SECURITY INFORMATION

[redacted]

Copy 304
RM E52G01

[redacted]

NACA

RESEARCH MEMORANDUM

AN ANALYTICAL METHOD FOR EVALUATING FACTORS AFFECTING APPLICATION OF TRANSPIRATION COOLING TO GAS TURBINE BLADES

By Jack B. Esgar

Lewis Flight Propulsion Laboratory
Cleveland, Ohio

[illegible] changed to Unclassified
by [illegible] NasaTech Pub Announcement #119
[illegible] TO CHANGE)

By.......... 29 Aug. 57

NK
GRADE OF OFFIC[illegible] CHANGE)

30 Mar 61
DATE

[redacted]

[redacted]

NATIONAL ADVISORY COMMITTEE
FOR AERONAUTICS

WASHINGTON
September 8, 1952

[redacted]

TABLE OF CONTENTS

APPENDIXES

NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS

RESEARCH MEMORANDUM

AN ANALYTICAL METHOD FOR EVALUATING FACTORS AFFECTING APPLICATION OF TRANSPIRATION COOLING TO GAS TURBINE BLADES

2597

By Jack B. Esgar

SUMMARY

An analytical investigation was conducted to provide a method for determining the cooling-air requirements for transpiration-cooled turbine stator and rotor blades for practically all engine and flight operating conditions, and the cooling-air requirements were evaluated for typical cases to determine trends that would be encountered for a wide range of variables. The cooling-air requirements and blade permeabilities are affected by the gas temperature level, the pressure level, and the pressure distribution around the turbine blades. The temperature and the pressures, in turn, vary with engine and flight operating conditions. Since the permeabilities around the blade periphery can be specified for only one set of design conditions, at off-design conditions some parts of the blade may have to be overcooled in order that other parts of the blade will be cooled adequately, with the result that an excessive amount of cooling air will be required at off-design conditions. By proper design procedures, however, the excess of cooling air required at off-design conditions can be minimized. For example, with a turbine-inlet temperature of 1600° F, typical turbine stator and rotor blades designed to cool properly at sea level would require a combined coolant flow ratio of 0.136 at an altitude of 50,000 feet as compared with the ideal coolant flow ratio of 0.039 (resulting in an excess of about 250 percent) for cooling the blades to a maximum temperature of 1000° F. On the other hand, if the blade had been designed to cool properly at an altitude of 50,000 feet, the maximum excess of cooling air required at any other altitude would be about 60 percent. Multiple orifices at the blade base to partly meter the cooling air to local chordwise positions on the blade periphery would reduce the maximum excess in required coolant flow to less than 10 percent. In addition to the improved cooling performance possible with this type of blade design, orifices at the blade base also permit the fabrication of transpiration-cooled blades that have a uniform permeability around the periphery, which will greatly simplify fabrication. The permeability range required for all types of transpiration-cooled gas-turbine blade can be obtained from single thicknesses of brazed and calendered stainless-steel wire cloth.

The trends shown in this report can probably be generalized for most engines, but because of the sensitivity of the coolant flow requirements to blade design, any engine design that will use transpiration-cooled turbine blades must have blades designed with consideration given to both the engine conditions and the type of flight plan to which it is anticipated the engine will be subjected.

INTRODUCTION

Transpiration cooling is the most effective air-cooling method known at present for surfaces that must be in contact with high-temperature gas streams; consequently, research is being conducted to find methods of applying this cooling process to the stator and rotor blades of gas-turbine engines and to determine the effects of engine and flight operating conditions on coolant requirements. A survey of some of the advantages and problems associated with transpiration cooling of gas-turbine engines is given in reference 1, and it is shown therein that high pressure gradients around the periphery of gas-turbine blades require that the blade wall permeability be varied around the blade periphery in order for uniform cooling to be obtained over the entire blade surface. This fact is verified in experimental investigations of transpiration-cooled turbine blades mounted in a static cascade (references 2 and 3) where it is shown that although transpiration cooling results in extremely effective cooling in the midchord region of the blade, there are very large variations in the chordwise temperature distribution because of improper permeability variation. Reference 2 also shows that the temperature distribution changes drastically as the cooling flow rate is varied, particularly in the midchord region on opposite sides of the blade. The reason for this great change in temperature distribution is a change in the relative amounts of coolant metered through the opposite surfaces of the blade as the cooling supply pressure is varied. For porous walls the flow from the inner surface through the wall to the outer surface is a function of the difference of the squares of the absolute pressure levels on opposite sides of the wall, so that small changes in pressure level can often result in very large changes in coolant flow rate.

Gas turbines for aircraft application must operate over a wide range of engine speed, flight speed, and altitude. The effect of each of these variables will be a variation in the pressure levels, and consequently the pressure distributions, around the outside of the turbine blades. Since the permeability of transpiration-cooled blades must be set to provide the proper cooling at some specified pressure distribution, changes in altitude, flight speed, or engine speed can often cause the specified permeability to become inconsistent with the aerodynamic pressure distribution, with the result that the blade will not be at a uniform temperature and excess coolant flow may be required for some portions of the blade in order to obtain adequate cooling in the hottest portions.

A possible method of improving the off-design coolant flow requirements in transpiration-cooled turbine blades is to provide multiple orifices at the blade base to partly meter the cooling air. Each orifice would be used to meter air to a particular location on the blade surface. The advantage to be gained by use of orifices comes from the fact that the laws governing flow through orifices are different from those governing flow through a porous wall, and they are more consistent with the requirements for transpiration cooling gas-turbine blades over a range of pressure levels. The use of orifices for partly metering the cooling air also greatly simplifies the blade fabrication. Instead of requiring built-in variable wall permeabilities around the blade periphery which will provide the proper coolant distribution, the orifices permit the use of uniform or other specified permeability variations that simplify manufacture. The proper metering is then accomplished by means of the orifices.

As shown in figure 5 of reference 4, transpiration cooling is most valuable at very high gas temperature levels because at those temperature conditions, no other known method of air cooling is practical. For very high temperature engines, it will probably be desirable to cruise at a lower gas temperature level to obtain improved fuel economy. This type of engine introduces a problem because the higher the gas temperature level, the more coolant will be required for adequate cooling and the more permeable the blade wall must be for a specified pressure difference across the blade wall. When the turbine-inlet temperature is reduced while the engine is at cruise conditions, however, excessive quantities of coolant will probably be metered through the permeable walls of transpiration-cooled turbine blades.

An analytical method developed at the NACA Lewis laboratory to determine the magnitude of coolant flow variations for transpiration-cooled turbine stator and rotor blades as engine and flight operating conditions are changed is presented herein. The analysis is based on the turbulent flow equation of Friedman (reference 5) and the flow variations with pressure and temperature that are obtained using typical porous materials. In order to illustrate the typical trends obtained for changes in flight altitude, flight speed, engine compressor pressure ratio, and turbine-inlet temperature, results of the analysis are shown for typical turbine stator and rotor blades for flight altitudes from sea level to 50,000 feet, flight Mach numbers from 0 to 1.0, engine compressor pressure ratios of 4 and 10, and turbine-inlet temperatures of 1600° and 2500° F. In addition, methods of design that would minimize coolant flow requirements are considered. For all calculations, the coolant supply pressure was allowed to vary with engine or flight conditions to provide the pressure required for adequate cooling, and for most calculations the blade wall permeability varied around the periphery and the coolant temperature was dependent upon the compression required for the cooling air.

CALCULATION PROCEDURE

In a determination of the effect of altitude and flight conditions on transpiration cooling of gas turbine blades, the following quantities must be determined:

(1) The temperature and pressure levels throughout the engine including the local static gas temperature and pressure distributions around the stator and rotor blades for typical stator and rotor blade 2597 configurations.

(2) The ideal amount of cooling air required, which is defined as the amount of air required to cool the blade to an assigned uniform temperature. These calculations do not require a knowledge of the blade permeability. The ideal air requirements are determined by the local Reynolds number on the outside blade surface, the gas temperature level, and the cooling-air temperature level.

(3) The blade permeability necessary to meter the ideal amount of cooling air for a given supply pressure and for a given pressure distribution on the outside surfaces of the blade.

(4) The off-design blade temperatures, coolant supply pressures, and coolant flow requirements. The blade permeability distribution required to meter the ideal amount of cooling air required for a given set of engine and flight operating conditions usually will not be the proper permeability distribution for any other set of engine and flight operating conditions. Therefore, for off-design conditions, the blade temperature distribution will no longer be uniform, higher coolant supply pressures will probably be required, and excess cooling air will be required. The quantity of cooling air can be evaluated in terms of the gas flow through the engine from calculations based on the blade surface and passage areas, the gas temperature, and the velocity and pressure distributions through the blade passages.

The equations used for these various calculations are set forth in the following paragraphs. The stations through the engine, indicated by the subscripts o through 5, are shown on figure 1. The symbols are defined in appendix A.

Temperature and Pressure Levels through Engine

The calculation of the temperature and pressure levels through the engine is accomplished by standard engine cycle analysis methods. The equations and the explanation of their use are given in appendix B.

Ideal Cooling-Air Requirements

The local ideal coolant flow rates necessary to obtain uniform cooling of the blade surface can be calculated using the turbulent flow equation of Friedman (reference 5). The modification of this equation for use in this report is given in appendix C and the final equation becomes

2597

$$\rho_a v_a = \frac{\ln\left(1 - r + \dfrac{r}{\dfrac{T_{B,g}-T_a}{T_{g,e}-T_a}}\right)}{71.3(Re_{B,g})^{0.1}(Pr_g)^{2/3}}\left(\frac{\rho_{g,B}V_{g,B}T_{g,B}}{T_{B,g}}\right) \tag{1}$$

where $\rho_a v_a$, $\rho_{g,B}$, $V_{g,B}$, $T_{B,g}$, $T_{g,B}$, and $T_{g,e}$ are local values of flow density, velocity, and temperature. The cooling-air supply temperature T_a is related to the supply pressure by equation (B5) unless an aftercooler is used.

The local cooling-air flow rate per unit area $\rho_a v_a$ can be obtained from equation (1) and from the values of $T_{B,g}$ (an assigned value), $\rho_{g,B}$, $V_{g,B}$, $T_{g,e}$, and $T_{g,B}$, which can be calculated as explained in appendix B.

The coolant flow ratio, which is defined as the ratio of cooling-air flow to engine gas flow, can be calculated from

$$\frac{w_a}{w_g} = \frac{\sum \rho_a v_a A_a}{\rho_{g,m}V_{g,m}A_{g,m}} \tag{2}$$

where

$\rho_a v_a$ obtained from equation (1)

$\rho_{g,m}V_{g,m} = p_{g,m}M_{g,m}\sqrt{\dfrac{\gamma_{g,m}g}{RT_{g,m}}}$

A_a local blade surface area from which local cooling-air flow $\rho_a v_a$ is flowing

$p_{g,m}$, $T_{g,m}$, $M_{g,m}$ mean values of static pressure, static temperature, and Mach number, respectively, in flow area $A_{g,m}$ at any chordwise position of passage between adjacent blades. Values are calculated from stream filament theory (references 6 and 7) and by use of equations similar to (B9), (B8), and (B7) in appendix B.

Whether the air is metered through only the blade wall or is metered partly through orifices and partly through the blade wall, the ideal coolant flow ratios are the same; therefore this method of calculating the ideal coolant flows is applicable to all methods of transpiration cooling. The equations used, however, are for turbulent boundary-layer flow, and at the blade leading edge, where the boundary-layer flow is probably laminar, it would be more exact to calculate the coolant flow rates by the method given in reference 8. Because the laminar boundary layer extends over only a small portion of the blade, the over-all coolant flow rates will be affected very little if the flow is considered to be turbulent over the entire blade. For this reason, the effects of laminar flow will be neglected in this report. 2597

Blade Permeability Calculations

Blades without metering orifices. - After determination is made of the coolant flow required to cool a blade to some specified temperature by use of equation (1), it is possible to determine the blade permeability required to meter that amount of flow for given pressure levels on the inside and the outside of the blade. Unless a device is used to meter the air to various portions of the blade coolant passage, the coolant supply pressure on the inside of the blade will be constant around the blade periphery. On the outside of the blade the local static pressure varies around the blade periphery because of variations in the gas velocity over the blade surfaces. The velocity and pressure distributions over the outside of the blade surface can be obtained as explained previously. It can be seen, then, that because of the constant coolant supply pressure and the varying pressure on the outside of the blade surface, it is necessary that the permeability vary around the blade surface in order to meter the coolant properly.

The relation used for correlating flow through a porous material is given in reference 3 as

$$\frac{\rho_a v_a}{\mu_{a,B}} = \left[\sqrt{\left(\frac{\alpha g}{2\beta}\right)^2 + \frac{g}{2\beta R}\,\frac{p_a^{\,2} - p_{g,B}^{\,2}}{\mu_{a,B}^{\,2}\,\tau T_{a,B}}}\right] - \frac{\alpha g}{2\beta} \qquad (3)$$

where $T_{a,B}$ is the cooling-air temperature inside the blade wall and is assumed to be the same as the blade wall temperature $T_{B,g}$, as discussed in reference 1.

Since α, g, β, and R are constants, $\frac{\rho_a v_a}{\mu_{a,B}}$ is a function of $\frac{p_a^{\,2} - p_{g,B}^{\,2}}{\mu_{a,B}^{\,2}\,\tau T_{a,B}}$. In order to correlate the permeability of a material at a standard temperature, $\frac{\mu_0}{\mu_{a,B}}(\rho_a V_a)$ can be plotted against $\frac{\mu_0{}^2 T_0}{\mu_{a,B}{}^2 T_{a,B}}\left(\frac{p_a{}^2 - p_{g,B}{}^2}{\tau}\right)$. Values of $\frac{\mu_0}{\mu_{a,B}}$ and $\frac{\mu_0{}^2 T_0}{\mu_{a,B}{}^2 T_{a,B}}$ are tabulated against $T_{a,B}$ in table I of reference 9. (In reference 9, the subscript a,B is omitted from μ and T.)

The permeability of a material can be varied by changing (1) the
2597 wall thickness τ, (2) the material density, or (3) the fabrication procedure. Because the fabrication procedure has a significant effect on permeability, the permeability correlations must be established experimentally for each method of fabrication. The effect of wall thickness for a given permeability is correlated in the expression $\frac{\mu_0{}^2 T_0}{\mu_{a,B}{}^2 T_{a,B}}\left(\frac{p_a{}^2 - p_{g,B}{}^2}{\tau}\right)$. For a coolant temperature T_a, stream conditions that result in a Reynolds number $Re_{B,g}$, and the blade in a gas stream having an effective gas temperature $T_{g,e}$, it is first necessary to calculate the coolant flow rate in order to cool a blade to some temperature $T_{B,g}$. From this flow rate and the pressures on the inside and the outside of the blade, the desired blade permeability can be obtained in the following manner:

(1) Calculate $\rho_a V_a$ from equation (1) using $\rho_{g,B}$ from equation (B10) or (B23), $V_{g,B}$ from equation (B11) or (B24), $T_{g,B}$ from equation (B8) or (B21), and the desired blade temperature $T_{B,g}$.

