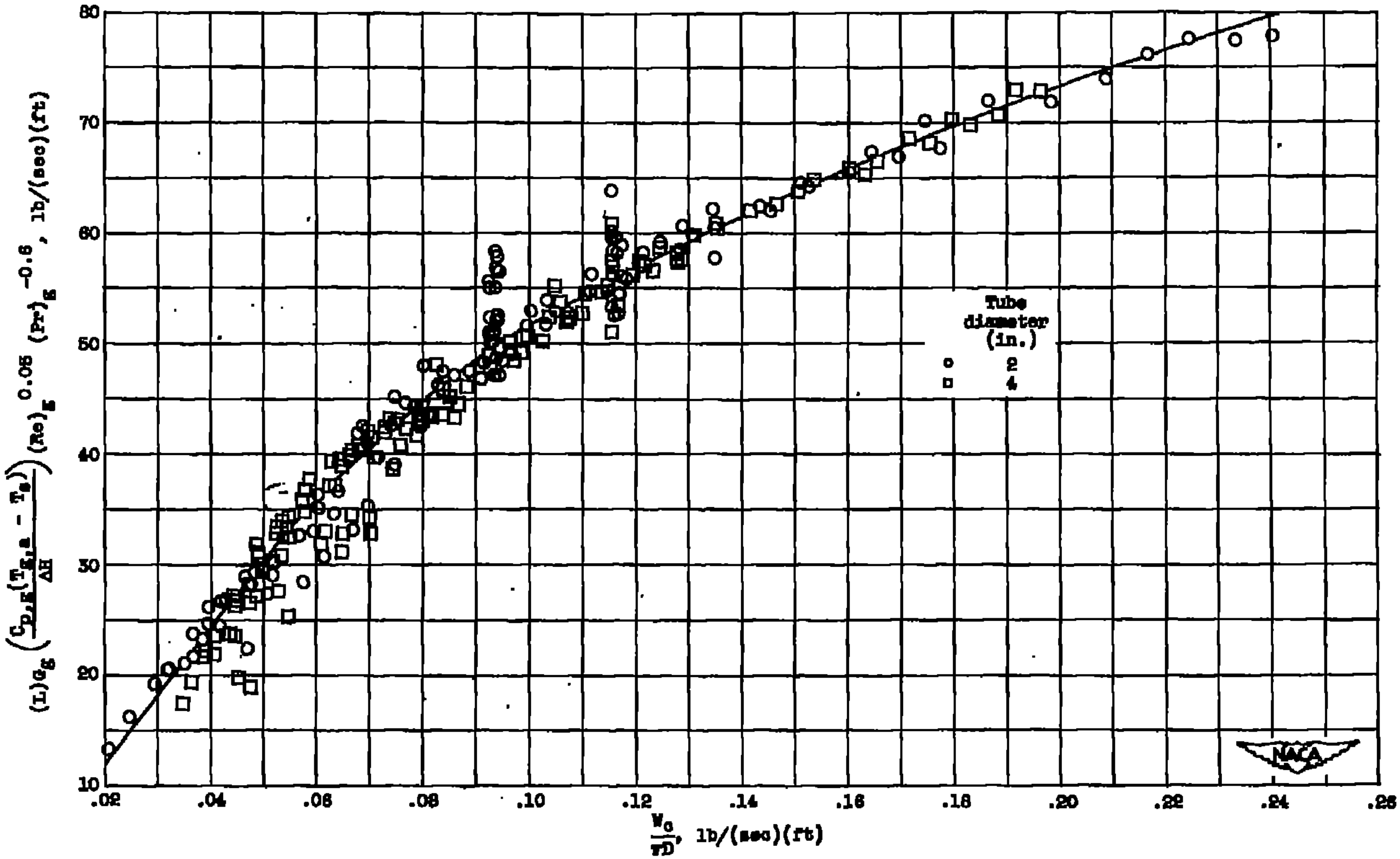


Figure 6. - Generalized plot of film-cooling data in smooth-surface tubes based on correlation of heat transfer from hot gas to liquid-cooling film. Coolant, water; gas mass velocities, 59.4 to 81.7 pounds per second per square foot; gas-stream Reynolds numbers, 2 to 11×10^5.

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