(2) Calculate $\frac{\mu_0}{\mu_{a,B}}(\rho_a V_a)$ from step (1) and the cooling-air viscosity ratio $\frac{\mu_0}{\mu_{a,B}}$.

(3) Using $\frac{\mu_0}{\mu_{a,B}}(\rho_a V_a)$ and figure 2 or a similar curve for another material, read the value of $\frac{\mu_0{}^2 T_0}{\mu_{a,B}{}^2 T_{a,B}}\left(\frac{p_a{}^2 - p_{g,B}{}^2}{\tau}\right)$.

(4) Calculate τ (τ is a method of specifying permeability for a material of a specified density and fabricated by a specified process) from step (3), the specified (or available) coolant supply pressure p_a, the local pressure on the outside of the blade $p_{g,B}$ from equation (B9) or (B22), and the ratio $\frac{\mu_0{}^2 T_0}{\mu_{a,B}{}^2 T_{a,B}}$ at temperature $T_{a,B}$ from table I of reference 9 ($T_{a,B}$ is equal to the blade temperature $T_{B,g}$).

The method outlined in step (4) was the method used in this analysis. The permeability curve (fig. 2) was obtained from reference 10 for 20×250 mesh stainless-steel wire cloth that was brazed and rolled to 37-percent thickness reduction. A complete family of permeability curves can be obtained by rolling to different thicknesses (which changes the material density) or by using different wire meshes. For porous sintered materials, families of curves can be obtained by compacting to various densities or by various fabrication procedures. An alternative method of determining the desired blade wall permeability would therefore be to choose some wall thickness τ and then to find the material with the proper density, or with the proper fabrication procedure, from the family of curves. In this case interpolation would probably be required to find some intermediate density or fabrication procedure other than that for which data are available.

2597

If a material has some specified permeability, the process reverse to the procedure just outlined can be used to determine the coolant flow rate for specified pressures or else to determine the supply pressure p_a to obtain some specified coolant flow rate.

A method frequently used for specifying permeability for porous materials is use of the permeability coefficient K which, based on Darcy's law, is given by

$$\frac{p_a^2 - p_{g,B}^2}{\tau} = \frac{1}{K}\left[2RT_{a,B}\,\mu_{a,B}\,(\rho_a v_a)\right] \tag{4}$$

where $K = 1/\alpha$ for $\rho_a v_a = 0$. The permeability coefficient K specifies the permeability for only one material thickness τ. A somewhat more general method of specifying permeability is to combine the permeability coefficient and the material thickness terms in a manner that actually indicates the quantity of flow through a porous material for given pressure levels on opposite sides of the wall. In addition, the value of K in equation (4) is specified for zero flow, where it was necessary to extrapolate experimental data and to determine the slope of a line at zero flow. Since this method is subject to error because measurement of very low air flows is difficult and may be inaccurate because of the necessity for extrapolation, the permeability values given herein as

$$\frac{K'}{\tau} = \frac{2RT_{a,B}\,\mu_{a,B}\,(\rho_a v_a)}{p_a^2 - p_{g,B}^2} \tag{5}$$

are for conditions that are more easily obtained, namely, for a pressure drop through the material of 10 pounds per square inch discharging to NACA standard sea-level conditions. Because of the variation of K'/τ with air flow, the values specified are not valid for any other set of pressure and temperature conditions.

2597

The method of calculating K'/τ is as follows:

(1) Using the value of τ as explained previously and using the standard conditions specified for the K'/τ calculations, calculate

$$\frac{\mu_0^2 T_0}{\mu_{a,B}^2 T_{a,B}}\left(\frac{p_a^2 - p_{g,B}^2}{\tau}\right) = (1)\,\frac{(3556)^2 - (2116)^2}{\tau}$$

(2) Read the corresponding value of $\frac{\mu_0}{\mu_{a,B}}\rho_a v_a$ from figure 2.

(3) For standard conditions, $\frac{\mu_0}{\mu_{a,B}}$ equals 1 and the use of other standard values of temperature, pressures, and viscosity in equation (5) gives

$$\frac{K'}{\tau} = \frac{(2)(53.3)(518.4)(3.77\times10^{-7})(\rho_a v_a)}{(3556)^2 - (2116)^2}$$

$$= 2.55\ (\rho_a v_a) \times 10^{-9}\ \text{feet}$$

Blades with metering orifices at base designed for specified pressure drop through wall. - For some applications it may be advantageous to meter the cooling air to local positions in the coolant passage around the blade periphery. In this way the local coolant supply pressure can be controlled so that the pressure drops across the blade wall are more uniform. A method of accomplishing this metering is to divide the coolant passage into sections and then to provide an orifice of such size as to meter the air to give the desired pressure drop across the orifice, as shown in figure 3.

If the orifice is in the form of a flow nozzle, the coolant flow through the orifice can be expressed as

$$\rho_a v_a = C_B \sqrt{\frac{h_B(p_a - h_B)}{T_a}} \qquad (6)$$

where

p_a coolant supply pressure ahead of orifice

$p_a - h_B$ local coolant supply pressure inside turbine blade

h_B pressure drop across orifice supplying air to specified position on blade periphery

C_B constant for each orifice $= \frac{B\ A_n}{A_a}\sqrt{\frac{2g}{R}}$

A_a local blade surface area being supplied by coolant through orifice

A flow nozzle was chosen as the form of metering orifice for use at the blade base because the influence of Reynolds number on the flow rate is negligible. Equation (6) is almost exact for Mach numbers in the throat of the flow nozzle up to 0.5, and the error is only 3 percent for a throat Mach number of 1.0. For supercritical pressure drops across the flow nozzle $(h_B \geq 0.472p_a)$, the flow equation can be written in the exact form (for $\gamma = 1.4$) 2597

$$\rho_a v_a = \frac{0.484\ C_B\ p_a}{\sqrt{T_a}} \tag{6a}$$

The constant C_B can be determined so that the pressure loss through the blade wall $(p_a - h_B) - p_{g,B}$ will be some specified amount. In order to calculate the blade permeability for this case exactly, the same procedure would be used as outlined on page 7 in the section Blades without metering orifices, except that $p_a - h_B$ would be substituted for p_a in the calculations.

<u>Blades with metering orifices at base and designed for constant (or specified variation of) permeability around periphery.</u> - On the basis of fabrication, it may be desirable to design blades for a uniform permeability or other specified permeability variations that will simplify manufacture. The calculations for this case are as follows:

(1) For an assumed value of T_a and the design value of the blade temperature $T_{B,g}$, calculate $\frac{\mu_0^2 T_0}{\mu_{a,B}^2 T_{a,B}}\left(\frac{p_a^2 - p_{g,B}^2}{\tau}\right)$ at a location on the blade surface near the stagnation point as outlined in steps (1), (2), and (3) in the section on blade permeability calculations for transpiration cooled blades without metering orifices (p.7). At this location on the blade, no metering orifice will be required.

(2) Using the design or specified value of τ for the blade and the values of $p_{g,B}$ and $\frac{\mu_0^2 T_0}{\mu_{a,B}^2 T_{a,B}}$ as explained in step (4) on page 7, calculate the required p_a from the value of $\frac{\mu_0^2 T_0}{\mu_{a,B}^2 T_{a,B}}\left(\frac{p_a^2 - p_{g,B}^2}{\tau}\right)$ from step (1) above.

(3) Using the value of p_a obtained in step (2) above, and equation (B5), determine whether the assumed value of T_a was correct - if not, repeat the procedure as many times as necessary.

At all other locations on the blade an orifice will be required for metering the cooling air. The orifice constant C_B, which is a function of the orifice size, can be calculated as follows:

2597

(4) Repeat steps (1) and (2) for each other location on the blade, except that the value of p_a thus found will be defined as $p_a - h_B$.

(5) Subtract the value of $p_a - h_B$ from step (4) from the final value of p_a in step (2) to obtain h_B and use the value of $\rho_a v_a$ also obtained from step (4) to solve for the orifice constant C_B from equation (6) or (6a), depending on the values of h_B and p_a.

The local over-all permeabilities for the blade with orifices in the base are now specified by τ and the orifice constant C_B.

Off-Design Blade Temperatures and Coolant Flow Requirements

Blades without metering orifices. - In previous sections, methods are presented for finding the cooling-air flow rate and blade permeability required in order for the blade to be cooled to some specified temperature. In the design of transpiration-cooled turbine blades, a set of design conditions must be chosen and the blades must then be fabricated so that they will cool to the specified temperature level at those design conditions. Once the blades are fabricated, the blade permeability is constant and the quantity of coolant that will flow through the blade walls is determined by the wall temperature and the pressure levels on the inside and outside of the blade walls. The pressure distribution around the outside of the blade can vary from the pressure distribution at the design conditions. For blades without metering orifices, however, the coolant supply pressure does not vary around the blade periphery, and it is therefore quite likely that the coolant flow will no longer be metered through the porous wall in such a manner that the entire turbine blade will be at a uniform temperature level. The coolant supply pressure must be so set that the part of the blade most difficult to cool is adequately cooled and this may result in overcooling of other portions of the blade.

The compression required to provide the proper coolant supply pressure will determine the coolant supply temperature. Since the coolant supply temperature has an effect in determining the coolant flow required, which, in turn, affects the coolant supply pressure, it is necessary that p_a and T_a be obtained by iteration using equation (B5) and the following procedure:

(1) For an assumed value of T_a and the maximum permissible value of blade temperature $T_{B,g}$, calculate $\frac{\mu_0{}^2 T_0}{\mu_{a,B}{}^2 T_{a,B}}\left(\frac{p_a{}^2 - p_{g,B}{}^2}{\tau}\right)$ for each position on the blade surface as outlined in steps (1), (2), and (3) (p. 7) in the section on blade permeability calculations for blades without metering orifices.

(2) Using the design value of τ and the values of $p_{g,B}$ and $\frac{\mu_0{}^2 T_0}{\mu_{a,B}{}^2 T_{a,B}}$ as explained in step (4) on page 7, calculate the required p_a for each selected position on the blade surface.

(3) Using the maximum value of p_a obtained in step (2) above and equation (B5), determine whether the assumed value of T_a was correct - if not, repeat the procedure as many times as necessary.

After determining the values of T_a and p_a for the position on the blade that resulted in the highest value of p_a, the local blade temperatures at all other positions on the blade must be obtained by iteration in the following manner:

(1) Assume a value of $T_{B,g}$ and remembering that $T_{a,B} = T_{B,g}$, calculate $\frac{\mu_0{}^2 T_0}{\mu_{a,B}{}^2 T_{a,B}}\left(\frac{p_a{}^2 - p_{g,B}{}^2}{\tau}\right)$

(2) Read $\frac{\mu_0}{\mu_{a,B}}(\rho_a v_a)$ from figure 2.

(3) Calculate $T_{B,g}$ by use or equation (1) written in the following form:

$$T_{B,g} = T_a + \frac{r(T_{g,e} - T_a)}{e^m - (1-r)} \tag{7}$$

where

$$m = 71.3 \cdot (\rho_a v_a)\left(\frac{T_{B,g}}{\rho_{g,B} V_{g,B} T_{g,B}}\right)(Re_{B,g})^{0.1}(Pr_g)^{2/3}$$

(4) If assumed and calculated values of $T_{B,g}$ do not agree, iteration is required until agreement is obtained. The effect of $T_{B,g}$ on the Reynolds number $Re_{B,g}$ can be neglected for small variations in $T_{B,g}$.

After the iteration to determine $T_{B,g}$ is complete, the final value of $\rho_a v_a$ that was used in equation (7) can be used in equation (2)
2597 to calculate the coolant flow ratio required.

Blades with metering orifices at base. - The method of calculating the coolant supply pressure and temperature, the blade temperature, and the required coolant flow for the blade with metering orifices is complicated by the fact that in addition to the iteration processes used to find p_a, T_a, and $T_{B,g}$, iteration must also be employed to find the pressure drop across the orifice h_B. The method of calculation can be divided into three parts: (1) the calculation of the coolant supply pressure p_a and the coolant temperature T_a, (2) the calculation of the pressure drop across the orifice h_B and the local blade temperature $T_{B,g}$, and (3) the calculation of the coolant flow ratio w_a/w_g. The method of calculation is the same for blades designed for a specified pressure drop through the wall or for blades designed for a specified permeability variation.

The method of calculating the coolant supply pressure p_a and the coolant temperature T_a for the portion of the blade that is most difficult to cool is as follows:

(1) Assume a value of coolant supply pressure p_a.

(2) Calculate the coolant temperature T_a using the assumed value of p_a and equation (B5).

(3) Using the maximum permissible blade temperature $T_{B,g}$ and the coolant temperature T_a from step (2), calculate $\rho_a v_a$ by means of equation (1).

(4) Calculate $\frac{\mu_0}{\mu_{a,B}}(\rho_a v_a)$ and read $\frac{\mu_0^2 T_0}{\mu_{a,B}^2 T_{a,B}}\left(\frac{p_a^2-p_{g,B}^2}{\tau}\right)$ from figure 2 as explained in the section on blade permeability calculations. Figure 2 is a curve relating the coolant flow with the pressure drop through the porous wall; therefore for a blade with orifices in the base the ordinate becomes

$$\frac{\mu_0^2 T_0}{\mu_{a,B}^2 T_{a,B}}\left[\frac{(p_a-h_B)^2-p_{g,B}^2}{\tau}\right]$$

(5) The value of (p_a-h_B) can be calculated using the design value of τ for a blade with orifices.

(6) Use the value of $\rho_a v_a$ that was used in step (4) to solve equation (6) for h_B.

2597

$$h_B = \frac{p_a \pm \sqrt{p_a^2 - 4T_a\left(\frac{\rho_a v_a}{C_B}\right)^2}}{2} \tag{8}$$

(7) If (p_a-h_B) from step (5) plus h_B from step (6) is not equal to the assumed value of p_a in step (1), iteration is required until agreement is obtained.

(7a) If the value of h_B from step (6) is greater than $0.472p_a$, step (7) should be replaced by the following procedure: Calculate $\rho_a v_a$ by use of equation (6a) and check with the value used in step (4). If the two values of $\rho_a v_a$ are not equal, iteration with p_a as the assumed variable is required until agreement is reached.

The calculations outlined in steps (1) through (7a) should be made at each selected location on the blade surface. The maximum values of p_a and T_a will then be used for calculating the orifice pressure drops and local blade temperatures for the rest of the blade.

The method for calculating the orifice pressure drop and the local blade temperature at all other locations on the blade surface is as follows:

(1) Assume a value of pressure drop through the orifice h_B.

(2) Assume a value of blade temperature $T_{B,g}$.

(3) Calculate $\dfrac{\mu_0^2 T_0}{\mu_{a,B}^2 T_{a,B}}\left[\dfrac{(p_a-h_B)^2-p_{g,B}^2}{\tau}\right]$ and read $\dfrac{\mu_0}{\mu_{a,B}}(\rho_a v_a)$ from figure 2.

(4) Calculate $\rho_a v_a$ from step (3) and then calculate h_B by use of equation (8). If h_B from this step is not equal to the assumed value in step (1), iteration is required until agreement is obtained.

(4a) When h_B is greater than $0.472p_a$, disregard step (4) and instead calculate $\rho_a v_a$ by means of equation (6a). If the two values of $\rho_a v_a$ in steps (3) and (4a) do not agree, iteration with h_B as the assumed variable is required until agreement is obtained.

2597

(5) Using the value of h_B obtained by iteration and remembering that for blades with orifices the ordinate in figure 2 should be $\frac{\mu_0{}^2 T_0}{\mu_{a,B}{}^2 T_{a,B}}\left[\frac{(p_a-h_B)^2-p_{g,B}{}^2}{\tau}\right]$, calculate $T_{B,g}$ by iteration in the same manner as on page 12 in the section for transpiration-cooled blades without metering orifices.

(6) If the calculated value of $T_{B,g}$ from step (5) does not agree with the assumed value in step (2), further iteration is required for both h_B and $T_{B,g}$.

The coolant flow ratio w_a/w_g is calculated by means of equation (2) using the value of $\rho_a v_a$ obtained in the final iterative calculation of the blade temperature $T_{B,g}$.

Assumed Conditions for Calculations

For all calculations in this report, the pressure distributions around the rotor and stator blades were as shown in figure 4. These distributions were obtained from the stream filament theory of reference 6 and by use of equations (B7), (B9), and (B13) to (B22). The distributions can be considered as typical for an engine compressor pressure ratio of 4; however, it must be emphasized that this turbine would not be matched to the compressor for all design conditions considered herein. The purpose of this report is to illustrate a method of analyzing transpiration-cooling requirements and to show some typical trends; therefore the actual pressure distributions for each design condition are not required. Calculations that were made for an engine compressor pressure ratio of 10 would indicate trends that may be obtained for only the first stage of a turbine. As a general rule, two or three turbine stages are required to drive a compressor with a pressure ratio of 10.

For all calculations, it was assumed that the turbine stator configuration was fixed and that the stator was choked; consequently, the mass flow through the engine would vary with turbine-inlet temperature and with compressor pressure ratio. The compressor pressure ratio was assumed invariable with altitude and flight speed. In actual practice, a variation in engine speed would be required to maintain the compressor pressure ratio constant.

The pressure distributions shown for the rotor blade in figure 4 were calculated for a rotor speed at the mean blade span location of 1055 feet per second, a stator discharge angle of $22\frac{1}{4}^{\circ}$ from the plane of rotation, and a stator discharge Mach number of 1.1. No consideration was given to radial variations in pressure distribution. The different rotor pressure distributions for the two gas temperatures result from different stator discharge velocities for a constant Mach number.

2597

The ratio of specific heats was taken as 1.4 for all calculations as recommended in reference 11. The Prandtl number was assumed to have a constant value of 0.66. Standard NACA temperatures and pressures were used for the calculations at the three altitudes of sea level, 25,000 feet, and 50,000 feet. The assumed values of total pressure recovery factor ψ used in equation (B2) for the engine inlet diffuser were as follows:

Flight Mach number M	Total pressure recovery factor ψ
0.5	0.98
.8	.965
1.0	.95

For all conditions, the total-pressure drop across the burners was assumed to be 3 percent of the total-pressure level at the burner inlet. The compressor efficiency was assumed to be 0.85. The local blade temperature recovery factor Λ was taken as 0.89 as recommended in reference 12.

Local temperatures and coolant flow rates were calculated for four representative locations on the stator blade, namely, 18.5 and 74 percent chord on the suction surface and 20 and 81 percent chord on the pressure surface. These locations corresponded to 0.05 and 0.20 foot from the leading edge on both surfaces. On the rotor blade the pressure distributions on opposite sides of the blade were more uniform; therefore calculations were made for only one location on each surface, namely, 51 percent chord on the suction surface (0.128 ft from leading edge) and 45 percent chord on the pressure surface (0.09 ft from leading edge). The cooling-air temperature was assumed to be the temperature that would result from the compression necessary for the coolant supply pressure, except for two cases where cooling-air aftercoolers were considered. An infinitely variable pressure ratio was assumed for the cooling-air compressor.

The variation of flow through the porous walls as the pressures and temperatures were varied was determined by means of the correlation plot in figure 2, which was obtained from actual data in reference 10. Extrapolation of the curve (indicated by the dashed portion) was required for a few calculations, but the majority of the calculations were in the range where experimental data were available. This method of calculation using an experimental curve eliminated any errors due to nonlinearity of the flow curve.

2597

Variables Investigated

A series of calculations were made to determine the effects of design altitude, design flight speed, and design gas temperature on the coolant flow requirements at off-design conditions, and to determine how engine compressor pressure ratio, design blade temperature, and aftercooling of the cooling air affected over-all transpiration-cooling requirements. In addition, consideration was given to the incorporation of metering orifices at the blade base to partly meter the cooling air to various peripheral locations on the blade in an effort to provide a blade that (1) is less sensitive to off-design altitude conditions, and (2) would be much easier for the manufacturer to fabricate. The conditions for these calculations are given in tables I and II and are discussed in the following paragraphs.

Effect of design altitude on off-design coolant flow and coolant compressor pressure ratios. - Coolant flow ratios and coolant compressor pressure ratios were calculated for altitudes of sea level, 25,000, and 50,000 feet for an engine with a compressor pressure ratio of 4 and for turbine-inlet temperatures of 1600° and 2500° F. All calculations were made for a maximum blade temperature of 1000° F and a flight Mach number of 1.0 for blades without metering orifices. For a turbine-inlet temperature of 1600° F, consideration was given to blades designed for each of the three altitudes. The ideal coolant flows and coolant pressure ratios were calculated for a uniform blade temperature of 1000° F at each altitude. These ideal conditions constitute design conditions. The off-design coolant flow and coolant pressure ratio requirements were then calculated for altitudes from sea level to 50,000 feet for blades designed for each of the three altitudes mentioned. For a turbine-inlet temperature of 2500° F, similar calculations were made, except that off-design coolant flows and pressure ratios were calculated for blades designed for altitudes of sea level and 50,000 feet only.

Effect of flight Mach number on off-design coolant flow and coolant compressor pressure ratios. - Calculations were made to determine the off-design coolant flow and coolant compressor pressure ratios for flight Mach numbers of 0, 0.5, and 0.8 for transpiration-cooled rotor and stator blades designed for a flight Mach number of 1.0, an engine with a compressor pressure ratio of 4, a turbine-inlet temperature of 1600° F, and blades without metering orifices operating at a maximum temperature of 1000° F. The ideal coolant flow and coolant pressure ratios were also calculated for the entire Mach number range.

Effect of design gas temperature on off-design coolant flow and coolant compressor pressure ratios. - Calculations were made to determine the effects on coolant flow and coolant compressor pressure ratio that would be obtained for transpiration-cooled rotor and stator blades without metering orifices operating at two turbine-inlet temperatures, 1600° 2597
and 2500° F. For operation at a turbine-inlet temperature of 1600° F, a flight Mach number of 1.0, an engine compressor pressure ratio of 4, and a maximum blade temperature of 1000° F, the required coolant flow and coolant pressure ratios were calculated for altitudes of sea level, 25,000 feet, and 50,000 feet for (1) blades designed for an altitude of 50,000 feet and a turbine-inlet temperature of 1600° F, and (2) blades designed for an altitude of 50,000 feet and a turbine-inlet temperature of 2500° F.

Effect of blade design conditions on stator blade temperature distribution at off-design conditions. - The peripheral blade temperature distributions obtained at off-design conditions were calculated and compared for three stator blade designs without metering orifices. The comparisons were made for the following engine and flight conditions: engine compressor pressure ratio, 4; turbine-inlet temperature, 1600° F; maximum blade temperature, 1000° F; flight Mach number, 1.0; and flight altitudes of sea level, 25,000 feet, and 50,000 feet. The three blades compared were designed for the following conditions: (a) sea-level altitude and a turbine-inlet temperature of 1600° F; (b) 50,000 feet of altitude and a turbine-inlet temperature of 1600° F; and (c) 50,000 feet of altitude and a turbine-inlet temperature of 2500° F.

Effect of metal temperature on rotor and stator coolant flow requirements. - The coolant flow requirements for rotor and stator blades without metering orifices were calculated for a maximum blade temperature of 1000° F for most cases in this analytical investigation because limited information is available for porous materials made of anything but copper, bronzes, and stainless steel. Stainless steel can withstand higher temperatures than copper or bronze, but the strength of even stainless steel is low at temperatures above 1000° to 1100° F.

For gas turbines that are not transpiration cooled, the usual practice is to allow the stator blades to operate at a higher temperature than the rotor blades because the rotor blade stress level is so much higher. For all materials considered for transpiration cooling, some method of support will probably be required in the rotor blades to relieve the stresses in the porous shell. Even if the porous shell were made of wire cloth, which has strength characteristics superior to those of sintered materials, an internal strut would still be required to provide the proper blade profile and rigidity. Because of the elasticity of wire cloth it will transfer its load to any member to which it is attached with the result that wire cloth will also have to be supported by an internal strut. The stresses in the porous metal will therefore be low and the stress level in both rotor and stator blades will be of the same order of magnitude; consequently, the maximum permissible temperature will be approximately the same for both blades. A differential of 100° F is probably about the maximum permissible for currently available porous materials. Calculations were made to determine the coolant requirements for two stator temperatures, 1000° and 1100° F, to compare with the coolant requirements for the rotor at a temperature of 1000° F. The calculations were made for an engine compressor pressure ratio of 4, a turbine-inlet temperature of 1600° F, a flight Mach number of 1.0, and altitudes of sea level, 25,000 feet, and 50,000 feet. The ideal coolant flow ratios and the required coolant flow ratios for a design altitude of 50,000 feet were calculated.

At the present time no known porous materials are available that will have a reasonable strength at a metal temperature of 1500° F; however, it appears feasible that with some development high-temperature materials could be made into either porous sintered compacts or wire cloth. Calculations were therefore made to determine the effect of allowing the metal temperature to increase from 1000° to 1500° F on the ideal coolant flow requirements for rotor and stator blades for the following engine and flight conditions: engine compressor pressure ratio, 4; turbine-inlet temperature, 2500° F; flight Mach number, 1.0; and altitudes of sea level, 25,000 feet, and 50,000 feet.

Effect of engine compressor pressure ratio on coolant flow. - In order to determine whether the coolant flow was appreciably affected by significant changes in the engine compressor pressure ratio, calculations were made to find the ideal and required flow rates for rotor and stator blades without metering orifices and designed for an altitude of 50,000 feet for engine compressor pressure ratios of 4 and 10. The calculations were made for a flight Mach number of 1.0, a maximum blade temperature of 1000° F, and turbine-inlet temperatures of 1600° and 2500° F. For an engine compressor pressure ratio of 10, the calculations can be considered applicable for only the first stage of the turbine. In actual application, two or three stages would probably be required to obtain the necessary turbine work to drive a compressor with this pressure ratio.

Effect of cooling-air aftercoolers on ideal coolant flow requirements. - For high compressor pressure ratios, the cooling-air temperature often becomes so high that the cooling effectiveness is greatly diminished. Calculations were therefore made to determine the effects on ideal coolant flow ratios of aftercooling the cooling air. The calculations were made for an engine compressor pressure ratio of 10, a flight

Mach number of 1.0, a turbine-inlet temperature of 2500° F, uniform rotor and stator blade temperatures of 1000° F, and altitudes of sea level, 25,000 feet, and 50,000 feet. Consideration was given to the following cases: (1) no cooling-air aftercooling, (2) aftercooling to limit the cooling-air temperature to 700° F (for some altitudes the cooling-air temperature would be lower, but never higher), and (3) aftercooling to limit the cooling-air temperature to 300° F.

Permeability variations around stator blades without metering orifices for various design conditions. - Transpiration-cooled turbine blades without metering orifices must have the permeability varied around the blade periphery in a specified manner in order to obtain uniform cooling at the design conditions. The permeabilities were calculated in terms of $12K'/\tau$ for a pressure drop through the blade wall of 10 pounds per square inch discharging to NACA standard sea-level conditions. The units of permeability were converted from feet to inches by use of a factor of 12 to be consistent with other investigators. All calculations were made for a maximum stator blade temperature of 1000° F and a flight Mach number of 1.0. The permeabilities were calculated for the following cases: 2597

(1) Blade designed for sea-level altitude, engine compressor pressure ratio of 4, and turbine-inlet temperature of 1600° F

(2) Blade designed for 50,000 foot altitude, engine compressor pressure ratio of 4, and turbine-inlet temperature of 1600° F

(3) Blade designed for 50,000 foot altitude, engine compressor pressure ratio of 4, and turbine-inlet temperature of 2500° F

(4) Blade designed for 50,000 foot altitude, engine compressor pressure ratio of 10, and turbine-inlet temperature of 2500° F

(5) Blade designed for 50,000 foot altitude, engine compressor pressure ratio of 10, and turbine-inlet temperature of 1600° F

Use of orifices at blade base for improving off-design coolant flow requirements for blades designed for specified pressure drop through wall. - By rearrangement of equation (3) it can be shown that for a constant temperature, in which case changes in viscosity can be neglected, the flow of air through a porous material of a specified thickness can be represented by

$$\rho_a v_a = f\left[\rho \Delta p \left(\frac{p_a}{p_{g,B}} + 1\right)\right] \tag{9}$$

and from equation (6), the flow of air through an orifice in the form of a flow nozzle can be represented by

$$\rho_a V_a = F\left(\sqrt{\rho \Delta p}\right) \tag{10}$$

where Δp is the pressure drop across the orifice or porous material and ρ is the air density on the discharge side of the orifice or porous material.

2597

Comparison of equations (9) and (10) shows that changes in the pressure drop and discharge air density have a much greater effect on the quantity of air flow through a porous material than through an orifice; therefore partly metering cooling air to local positions on the blade through orifices might be better than metering only through a porous material for blades that must operate over a wide range of altitudes. For the calculations made for blades with orifices for partly metering the cooling air, it was assumed that a number of small orifices were employed at the base of each blade with each orifice supplying air to a separate compartment inside the coolant passage, as illustrated in figure 3. These orifices were of various sizes so that coolant supplied to the blade base at a constant supply pressure for the whole blade would be metered with varying pressures to the numerous locations around the blade periphery. At the leading-edge portion of the blade, metering orifices are probably unnecessary. For the sea-level design condition, the orifices were assumed of such size that the pressure drop across the porous wall of the blade would be 2 pounds per square inch at all locations on the blade periphery. The coolant flow ratios and coolant compressor pressure ratios were calculated at off-design altitudes for an engine compressor pressure ratio of 4, a flight Mach number of 1.0, turbine-inlet temperatures of 1600° and 2500° F, and a maximum blade temperature of 1000° F. The coolant flow ratios and coolant compressor pressure ratios for the blade with orifices in the blade base for partly metering the cooling air were compared with results obtained for blades designed for an altitude of 50,000 feet without metering orifices at the blade base.

Effect of constant permeability around periphery of blades utilizing orifices at base on off-design coolant flow and coolant compressor pressure ratio requirements. - The manufacture of transpiration-cooled gas-turbine blades can be greatly simplified if the wall permeability is maintained constant around the blade periphery. The proper distribution of cooling air for such a blade would be obtained by proper choice of the metering orifices in the blade base, in a manner similar to that used in designing the blade for specified pressure drop through the wall. Calculations were made for a stator blade having a constant permeability $12K'/\tau$ of 45×10^{-9} inches and for a rotor blade having a constant permeability of 35×10^{-9} inches. The blades were designed for sea-level altitude and off-design coolant flows, and coolant compressor pressure ratios were calculated for altitudes of 25,000 and 50,000 feet for an engine compressor pressure ratio of 4, a flight Mach number of 1.0, a turbine-inlet temperature of 2500° F, and a maximum blade temperature of 1000° F. The results obtained for these blades were compared with those obtained for the blade with orifices at the base that was designed for a pressure drop of 2 pounds per square inch through the blade wall.

Effect of orifices at blade base on wall permeabilities and temperature distribution. - The permeability variations required for transpiration-cooled blades utilizing orifices in the blade base were calculated for a blade designed for sea-level altitude with a pressure drop through the porous wall of 2 pounds per square inch. The calculations were made for two design turbine-inlet temperatures, 1600° and 2500° F, with an engine compressor pressure ratio of 4, a flight Mach number of 1.0, and a design blade temperature of 1000° F. 2597

Rotor and stator blade temperature distributions were calculated for off-design altitudes for the following three blade designs: (1) blades designed for a pressure drop through the blade wall of 2 pounds per square inch for a turbine-inlet temperature of 1600° F and sea-level altitude, (2) blades designed for a pressure drop through the wall of 2 pounds per square inch for a turbine-inlet temperature of 2500° F and sea-level altitude, and (3) blades designed for a uniform permeability around the blade periphery, a turbine-inlet temperature of 2500° F, and sea-level altitude. All three blades were designed for a flight Mach number of 1.0, an engine compressor presure ratio of 4, and a blade temperature of 1000° F.

RESULTS AND DISCUSSION

The results of this analytical investigation to determine the effects of a number of variables on transpiration cooling of gas-turbine blades, using the calculation procedure described herein, are presented in figures 5 to 17 and are discussed in the following sections. A summary of the coolant flow and coolant compressor pressure ratio requirements for all calculations made is given in tables I and II and rotor blade permeabilities are given in table III.

Effect of Design Conditions on Coolant-Flow Requirements at Off-Design Conditions for Blades without Metering Orifices

Effect of design altitude. - Figure 5(a) shows the coolant flow and coolant pressure ratio requirements for stator and rotor blades designed for altitudes of sea level, 25,000 feet, and 50,000 feet for a turbine-inlet temperature of 1600° F. The maximum blade temperature was 1000° F, the flight Mach number was 1.0, and the engine compressor pressure ratio was 4. The dashed lines labeled "Ideal requirements" represent the coolant flow and coolant pressure requirements for blades cooled to a uniform temperature of 1000° F. For some conditions the ideal coolant pressure requirements could not be indicated by the dashed lines because the ideal curve was almost identical with curves shown for other conditions.

In order to cool the blades to a uniform temperature, a different permeability would be necessary at each altitude. Since in actual practice a different blade design cannot be used at each altitude, parts of
2597 the blade will be overcooled at off-design conditions and the cooling air flow will therefore be more than that ideally required. This effect is shown for blades designed at each of the three altitudes.

Blades designed for sea-level conditions must be less permeable than blades designed for altitude because the pressure levels are higher. As can be seen from equation (3), the coolant flow through a porous wall is a function of the difference of the squares of the pressures on opposite sides of the blade wall. The ideal coolant flow is more nearly a function of the first power of the pressure level on the outside of the blade. This is evidenced by the fact that the ideal coolant flow ratio is not greatly affected by altitude, as shown in figure 5. As the blades designed for sea level are taken to altitude, the coolant supply pressure ratio must be increased substantially above the engine compressor pressure ratio in order to force enough coolant through the blade wall. At 50,000 feet, the coolant compressor pressure ratio for the stator blade shown in figure 5(a) would have to be increased from the sea-level value of 3.97 to 10.81. Unequal cooling of the turbine blades around their periphery plus the fact that the high required coolant supply pressure results in a high cooling-air temperature make the coolant flow exhorbitant for blades designed for sea level that are operating at 50,000 feet of altitude. For the blades shown in figure 5(a), the combined coolant flow ratio for the rotor and stator blades at 50,000 feet is 0.136 as compared with the ideally required value of 0.039 for cooling the blades to a uniform temperature. The excess coolant flow at 50,000 feet is about 250 percent.

When blades designed for a high altitude must operate at a lower altitude, the permeability is too great for most of the blade and causes excess cooling-air flow, but the coolant supply pressure levels are quite low.

The coolant supply pressures at lower altitudes for rotor blades designed for high altitudes are even lower than the ideal coolant supply pressures because for the ideal conditions the coolant supply pressure was assumed to be 2 pounds per square inch higher than the pressure at any location on the outside surface of the blade. For blades designed for high altitudes, the actual pressure difference is sometimes less than 2 pounds per square inch when operating at lower altitudes.

For the stator and rotor blades in figure 5(a) designed to cool uniformly at an altitude of 50,000 feet, the maximum excess of cooling air supplied at any other altitude is less than 60 percent. The results in figure 5 indicate that it is best to design transpiration-cooled turbine blades for the highest altitude at which they will operate.

The effect of altitude on coolant flow requirements is less severe for rotor blades than for stator blades. As discussed previously, transpiration-cooled rotor and stator blades will probably have to operate at about the same metal temperature. For this condition the rotor blade requires less cooling than the stator blade because the total gas temperature is lower. For instance, for the configurations investigated in this report at a turbine-inlet total temperature of 1600° F, the total gas temperature relative to the rotor blades was 1334° F, and at a turbine-inlet total temperature of 2500° F, the total gas temperature relative to the rotor blades was 2177° F. The small effect of altitude on the magnitude of the coolant flow of the rotor blades in figure 5(a) is primarily due to the very low coolant flow rates. For a turbine-inlet temperature of 2500° F, as shown in figure 5(b), and higher rotor cooling-air requirements, the effect of altitude on the coolant flow requirements for blades designed for sea-level operation is much greater. However, there is less variation in the coolant pressure requirements for blades designed to operate at a gas temperature of 2500° F than for blades designed for 1600° F. The blades designed for a turbine-inlet temperature of 2500° F are made much more permeable to take care of the greater quantities of cooling air required, and the more permeable materials do not require such a wide range of pressure variations to meter the flow.

Comparison of figures 5(a) and 5(b) shows that to increase the turbine-inlet temperature from 1600° to 2500° F, or about 900° F above current practice, for blades designed for an altitude of 50,000 feet and operating at sea level would require an increase in combined coolant flow ratio for both rotor and stator blades from 0.056 to 0.142, or about $2\frac{1}{2}$ times as much coolant would be required at the high temperature as at the lower temperature.

It is shown in figure 5 that designing the blades to cool uniformly at the highest altitude considered gives a required cooling-air pressure for the rotor blades sufficiently low that cooling air could be bled from the engine compressor, which has a pressure ratio of 4. For nearly all conditions for the stator blades, however, a boost compressor would be required.

Effect of design flight Mach number. - Figure 6 shows the coolant flow requirements for a range of flight Mach numbers from 0 to 1.0 for blades designed to cool uniformly at a flight Mach number of 1.0. The

ideal coolant flow requirements are also shown for the range of Mach numbers. The effect of flight speed on coolant flow requirements is much less than the effect of altitude and does not constitute a serious problem. Except for take-off, most aircraft operate at a relatively constant flight speed so that if the turbine blades are designed for operation at design flight speed, only a very small excess of coolant flow will be required at other speeds. For take-off, slightly over 50 percent excess cooling air may be required for the stator blades, but
2597 since this operation is only for a short period of time the excess cooling-air requirement can probably be tolerated.

As shown in table I, the coolant compressor pressure ratios required over the range of flight Mach numbers vary only slightly.

Effect of design gas temperature. - For some applications, an engine capable of operation at very high gas temperatures may be desirable in order to obtain very high power output for relatively short periods of time. Most operation, however, will probably require cruise at a lower gas temperature level. An example of the cooling requirements for such a case is shown in figure 7 for operation at a gas temperature of 1600° F, a flight Mach number of 1.0, an engine compressor pressure ratio of 4, and a maximum blade temperature of 1000° F. Curves are presented showing the cooling requirements for (1) the ideal case, (2) blades designed to obtain uniform cooling at an altitude of 50,000 feet and a turbine-inlet temperature of 1600° F, and (3) blades designed to obtain uniform cooling at an altitude of 50,000 feet and a turbine-inlet temperature of 2500° F; all the blades are operating at a turbine-inlet temperature of 1600° F. A design altitude of 50,000 feet was chosen for both design gas temperatures because figure 5 showed that the 50,000 foot design resulted in smaller excesses in cooling air than designs for other altitudes. As seen in figure 7, when blades are designed for operation at a gas temperature of 2500° F, excessive quantities of cooling air are used when the gas temperature is reduced to 1600° F. For the rotor and stator blades at an altitude of 25,000 feet, the combined coolant flow ratio for the blades designed for a turbine-inlet temperature of 2500° F was 0.115 when operating at a turbine-inlet temperature of 1600° F, as compared with a value of 0.052 for blades designed for 1600° F. This excess of air is caused by the fact that the blades designed for high gas temperatures must be more permeable than blades designed for lower gas temperatures. The coolant supply pressure must be almost as high at low gas temperatures as at high gas temperatures in order that the blade leading edges will be adequately cooled (see table I); and since the blade designed for the high gas temperature is more permeable, excessive quantities of coolant will be supplied to the rest of the blade at low gas temperatures. There is some reduction in coolant flow as the gas temperature is reduced, however, as can be noted by comparing the cooling requirements for the 50,000 foot design in figure 5(b) with the requirements shown in figure 7. The same blade designs were used in both figures.

To design a blade for uniform cooling at a low gas temperature and then to operate it at a high gas temperature would probably be impractical because of the high coolant supply pressures required to meter an adequate amount of air through the less permeable blade at high gas temperatures. For this reason, blades designed for uniform cooling at a gas temperature of 1600° F were not considered for operation at a gas temperature of 2500° F in this analysis.

Effect of Design Conditions on Stator Blade Temperature Distribution at Off-Design Conditions for Blades without Metering Orifices

Stator blade temperature distributions for altitudes of sea level, 25,000 feet, and 50,000 feet are shown in figure 8 for three blade designs operating at a turbine-inlet temperature of 1600° F. In figure 8(a) are shown the temperature distributions for stator blades designed to cool uniformly at sea level. At altitudes of 25,000 and 50,000 feet, there are temperature variations in excess of 500° F around the blade periphery. These high temperature gradients are the result of the excess cooling air in these regions. (The cooling-air requirements are shown in figure 5(a).)

In figure 8(b) are shown the temperature distributions for a stator blade designed to cool uniformly at an altitude of 50,000 feet. At off-design altitudes the temperature gradients are less severe than for the blade designed for uniform cooling at sea level. The maximum temperature variation was about 250° F for this design. This smaller temperature variation follows from figure 5(a), which shows that considerably more coolant is required at off-design conditions for blades designed for sea level than for blades designed for an altitude of 50,000 feet. It is also of interest that for the blade designed for sea level the lowest blade temperatures occur near the leading edge at off-design conditions, but for the blade designed for an altitude of 50,000 feet the lowest temperatures occur near the trailing edge at off-design conditions.

In figure 8(c) are shown the temperature distributions for a stator blade designed for an altitude of 50,000 feet and a turbine-inlet temperature of 2500° F but operating at a turbine-inlet temperature of 1600° F, for which coolant flow ratios are shown in figure 7(a). Figures 8(b) and 8(c) show that at all locations except the leading edge the temperatures of the blade designed for a turbine-inlet temperature of 2500° F are much cooler than those of the blade designed for a turbine-inlet temperature of 1600° F. These temperature distributions further illustrate why the excess cooling air, shown in figure 7(a), is required when a blade designed for operation at 2500° F is operated at the lower temperature of 1600° F.

So far as engine performance is concerned, the most important consideration in transpiration cooling is the quantity of air that must be used in cooling, but the high temperature gradients that result in blades that require excess cooling air at off-design conditions may introduce structural difficulties in the blades due to thermal stresses. Both the quantity of excess cooling air required and the temperature gradients are therefore important, but the problems are so related that reduction of excess cooling air automatically reduces the problems of high temper-
2597 ature gradients.

Evaluation of Factors Affecting Coolant Flow Requirements for Specific Design Conditions for Blades without Metering Orifices at Base

Effect of metal temperature on rotor and stator coolant flow requirements. - As discussed previously, transpiration-cooled rotor and stator blades will probably have to operate at approximately the same temperature level. In figure 9 are shown comparisons of rotor and stator blade cooling-air requirements for the blades operating at the same temperature and for the stator blade operating at a temperature 100° F higher than the rotor blade. In figure 9(a) are shown the ideal coolant flow requirements to obtain uniform blade temperatures. For a turbine-inlet temperature of 1600° F and the rotor and stator blades cooled to the same temperature, the stator blade requires from $2\frac{1}{2}$ to 3 times as much coolant as the rotor blade because the gas temperature level relative to the rotor is over 250° F less than it is relative to the stator, and the cooling-air temperature is lower for the rotor blade because a lower cooling-air pressure is required. Even when the stator temperature is allowed to increase to 1100° F, the stator blade requires approximately twice as much coolant as the rotor blade.

The actual coolant flow ratios for blades designed to cool uniformly at an altitude of 50,000 feet are shown in figure 9(b). The trends are the same as those shown in figure 9(a), except that the difference in coolant flow requirements between the rotor and the stator blades is even greater. For instance, at an altitude of 25,000 feet for both blades cooled to the same maximum temperature, the stator blade requires a coolant flow ratio of 0.039 as compared with 0.013 for the rotor blade. For engine designs where stator cooling is necessary, the coolant requirements for the stator will probably always be in excess of the requirements for the rotor unless improvement of materials or design method makes possible a substantial difference in permissible material temperature level between the rotor and the stator blades.

In figure 10 is shown the substantial reduction in ideal coolant flow requirements that could be obtained if transpiration-cooled turbine blades could be operated at a metal temperature of 1500° F instead of 1000° F. At a turbine-inlet temperature of 2500° F, cooling either rotor or stator blades to a temperature of 1000° F requires almost three times as much cooling air as cooling them to a temperature of 1500° F. As an example, at sea level the combined coolant flow ratio for the rotor and stator blades is 0.102 for a blade metal temperature of 1000° F as compared with only 0.035 for a metal temperature of 1500° F. This reduction in coolant flow ratio as the permissible blade metal temperature is increased stresses the importance of conducting research on porous metals made from materials that have better strength properties at high temperatures than those of stainless steels.

Effect of engine compressor pressure ratio on coolant flow requirements. - Figure 11 shows the effect of compressor pressure ratio on the coolant flow requirements of the first stage of a turbine for turbine-inlet temperatures of 1600° and 2500° F. The coolant flow required for a compressor pressure ratio of 10 is generally higher than that required for a pressure ratio of 4 because the added compression raises the cooling-air temperature. As noted in table I, however, the cooling-air temperature for these calculations was not allowed to exceed 700° F. The total cooling-air flows for the turbine of an actual engine with a compressor pressure ratio of 10 would probably be about double the values shown in figure 11 because for such a high pressure ratio two or three stages would be required, and for a turbine-inlet temperature of 2500° F all the stages would have to be cooled.

Effect of cooling-air aftercoolers on ideal coolant flow requirements. - With high compressor pressure ratios, the cooling-air temperature can become so high that the air is no longer effective for cooling. This effect is illustrated in figure 12, which shows the ideal cooling-air requirements for an engine with a compressor pressure ratio of 10 and a turbine-inlet temperature of 2500° F. The dash-dot line shows the ideal cooling-air requirements and the cooling-air temperature for no aftercooling of the cooling air. At sea level, where the ambient-air temperatures are considerably higher than at altitude, a very high coolant flow ratio of 0.105 is required for the stator blade because the cooling-air temperature is only 160° F less than the desired blade temperature. At sea level, use of an aftercooler to limit the cooling-air temperature to 700° F would reduce the coolant flow about 25 percent, and aftercooling to reduce the cooling-air temperature to 300° F would reduce the cooling-air flow over 50 percent.

Although advantageous, the effect of aftercooling the cooling air was less effective for the rotor blades because the cooling-air pressure is considerably less for the rotor blade, and therefore the cooling-air temperature is also less. If the cooling air had to be bled at the same pressure for both the rotor and the stator blades, the rotor blade coolant flows would be higher than those shown and aftercooling for rotor blades would be more beneficial than under the conditions considered herein.

[redacted]

The coolant flow ratios shown in figure 12 are all ideal values. For actual blade designs the cooling-air pressures required at off-design conditions become higher so that aftercooling is even more advantageous (see table I). In fact, for some conditions aftercooling is absolutely necessary because without aftercooling the high pressure required for cooling results in high temperatures; the high temperatures in turn result in a need for more cooling air and consequently higher pressures until the point is reached where the cooling air temperature exceeds the maximum allowable blade temperature.

2597

No consideration was given to the method of aftercooling or its effect on the over-all aircraft performance in this analysis. Aftercooling was considered here only to determine its effect on transpiration-cooling requirements.

Blade Permeability Requirements for Various Design Conditions for Blades without Metering Orifices

In order to give the manufacturer of permeable materials an idea of the material permeabilities and the peripheral variation in permeability required for gas-turbine blades, permeability values are given in figure 13 for stator blades designed for five different design conditions. The ranges of permeability required for rotor blades for the same five design conditions are given in table III. The permeability is presented in terms of $12K'/\tau$ for the reasons given in CALCULATION PROCEDURE. The number 12 is merely a conversion factor to change the units from feet to inches because in most of the literature the permeability coefficient K is in the dimension of square inches. It must also be emphasized that the permeabilities presented in figure 13 and in table III were not calculated for zero air flow as is sometimes done, but were calculated for a pressure drop through the wall of 10 pounds per square inch discharging to NACA standard sea-level atmospheric conditions because this value can be more easily measured in a laboratory. However, the values given in figure 13 and table III can be converted to permeabilities calculated for zero flow by use of figure 14. The curve presented in figure 14 was found to be valid for all five different samples tested of brazed and calendered stainless-steel wire cloth in references 9 and 10. The dotted portion of the curve was calculated from a single data point that was extrapolated from a permeability curve in reference 9; consequently, that portion of the curve requires further verification. Actual data were not available for a pressure drop of 10 pounds per square inch for such a high permeability material. By use of figures 13 and 14 and table III, the permeability ranges that can be expected for transpiration-cooled turbine blades can be determined for two different types of permeability measurement.

In figure 13(a) is shown the permeability variation required for blades designed for sea-level altitude, an engine compressor pressure ratio of 4, and a turbine-inlet temperature of 1600° F. This is the blade design shown to require very high coolant flows and coolant supply pressures at off-design conditions in figure 5(a) and therefore deemed an unsatisfactory design. The reason for the high coolant supply pressures is obvious when it is observed that the blade permeabilities are lower for this blade than for any of the others shown in figure 13. For example, the blade design for an altitude of 50,000 feet shown in figure 13(b), which is the best design shown in figure 5(a), has permeabilities near the trailing edge nearly twice as high as those shown for the blade designed for uniform cooling at sea level (fig. 13(a)). 2597 The permeability required at the blade leading edge is approximately the same for both blades operating in an engine with the same compressor pressure ratio and the same turbine-inlet temperature.

In figures 13(c) to 13(e) are shown the effects of engine compressor pressure ratio and turbine-inlet temperature on permeability requirements. The permeability distribution has approximately the same trend in all cases; only the values of the permeability vary. Compressor pressure ratio has a relatively small effect on the required permeability, but a turbine-inlet temperature of 2500° F requires permeabilities from 35 to 80 percent higher than those for a blade designed for a turbine-inlet temperature of 1600° F. The higher permeability is necessary because of the higher coolant flows required at higher gas temperatures.

The maximum ranges of permeability shown in figure 13 and table III can be obtained from single thicknesses of brazed and calendered 20X200 mesh stainless-steel wire cloth (reference 9) which has been reduced from 37 to 42 percent in thickness by rolling. Slightly less rolling would be satisfactory if multiple thicknesses of the wire cloth were used. Previous discussions herein show that improper blade design results in highly excessive required coolant flow. This blade design was improper because the blade permeability used was incorrect. On the suction surfaces of the blades designed for operation at a gas temperature of 1600° F and an engine compressor pressure ratio of 4, the permeability $12K'/\tau$ of the blade designed to cool uniformly at sea level (fig. 13(a)) was 3.5×10^{-9} inches and that of the blade designed to cool uniformly at an altitude of 50,000 feet (fig. 13(b)) was 7.5×10^{-9} inches at the same location (75 percent chord on the suction surface). To obtain this difference in permeability with 20X200 mesh calendered wire cloth would require rolling to 41 percent reduction in thickness for the higher permeability and to 42 percent for the lower permeability. There is a very great difference in the performance of these two blades, as can be observed in figure 5(a); yet there is only a very small difference in the amount of thickness reduction required to obtain the necessary difference in permeability. These figures emphasize the extreme care required in blade fabrication in order to provide a permeability variation around transpiration-cooled turbine blades without metering orifices that will result in economical expenditures of cooling air.

Use of Orifices at Blade Base for Improving Off-Design Coolant Flow Requirements

2597 It is shown in figure 5 that excessive coolant flow ratios are required at off-design altitudes for transpiration-cooled blades without metering orifices. The discussion of figure 13 showed that very exact control of permeabilities around the periphery of transpiration-cooled blades without metering orifices is required. In the calculation procedure, the difference between the laws relating flow rate to pressure drop through orifices and those relating it to pressure drop through porous walls was found to be great, with the result that orifices might provide a better means of metering the cooling air to transpiration-cooled turbine blades. Consideration was therefore given to transpiration-cooled rotor and stator blades with orifices at the blade base, as illustrated in figure 3, for two types of blade design: (1) blades designed for a specified pressure drop through the porous wall, and (2) blades designed for a constant permeability around the periphery. The results obtained from calculations to determine the coolant flow ratios and coolant compressor pressure ratios for such blades for a range of altitudes are discussed in the following paragraphs.

Blades designed for specified pressure drop through wall. - In figure 15 are shown the coolant flow requirements for blades using multiple orifices in the blade base for partial metering of the cooling air. The blades were designed for an altitude of sea level, gas temperatures of 1600° and 2500° F, and a pressure drop through the blade wall of 2 pounds per square inch. For purposes of comparison, the best designs for blades without metering orifices are also shown. The use of orifices at the blade base gives required and ideal cooling-air flows that are almost the same for the entire altitude range for a turbine-inlet temperature of 1600° F (fig. 15(a)). The maximum difference between required and ideal flows occurs at 25,000 feet and is less than 10 percent for both the rotor and the stator blades. The curves reproduced from figure 5(a), which denote the quantity of cooling air required for the 50,000 foot design if orifices had not been used in the blade base, show the savings in cooling air obtainable by use of multiple orifices in the base of each blade. For stator blades designed for an altitude of 50,000 feet, at least 50 percent more cooling air is required at altitudes of sea level and 25,000 feet than for the blades having orifices.

The cooling-air supply pressures are also quite low when orifices are used for partial metering of the cooling air. In table II it can be seen that the required and ideal coolant compressor pressure ratios are the same for the rotor blade, and for the stator blade the pressure ratios are even smaller at 25,000 and 50,000 feet than the ideal values calculated for a pressure drop through the wall of 2 pounds per square inch. It will be noted, however, that although the engine compressor could supply cooling air for the rotor blades, a boost compressor would still be required for the stator blades. This is a condition similar to that discussed in connection with figure 5.

The coolant flows required for blades with orifices in the blade base at a gas temperature of 2500° F (fig. 15(b)) are not so close to the ideal as they are for blades operating at a gas temperature of 1600° F; however, there is a marked improvement over the best design that does not utilize orifices for partial metering of the cooling air. The maximum excess cooling air required for the blades with orifices is about 14 percent for the rotor blade at an altitude of 50,000 feet. Although this difference most likely is not serious, further improvement could probably be obtained by designing the blades for a higher altitude rather than for sea level.

Blades designed for constant permeability around periphery. - As discussed previously, the manufacture of transpiration-cooled turbine blades is very greatly simplified if the blades can be built with a constant, or specified variation of, permeability around the periphery. The use of orifices at the blade base for partly metering the cooling air makes this type of blade design possible. The coolant flows required for this design at a turbine-inlet temperature of 2500° F are compared in figure 16 with the ideal flows and the flows for the blade discussed in connection with figure 15(b) that was designed for a specified pressure drop through the wall. The permeability of the blades will be discussed later.

Except for the stator blade at an altitude of 50,000 feet, the coolant flow requirements for the constant permeability blade are even lower than those for the blade designed for a specified pressure drop through the wall. The maximum excess in coolant flow above the ideal values was about 10 percent for the blades over the entire range of altitude. It would be fortuitous if the coolant flow for all blades designed for constant permeability would always be less than the flow for blades designed for a specified pressure drop through the wall. These results will depend largely upon the permeability chosen for the constant permeability blade; in general, the permeability should be high. This point will be discussed later.

Orifice sizes required. - The size of orifices required for partial metering of the cooling air will depend upon the wall permeability, the blade size, and the number of compartments into which the blade is to be divided. As a general example of the magnitude of the orifice size, the stator blades considered in this report with a perimeter of 6.20 inches and a span of 3.75 inches for an engine compressor pressure ratio of 4, a flight Mach number of 1.0, a turbine-inlet temperature of 2500° F, and the turbine blade divided into five compartments (one at the leading edge and two along each side) would require that the smallest orifice on the suction surface near the trailing edge have a diameter of approximately 0.100 inch. At the leading edge no orifice would be required and the orifices in the other compartments would range up to about 0.150 inch in diameter.

2597

Effect of orifices at blade base on wall permeabilities and peripheral temperature distribution. - The peripheral permeability variations for the three stator blade designs considered in figures 15 and 16 are shown in figure 17. The range of permeability variations for the rotor blades is given in table III. The use of orifices in the base to meter air to local positions around the periphery of blades designed for a constant pressure drop through the porous wall and the use of low pressure drops require higher permeabilities than are required for any other type of blade design, as is shown by comparing figures 13 and 17. The range of permeabilities shown in figure 17 can be obtained with 20X200 mesh stainless-steel wire cloth for a range of thickness reductions from about 30 to 40 percent. The blade design using orifices requires the highest permeability on the suction surface of the blade, whereas all other designs require the lowest permeability on the suction surface of the blade because the pressure level on the outside surface of the blades is lowest on the suction surface. When a low pressure drop through the blade wall is obtained by use of orifices for partly metering the air, the difference in squares of the pressures on the inside and outside of the wall is lower on the suction surface than at any other position on the blade; therefore high permeability is required in order to pass the air flow. For ordinary blades without orifices in the blade base, the difference in the squares of the pressures on the inside and outside of the wall is higher on the suction surface than at any other position, and a low permeability is therefore required to meter the air adequately.

As a general rule the manufacture of materials having a relatively high permeability is simpler, and the permeability is more easily controlled than it is for the manufacture of materials having lower permeabilities; however, the blades designed for a specified pressure drop through the blade wall still require a permeability variation around the periphery. This variation can be eliminated, as previously discussed, by proper choice of the orifice sizes at the blade base. In order to insure good cooling performance, a high permeability is required; yet the permeability must not be so high that poor distribution is obtained along the blade span. The permeabilities arbitrarily chosen for the constant permeability blades investigated herein were somewhat higher than the mean values of the permeability variations around the blades designed for a constant pressure drop through the blade wall of 2 pounds per square inch at sea level. For a turbine-inlet temperature of 2500° F, the permeability $12K'/\tau$ was taken as 45×10^{-9} inches for the stator blades and 35×10^{-9} inches for the rotor blades.

The use of orifices for partly metering the cooling air greatly reduces the blade temperature varaitions as well as the quantity of cooling air required at off-design altitudes. Calculations for blades designed for a constant pressure drop through the blade wall show that the maximum variation in temperature around the blade periphery occurs 2597
at a gas temperature of 2500° F; the variation was 62° F for the stator blade and 98° F for the rotor blade. The maximum temperature variation for blades designed for a constant permeability around the periphery was 33° F for the rotor blade. For the stator blade, the maximum variation was 54° F except for one case at an altitude of 50,000 feet where the pressure surface near the leading edge dropped 350° F below the temperature level of the rest of the blade. This singular case could be corrected by a better choice of permeabilities or by a slightly different method of blade design. For the case considered no orifice was used at this location because it was the blade location that had the highest pressure on the outside surface. Without an orifice at this location it was possible to operate at lower coolant supply pressures. Because of the flow variations that can sometimes occur with small pressure changes through porous walls, however, as was demonstrated for blades without metering orifices, this type of design can be subject to wide temperature variations in the portion of the blade where no metering orifice is used. It may therefore be wiser to design blades to use metering orifices in all locations of the blade, even though the coolant supply pressure required would be somewhat higher. Since this difficulty arises at the leading edge of the blade only, the effect on coolant flow requirements is quite small. If large temperature gradients are considered serious, the extra orifice and higher coolant supply pressures would be advisable.

Generalization of Results

The results shown in figures 5 to 17 and in tables I to III are all for the specific conditions outlined in the CALCULATION PROCEDURE, but practically all the trends shown can be generalized to any engine that has turbine stator and rotor designs that will produce pressure distributions similar to those shown in figure 4. The method of calculation outlined in the CALCULATION PROCEDURE should be applicable to most engines. The sensitivity of the coolant flow requirements to the blade design necessitates that any engine design that will use transpiration-cooled turbine blades have the blades designed with consideration given to both the engine conditions and the type of flight plan to which it is anticipated the engine will be subjected. It was found, however, that blade designs utilizing multiple orifices at the blade base for partly metering the cooling air permitted operation over a wider range of flight conditions with less expenditure of cooling air than blade designs not using orifices. The results shown in this report indicate the type of result that can be expected for other engines and also indicate the great advantages of blades utilizing multiple orifices at the blade base both from cooling and fabrication considerations.

2597

CONCLUDING REMARKS

An analytical method has been developed for evaluating the effects of flight speed, flight altitude, engine compressor pressure ratio, and turbine-inlet temperature on cooling-air flow and cooling-air supply pressure requirements for transpiration-cooled stator and rotor turbine blades.

Transpiration-cooled turbine blades without metering orifices may require excessive quantities of cooling air at off-design conditions because of nonuniformity of the blade temperature distributions, but proper choice of design conditions helps to reduce this effect. For example, turbine stator and rotor blades designed for a sea-level altitude and a turbine-inlet temperature of 1600° F would require a combined coolant flow ratio of 0.136 at an altitude of 50,000 feet as compared with the ideal value of 0.039 (resulting in an excess of about 250 percent) for cooling the blades to a maximum temperature of 1000° F. On the other hand, if the blade had been designed for an altitude of 50,000 feet, the maximum excess required at any other altitude would be about 60 percent.

For engines that must operate over a range of gas temperatures from 1600° to 2500° F, the blades must usually be designed for the higher gas temperature, but excessive coolant flows are required at the lower temperature. For the design considered, the combined coolant flow ratio for rotor and stator blades designed for a turbine-inlet temperature of 2500° F was as high as 0.115 when the blades were operating at a turbine-inlet temperature of 1600° F, as compared with a value of 0.052 for blades designed for 1600° F.

The use of multiple orifices at the blade base for partial metering of the cooling air to local chordwise positions on the blade periphery can reduce the excess cooling air required at off-design altitude conditions to a maximum of about 10 percent. At the same time the orifices permit the fabrication of blades with a uniform permeability around the periphery, which will greatly simplify fabrication.

For all blade designs considered, the engine compressor can supply air at sufficient pressure for cooling the rotor blades provided they are properly designed, but a boost compressor is required for the stator blades for most operating conditions.

For high engine compressor pressure ratios (of the order of 10 or higher), cooling-air aftercoolers are desirable for most operating contitions, and for some conditions they are absolutely necessary.

The permeability range required for transpiration-cooled gas-turbine blades can be obtained from single thicknesses of brazed and calendered stainless-steel wire cloth, but extreme care will be required in the fabrication of blades without metering orifices in order to provide the permeability variations around the blade periphery that will 2597
result in economical expenditures of cooling air.

The trends shown in this report can probably be generalized for most engines, but the sensitivity of coolant flow requirements to blade design necessitates that any engine design using transpiration-cooled turbine blades have the blades designed with consideration given to both the engine conditions and the type of flight plan to which it is anticipated the engine will be subjected.

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

APPENDIX A

SYMBOLS

The following symbols are used in this report:

A area, sq ft

2597

B flow coefficient for flow nozzle

C constant

c_p specific heat at constant pressure, Btu/(lb)(°F)

f,F indicate function of

g acceleration due to gravity, ft/sec^2

H' gas-to-surface heat-transfer coefficient for solid surface, Btu/(°F)(sq ft)(sec)

h pressure drop through orifice, lb/sq ft

K permeability coefficient, sq ft, for $\rho_a V_a = 0 \left(K = \frac{1}{\alpha}\right)$

K' permeability coefficient, sq ft; for $p_a^2 - p_{g,B}^2 = 3556^2 - 2116^2$, lb^2/ft^4

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

M Mach number

m $\left(\frac{\rho_a V_a T_{B,g}}{\rho_{g,B} V_{g,B} T_{g,B}}\right)(71.3)(Re_{B,g})^{0.1}(Pr_g)^{2/3}$

Pr Prandtl number, $c_p\mu/k$

p static pressure, lb/sq ft absolute

p' total pressure, lb/sq ft absolute

R gas constant, ft-lb/(lb)(°F)

$Re_{B,g}$ Reynolds number with fluid properties based on blade wall temperature, $\rho_{B,g}V_{g,B}x/g\mu_{B,g}$

$Re_{g,B}$ Reynolds number with fluid properties based on gas temperature, $\rho_{g,B}V_{g,B}x/g\mu_{g,B}$

r ratio of velocities in boundary layer (equation (C3) or (C7)) 2597

T static temperature, $^{\circ}R$

T' total temperature, $^{\circ}R$

V velocity, ft/sec

v flow volume per unit area and time, ft/sec

w weight flow rate, lb/sec

x distance from blade leading edge, ft

α flow coefficient in equation (3), 1/(sq ft)

β flow coefficient in equation (3), 1/ft

γ ratio of specific heats

δ stator discharge angle from plane of rotation, deg

η compressor adiabatic efficiency

Λ local temperature recovery factor, assumed equal to 0.89

μ absolute viscosity, lb-sec/(sq ft)

ρ density, lb/(cu ft)

τ thickness, ft

φ $\rho_a c_p v_a/H'$

ψ total-pressure recovery factor (equation (B2))

Subscripts:

A engine air

a cooling air or surface through which cooling air is passing

B blade (local conditions) or refers to blade wall temperature[1]

e effective

ex exit

F used with M to denote flight Mach number

2597

g combustion gas or gas side[1]

i inlet

m mean

n flow nozzle

O for reference temperature of 518.4° R

r rotor

s stator

0,1,2,3,4,5 stations in engine (see fig. 1)

---

[1]The subscript B,g refers to a local condition based on the blade wall temperature at the surface next to the gas stream; the subscript g,B refers to a local condition in the gas stream just outside the boundary layer around the blade.

APPENDIX B

TEMPERATURE AND PRESSURE LEVELS THROUGH ENGINE

The equations used for calculation of the temperature and pressure levels through the engine are standard equations used for cycle analysis and will be listed without detailed explanation.

Total pressure at engine diffuser inlet. - 2597

$$p_1' = p_0\left(1 + \frac{\gamma_A - 1}{2} M_F^2\right)^{\frac{\gamma_A}{\gamma_A - 1}} \tag{B1}$$

Total pressure at engine diffuser outlet. -

$$p_2' = \psi(p_1' - p_0) + p_0 \tag{B2}$$

Total temperature at engine diffuser outlet. -

$$T_2' = T_0\left(1 + \frac{\gamma_A - 1}{2} M_F^2\right) \tag{B3}$$

Total temperature at compressor discharge. -

$$T_3' = T_2'\left[1 + \frac{\left(\dfrac{p_3'}{p_2'}\right)^{\frac{\gamma_A - 1}{\gamma_A}} - 1}{\eta}\right] \tag{B4}$$

where p_3'/p_2' is compressor pressure ratio.

Cooling-air temperature. - In the coolant system, the velocities are assumed low enough that the velocity head can be neglected. With this assumption, the coolant temperature can be calculated from

$$T_a = T_a' = T_2'\left[1 + \frac{\left(\frac{p_a}{p_2'}\right)^{\frac{\gamma_a-1}{\gamma_a}} - 1}{\eta}\right] \tag{B5}$$

2597

Total pressure at stator inlet. - It is assumed that there is a 3-percent pressure loss through the burners; therefore

$$p_4' = 0.97\ p_3' \tag{B6}$$

Velocity, temperature, and pressure distributions over stator blades. - The total temperature at the stator inlet T_4' was an assumed value. The assumed value of T_4', p_4' from equation (B6), and the assumption that the gas flow through the stator blades was choked at the throat $(M = 1.0)$ can be used to calculate the velocity distribution over the stator blades by means of the stream filament theory of references 6 and 7. With these calculated velocities the Mach number distribution can be calculated from

$$M_{g,B,s} = \frac{1}{\sqrt{\frac{\gamma_g gRT_4'}{(V_{g,B,s})^2} - \frac{\gamma_g-1}{2}}} \tag{B7}$$

Inasmuch as the gas flow through the stator blades is choked, which is the normal operating condition for gas-turbine engines, the Mach number distribution will remain constant. These local Mach numbers can then be used for calculating local temperatures, pressures, densities, and velocicites for large ranges of stator inlet temperature and pressure using the following equations:

$$T_{g,B,s} = \frac{T_4'}{1 + \frac{\gamma_g-1}{2}(M_{g,B,s})^2} \tag{B8}$$

$$p_{g,B,s} = \frac{p_4'}{\left[1 + \frac{\gamma_g-1}{2}(M_{g,B,s})^2\right]^{\frac{\gamma_g}{\gamma_g-1}}} \tag{B9}$$

$$\rho_{g,B,s} = \frac{p_{g,B,s}}{RT_{g,B,s}} \qquad \text{(B10)}$$

$$V_{g,B,s} = M_{g,B,s}\sqrt{\gamma_g gRT_{g,B,s}} \qquad \text{(B11)}$$

In equation (B9) the ratio $p_{g,B,s}/p_4'$ is a function of Mach number only for a constant value of γ_g, and since the Mach number distribution remains constant for choked flow through the stator blades the distribution of $p_{g,B,s}/p_4'$ also remains constant and can be plotted as shown in figure 4(a) for a typical stator blade configuration. 2597

The local effective gas temperature of the stator blades is

$$T_{g,e,s} = T_4'\left\{1 - (1-A)\left[1 - \frac{1}{1 + \frac{\gamma_g - 1}{2}(M_{g,B,s})^2}\right]\right\} \qquad \text{(B12)}$$

Velocity, temperature, and pressure distributions over rotor blades. - The total temperature and pressure of the gas relative to the rotor blades are considerably reduced from the values relative to the stator blade because of the rotation of the rotor. The gas static temperature at the stator exit and the rotor inlet is the same; therefore

$$T_5 = \frac{T_4'}{1 + \frac{\gamma_g - 1}{2}(M_{g,s,ex})^2} \qquad \text{(B13)}$$

where the Mach number at the stator exit $M_{g,s,ex}$ is determined by the turbine design and will be considered constant for choked flow through the stator.

The velocity at the stator exit is

$$V_{g,s,ex} = M_{g,s,ex}\sqrt{\gamma_g gRT_5} \qquad \text{(B14)}$$

With the use of vectors, the velocity relative to the rotor at the rotor inlet is found to be

$$V_{g,r,i} = \sqrt{V_r^2 + (V_{g,s,ex})^2 - 2V_rV_{g,s,ex}\cos\delta} \qquad \text{(B15)}$$

where V_r is the tangential velocity of the rotor blades.

The Mach number relative to the rotor at the rotor inlet is

$$M_{g,r,i} = \frac{V_{g,r,i}}{\sqrt{\gamma_g g R T_5}} \tag{B16}$$

The total temperature relative to the rotor at the rotor inlet is

2597

$$T_5' = T_5\left[1 + \frac{\gamma_g - 1}{2}\,(M_{g,r,i})^2\right] \tag{B17}$$

The total pressure relative to the rotor at the rotor inlet is calculated in a manner similar to that in which the total temperature relative to the rotor was calculated. The static pressures at the stator exit and at the rotor inlet are the same

$$p_5 = \frac{p_4'}{\left[1 + \frac{\gamma_g - 1}{2}\,(M_{g,s,ex})^2\right]^{\frac{\gamma_g}{\gamma_g - 1}}} \tag{B18}$$

The total pressure relative to the rotor is then calculated from the static pressure given by equation (B18) and the rotor inlet Mach number given by equation (B16)

$$p_5' = p_5\left[1 + \frac{\gamma_g - 1}{2}\,(M_{g,r,i})^2\right]^{\frac{\gamma_g}{\gamma_g - 1}} \tag{B19}$$

The velocity distribution over the turbine rotor blades can be calculated by the stream filament theory of references 6 and 7 similar to the method used for stator blades, except that the flow rate is determined by the flow through the stator blades. The total temperature T_5' and the total pressure p_5' are obtained from equations (B17) and (B19), respectively. The Mach number distribution can be calculated from

$$M_{g,B,r} = \frac{1}{\sqrt{\frac{\gamma_g g R T_5'}{(V_{g,B,r})^2} - \frac{\gamma_g - 1}{2}}} \tag{B20}$$

In this report the Mach number distribution around the rotor blades is assumed to remain constant for choked flow in the stator (effect of angle of attack neglected). The local values of gas temperature, pressure, density, and velocity for large ranges of rotor-inlet temperatures and pressures can be calculated in a manner similar to that used for the stator blades:

$$T_{g,B,r} = \frac{T_5'}{1 + \frac{\gamma_g - 1}{2}\left(M_{g,B,r}\right)^2} \tag{B21}$$ 2597

$$p_{g,B,r} = \frac{p_5'}{\left[1 + \frac{\gamma_g - 1}{2}\left(M_{g,B,r}\right)^2\right]^{\frac{\gamma_g}{\gamma_g - 1}}} \tag{B22}$$

$$\rho_{g,B,r} = \frac{p_{g,B,r}}{RT_{g,B,r}} \tag{B23}$$

$$V_{g,B,r} = M_{g,B,r}\sqrt{\gamma_g gRT_{g,B,r}} \tag{B24}$$

Observation of equations (B13) to (B22) leads to a relation of the local pressures around the rotor blade $p_{g,B,r}$ to the pressure ahead of the stator p_4' as a function of the local Mach numbers around the rotor blade $M_{g,B,r}$ and as a function of the turbine-inlet total temperature T_4' when the Mach number at the exit of the stator $M_{g,s,ex}$ is a constant. The distribution of $p_{g,B,r}/p_4'$ can therefore be plotted as shown in figure 4(b) for various turbine-inlet temperatures.

The local effective gas temperature at the rotor blades is

$$T_{g,e,r} = T_5'\left\{1 - (1-\Lambda)\left[1 - \frac{1}{1 + \frac{\gamma_g - 1}{2}\left(M_{g,B,r}\right)^2}\right]\right\} \tag{B25}$$

APPENDIX C

MODIFICATION OF TURBULENT FLOW EQUATION OF FRIEDMAN

The local ideal coolant flow rates necessary to obtain uniform cooling of the blade surface can be calculated from the turbulent flow equation of Friedman (reference 5). In the nomenclature of the present report the equation is

2597

$$\frac{T_{B,g} - T_a}{T_{g,e} - T_a} = \frac{r}{e^{r\varphi} + r - 1} \tag{C1}$$

In reference 1 the values of φ and r are given as

$$\varphi_{g,B} = \frac{\rho_a c_p V_a}{H'} = \frac{(Re_{g,B})^{1/5}(Pr_g)^{2/3}}{0.0296} \frac{\rho_a V_a}{\rho_{g,B} V_{g,B}} \tag{C2}$$

and

$$r_{g,B} = \frac{2.11}{(Re_{g,B})^{0.1}} \tag{C3}$$

The effect of temperature level on the Prandtl number is small so that a mean value of Prandtl number can be used for all cases.

If the density and viscosity in the Reynolds number are based on blade wall temperature in order to better correlate heat-transfer data over a large range of $T_{B,g}/T_{g,B}$ (see references 2 and 13), equation (C2) takes the form

$$\varphi_{B,g} = \frac{(Re_{B,g})^{1/5}(Pr_g)^{2/3}}{0.0296} \frac{\rho_a V_a}{\rho_{B,g} V_{g,B}} \tag{C4}$$

where $\rho_{B,g}$ is the local gas density based on wall temperature and defined as

$$\rho_{B,g} = \frac{p_{g,B}}{RT_{B,g}} \tag{C5}$$

Equation (C4) can also be written

$$\varphi_{B,g} = \frac{(Re_{B,g})^{1/5}(Pr_g)^{2/3}}{0.0296} \frac{\rho_a v_a}{\rho_{g,B} V_{g,B}} \frac{T_{B,g}}{T_{g,B}} \tag{C6}$$

The magnitude of the Reynolds number has a very small effect on the value of r (equation (C3)) because of the small exponent; therefore it should be permissible to define 2597

$$r_{B,g} = \frac{2.11}{(Re_{B,g})^{0.1}} \tag{C7}$$

Equations (C1), (C6), and (C7) can be combined and written as

$$\rho_a v_a = \frac{\ln\left[1 - r + \frac{r}{\frac{T_{B,g} - T_a}{T_{g,e} - T_a}}\right]}{71.3(Re_{B,g})^{0.1}(Pr_g)^{2/3}} \left(\frac{\rho_{g,B} V_{g,B} T_{g,B}}{T_{B,g}}\right) \tag{1}$$

REFERENCES

1. Eckert, E. R. G., and Esgar, Jack B.: Survey of Advantages and Problems Associated with Transpiration Cooling and Film Cooling of Gas-Turbine Blades. NACA RM E50K15, 1951.

2. Schafer, Louis, J., Jr., Bartoo, Edward R., and Richards, Hadley T.: Experimental Investigation of the Heat-Transfer Characteristics of an Air-Cooled Sintered Porous Turbine Blade. NACA RM E51K08, 1952.

3. Bartoo, Edward R., Schafer, Louis J., Jr., and Richards, Hadley T.: Experimental Investigation of Coolant-Flow Characteristics of a Sintered Porous Turbine Blade. NACA RM E51K02, 1952.

4. Ellerbrock, Herman H., and Schafer, Louis J., Jr.: Application of Blade Cooling to Gas Turbines. NACA RM E50A04, 1950.

5. Friedman, Joseph: A Theoretical and Experimental Investigation of Rocket-Motor Sweat Cooling. Jour. Am. Rocket Soc., no. 79, Dec. 1949, pp. 147-154.

6. Huppert, M. C., and MacGregor, Charles: Comparison Between Predicted and Observed Performance of Gas-Turbine Stator Blade Design for Free-Vortex Flow. NACA TN 1810, 1949.

7. Hubbartt, James E., and Schum, Eugene F.: Average Outside-Surface Heat-Transfer Coefficients and Velocity Distributions for Heated and Cooled Impulse Turbine Blades in Static Cascades. NACA RM E50L20, 1951.

8. Eckert, E. R. G., and Livingood, J. N. B.: Method for Calculation of Heat Transfer in Laminar Region of Air Flow Around Cylinders of
2597 Arbitrary Cross Section (Including Large Temperature Differences and Transpiration Cooling). NACA TN 2733, 1952.

9. Donoughe, Patrick L., and McKinnon, Roy A.: Experimental Investigation of Air-Flow Uniformity and Pressure Level on Wire Cloth for Transpiration-Cooling Applications. NACA RM E52E16, 1952.

10. Eckert, E. R. G., Kinsler, Martin R., and Cochran, Reeves P.: Wire Cloth as Porous Material for Transpiration-Cooled Walls. NACA RM E51H23, 1951.

11. Spooner, Robert B.: Effect of Heat-Capacity Lag on a Variety of Turbine-Nozzle Flow Processes. NACA TN 2193, 1950.

12. Esgar, Jack B., and Lea, Alfred L.: Determination and Use of the Local Recovery Factor for Calculating the Effective Gas Temperature for Turbine Blades. NACA RM E51G10, 1951.

13. Humble, Leroy V., Lowdermilk, Warren H., and Desmon, Leland G.: Measurements of Average Heat-Transfer and Friction Coefficients for Subsonic Flow of Air in Smooth Tubes at High Surface and Fluid Temperatures. NACA Rep. 1020, 1951. (Supersedes NACA RM's E7L31, E8L03, E50E23, E50H23.)

TABLE I - SUMMARY OF COOLANT FLOW AND COOLANT COMPRESSOR PRESSURE RATIO REQUIREMENTS FOR RANGES OF ENGINE AND FLIGHT CONDITIONS FOR BLADES WITHOUT METERING ORIFICES

[Unless otherwise noted, cooling-air temperature was at coolant compressor discharge temperature, coolant was supplied to complete blade periphery at constant pressure without metering, and maximum blade temperature was 1000° F]

Flight and engine conditions				Blade design conditions			Stator blades		Rotor blades		Remarks
Altitude (ft)	Flight Mach number	Engine compressor pressure ratio	Turbine inlet temperature (°F)	Altitude (ft)	Flight Mach number	Turbine inlet temperature (°F)	p_s/p_2'	w_a/w_g	p_s/p_2'	w_a/w_g	
0	1.0	4	1600	0	1.0	1600	3.97	0.024	2.45	0.008	
25,000							5.51	.051	3.58	.014	
50,000							10.81	.111	7.05	.035	
0				25,000			4.05	.029	2.28	.011	
25,000							4.28	.024	2.58	.009	
50,000							7.99	.086	4.70	.018	
0				50,000			4.04	.059	2.18	.017	
25,000							4.50	.059	2.55	.013	
50,000							5.38	.029	3.08	.010	
0				50,000		2500	3.89	.080	2.38	.055	
25,000							4.00	.063	2.38	.052	
50,000							4.50	.050	2.38	.019	
0				50,000		1600	4.02	.032	——	——	Maximum blade temperature was 1100° F instead of 1000° F
25,000							4.29	.029	——	——	
50,000							5.38	.021	——	——	
0				0			3.97	.017	——	——	
25,000				25,000			4.28	.017	——	——	
0	0			0			4.47	.034	2.93	.011	
	.5						4.25	.030	2.79	.010	
	.8						4.00	.024	——	——	
	0				0		4.12	.022	2.52	.008	
	.5				.5		4.07	.022	2.49	.008	
	.8				.8		4.01	.022	——	——	
0	1.0		2500	0	1.0	2500	3.97	.057	2.68	.038	
25,000							4.26	.080	3.25	.048	
50,000							7.45	.204	5.66	.089	
0				50,000			4.18	.022	2.58	.030	
25,000							4.38	.085	2.58	.031	
50,000							5.38	.079	3.22	.043	
25,000				25,000			4.28	.086	2.78	.058	

NACA

TABLE I - SUMMARY OF COOLANT FLOW AND COOLANT COMPRESSOR PRESSURE RATIO REQUIREMENTS FOR RANGES OF ENGINE AND FLIGHT CONDITIONS FOR BLADES WITHOUT METERING ORIFICES - Concluded

[Unless otherwise noted, cooling-air temperature was at coolant compressor discharge temperature, coolant was supplied to complete blade periphery at constant pressure without metering, and maximum blade temperature was 1000° F]

Flight and engine conditions				Blade design conditions			Stator blades		Rotor blades		Remarks
Altitude (ft)	Flight Mach number	Engine compressor pressure ratio	Turbine inlet temperature (°F)	Altitude (ft)	Flight Mach number	Turbine inlet temperature (°F)	p_a/p_2'	w_a/w_g	p_a/p_2'	w_a/w_g	
0	1.0	10	1600	0	1.0	1600	9.64	0.051*	6.02	0.012	Cooling-air temperature was at coolant compressor discharge temperature except maximum temperature was limited to 700° F; w_a/w_g is marked with an asterisk (*) where temperature was limited.
25,000							9.88	.053	6.80	.012	
50,000							17.20	.100*	12.62	.026	
0				50,000			9.93	.057*	5.95	.022	
25,000							10.08	.058	5.95	.020	
50,000							11.06	.055	6.59	.011	
25,000				25,000			9.95	.050	6.14	.100	
0			2500	0		2500	9.64	.077*	6.55	.046	
25,000							9.55	.172	6.78	.096	
50,000							9.64	.223	7.72	.158	
0				50,000			10.96	.085*	6.58	.049	
25,000							10.99	.095*	6.87	.056	
50,000							11.06	.086	7.10	.045	
25,000				25,000			9.95	.078	6.66	.040	
0				0			9.64	.044	6.55	.027	Ideal coolant flow requirements for a cooling air temperature of 300° F
25,000				25,000			9.95	.055	6.66	.032	
50,000				50,000			11.06	.064	7.10	.039	
0		4		0			3.97	.024	2.66	.011	Ideal coolant flow requirements for a design blade temperature of 1500° F instead of 1000° F
25,000				25,000			4.28	.026	2.78	.012	
50,000				50,000			5.38	.032	3.22	.015	

TABLE II - SUMMARY OF COOLANT FLOW AND COOLANT COMPRESSOR PRESSURE RATIO REQUIREMENTS FOR A RANGE OF FLIGHT ALTITUDES FOR BLADES WITH METERING ORIFICES AT BASE

[Cooling-air temperature was at coolant compressor discharge temperature, and maximum blade temperature was 1000° F]

Flight and engine conditions				Blade design conditions			Stator blades		Rotor blades		Remarks
Altitude (ft)	Flight Mach number	Engine compressor pressure ratio	Turbine inlet temperature (°F)	Altitude (ft)	Flight Mach number	Turbine inlet temperature (°F)	p_a/p_2'	w_a/w_g	p_a/p_2'	w_a/w_g	
0	1.0	4	1600	0	1.0	1600	3.86	0.024	2.45	0.009	Blades designed for pressure drop through wall of 2 lb/sq in.
25,000							3.93	.026	2.58	.010	
50,000							4.28	.030	3.02	.011	
0			2500			2500	3.86	.067	2.66	.035	
25,000							3.89	.073	2.78	.041	
50,000							4.11	.082	3.22	.050	
0							3.80	.067	2.60	.035	Stator blades designed for a constant permeability $12K'/\tau$ of 45×10^{-9} inch and rotor blades for a permeability of 35×10^{-9} inch
25,000							3.82	.072	2.65	.038	
50,000							3.93	.088	2.86	.045	
0			1600	0		1600	3.86	.024	2.45	.009	Ideal coolant flow and coolant compressor pressure ratio requirements
25,000				25,000			3.98	.024	2.58	.009	
50,000				50,000			4.42	.029	3.02	.010	
0			2500	0		2500	3.86	.067	2.66	.035	
25,000				25,000			3.98	.066	2.78	.036	
50,000				50,000			4.42	.079	3.22	.043	

NACA

TABLE III - PERMEABILITY RANGES REQUIRED FOR ROTOR BLADES FOR VARIOUS DESIGN CONDITIONS

[Maximum blade temperature 1000° F and flight Mach number of 1.0; permeabilities calculated for pressure drop through wall of 10 lb/sq in. discharging to NACA standard sea-level atmosphere]

Blade design conditions			Range of permeabilities around blade periphery[1] $12K'/\tau$ (in.)	Remarks
Altitude (ft)	Engine compressor pressure ratio	Turbine inlet temperature (°F)		
0	4	1600	3.07 to 5.19×10^{-9}	Transpiration-cooled blades without orifices at base
50,000	4	1600	8.40 to 10.20×10^{-9}	
50,000	4	2500	16.45 to 19.24×10^{-9}	
50,000	10	2500	13.85 to 18.05×10^{-9}	
50,000	10	1600	6.64 to 9.20×10^{-9}	
0	4	1600	14.10 to 17.95×10^{-9}	Transpiration-cooled blades with orifices at base for partially metering cooling air and designed for 2 lb/sq in. pressure drop through wall
0	4	2500	31.80 to 39.80×10^{-9}	
0	4	2500	35×10^{-9}	Constant permeability blade with metering orifices at base for partially metering cooling air

[1]This range does not include a small area directly at the blade leading edge where, for blades without metering orifices at the base, permeabilities 50 to 100 percent higher than those specified in this table would be desirable.

NACA

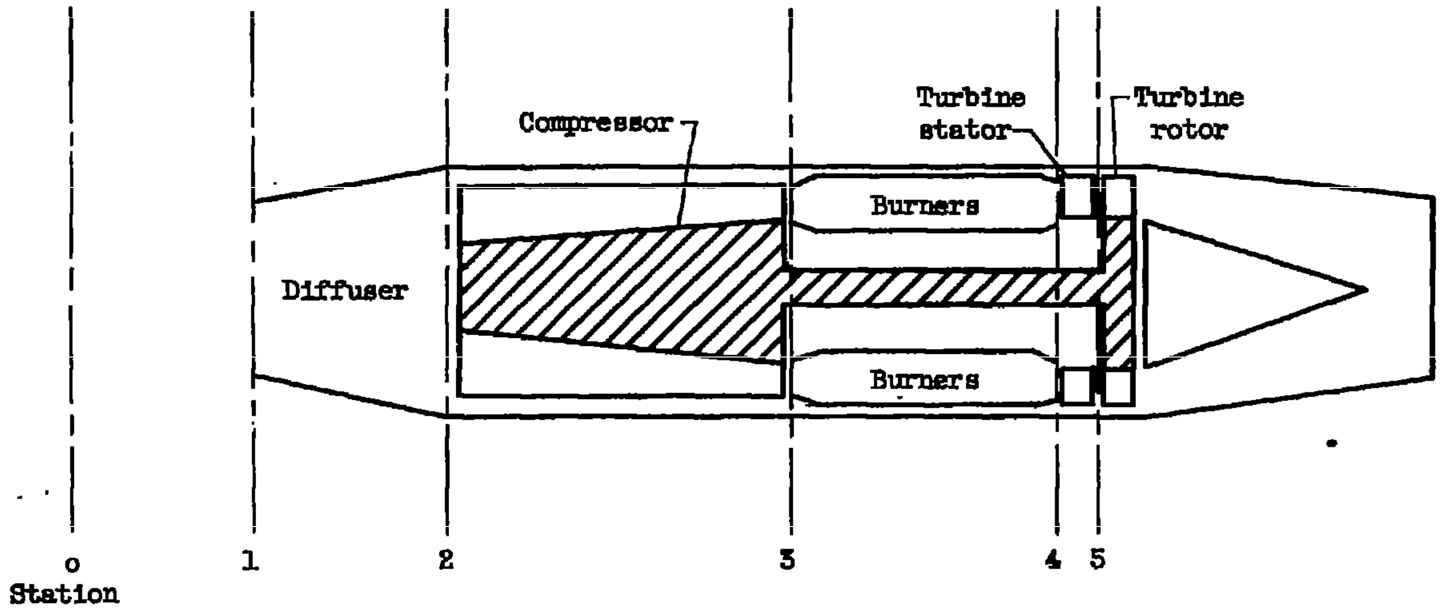


Figure 1. - Stations through engine used in analysis.

2597

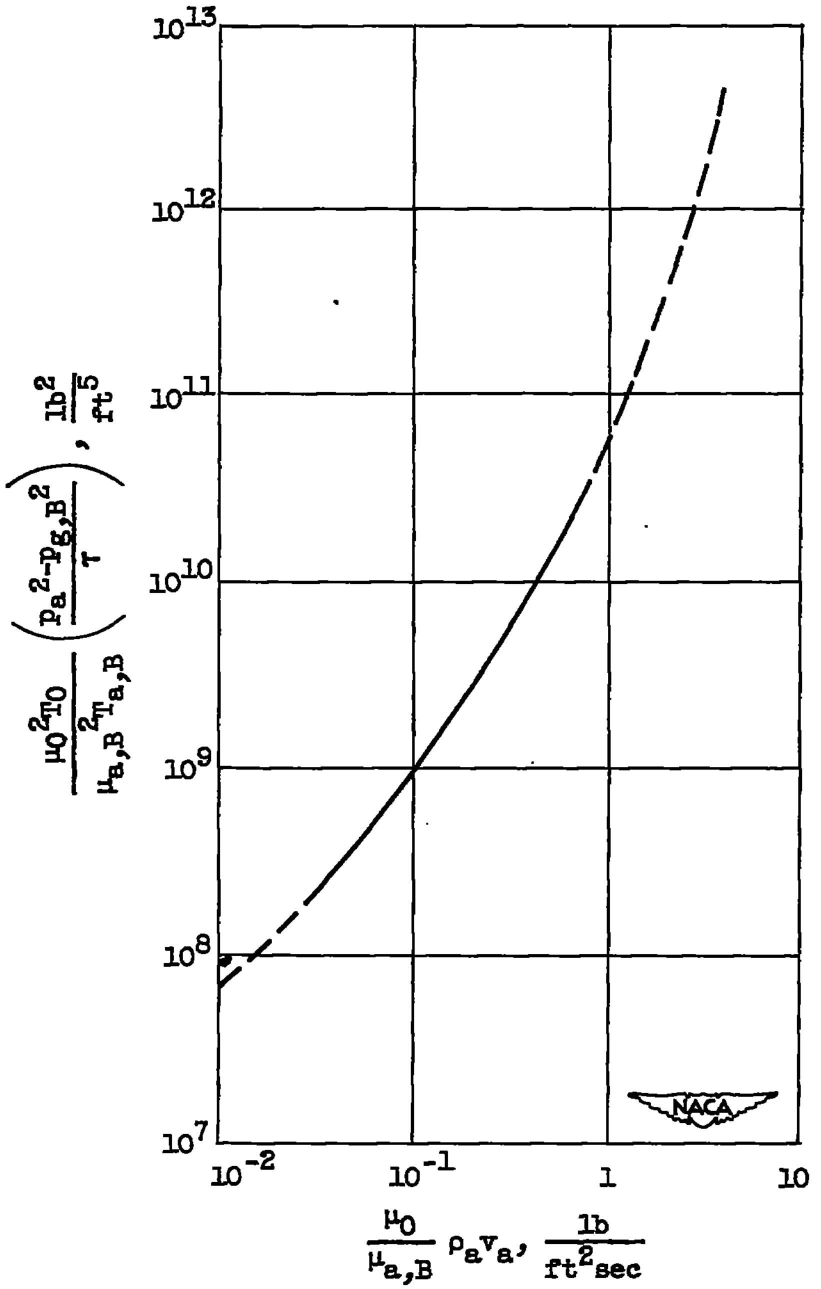


Figure 2. - Permeability curve used for transpiration-cooling analysis. Data obtained from reference 10.

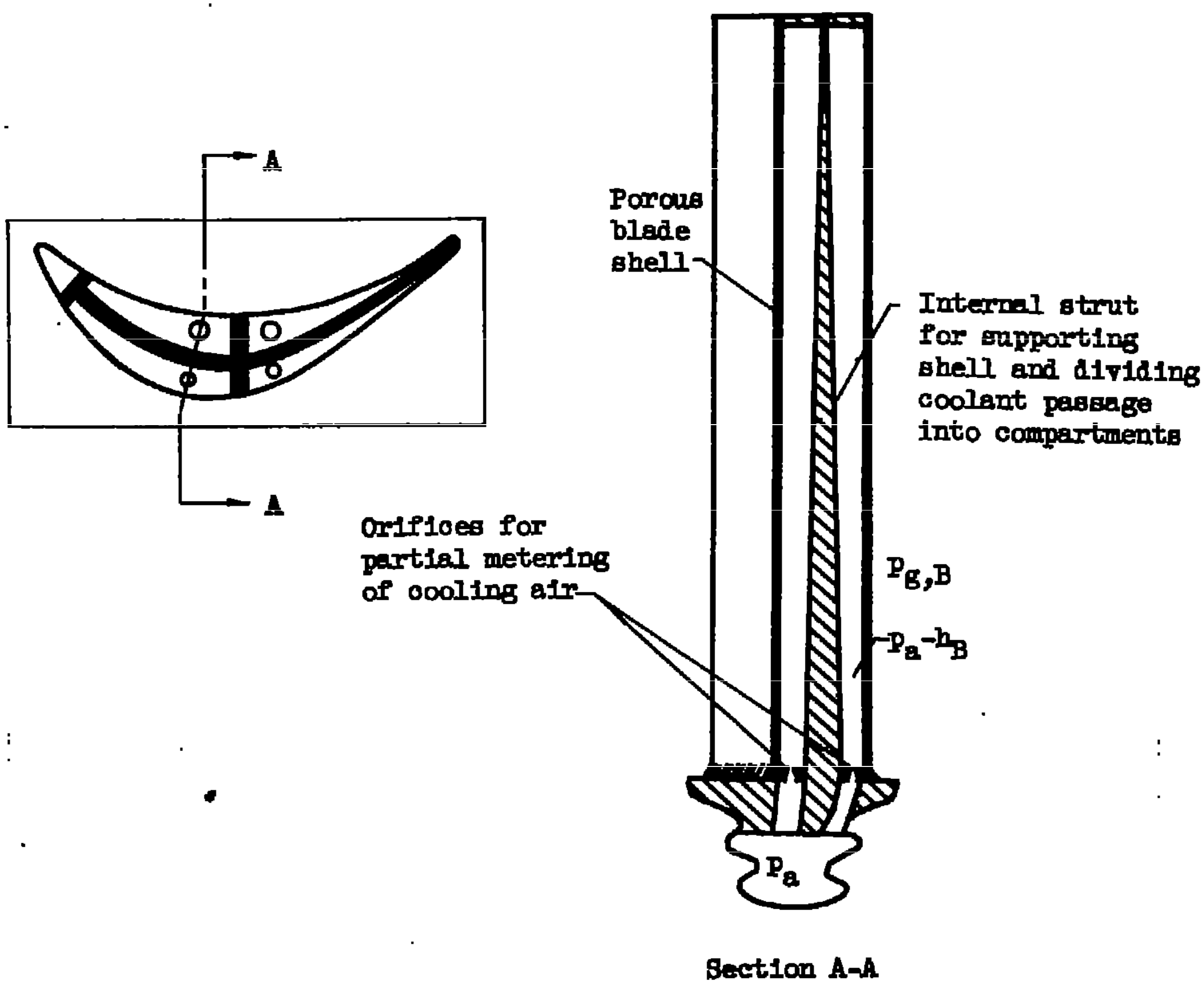
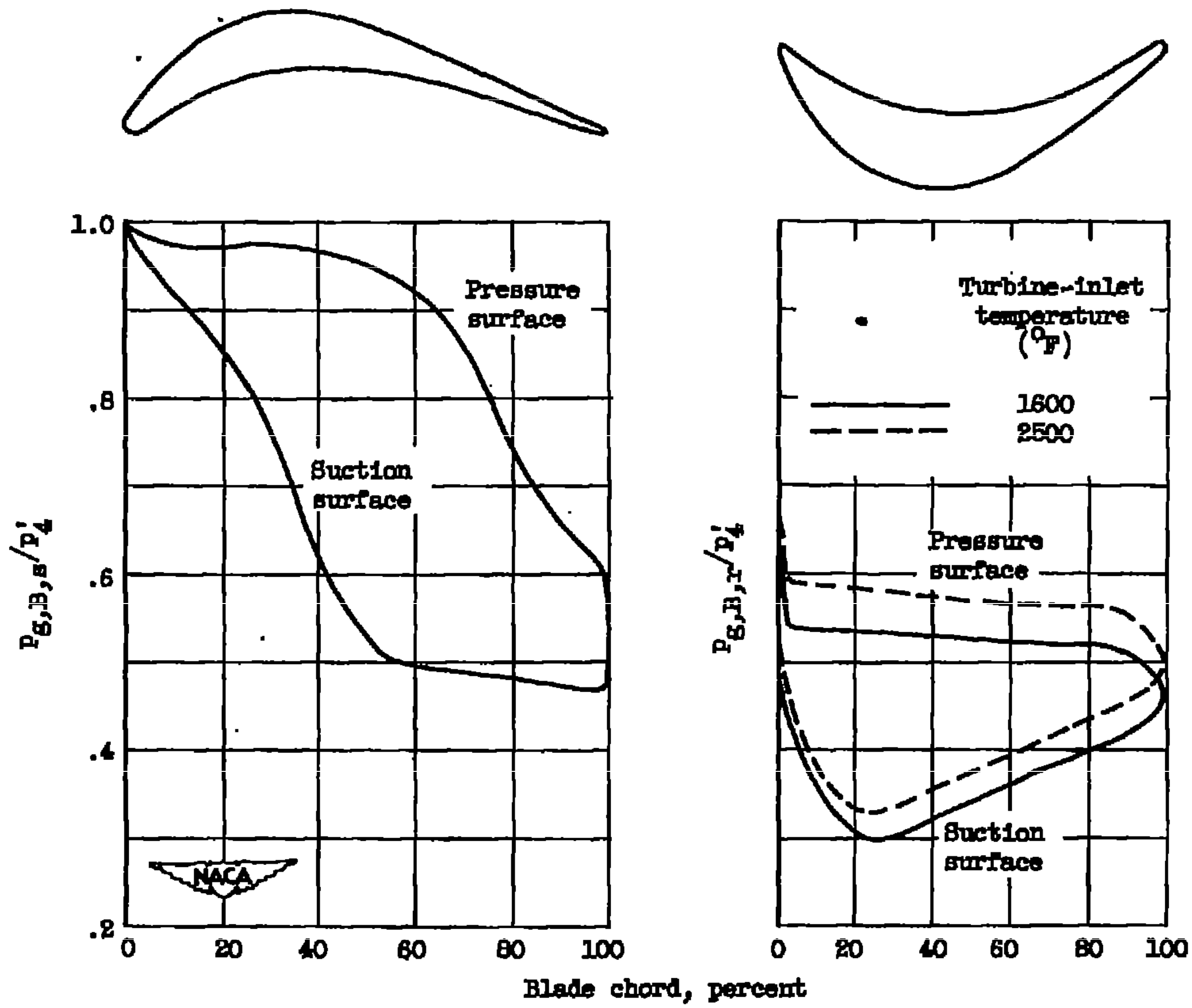


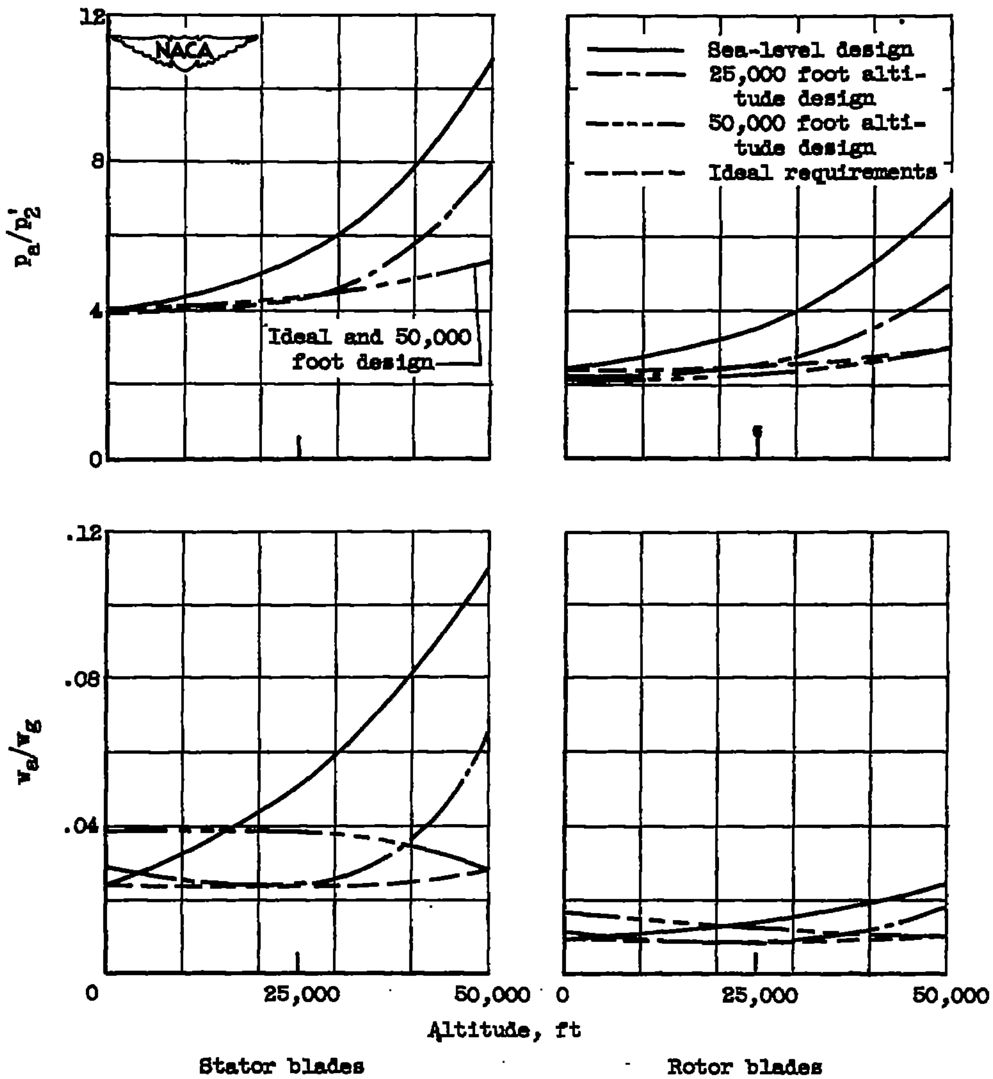
Figure 3. - Sketch of strut-supported, transpiration-cooled turbine blade utilizing multiple orifices at blade base for partial metering of cooling air.



(a) Stator blade.

(b) Rotor blade.

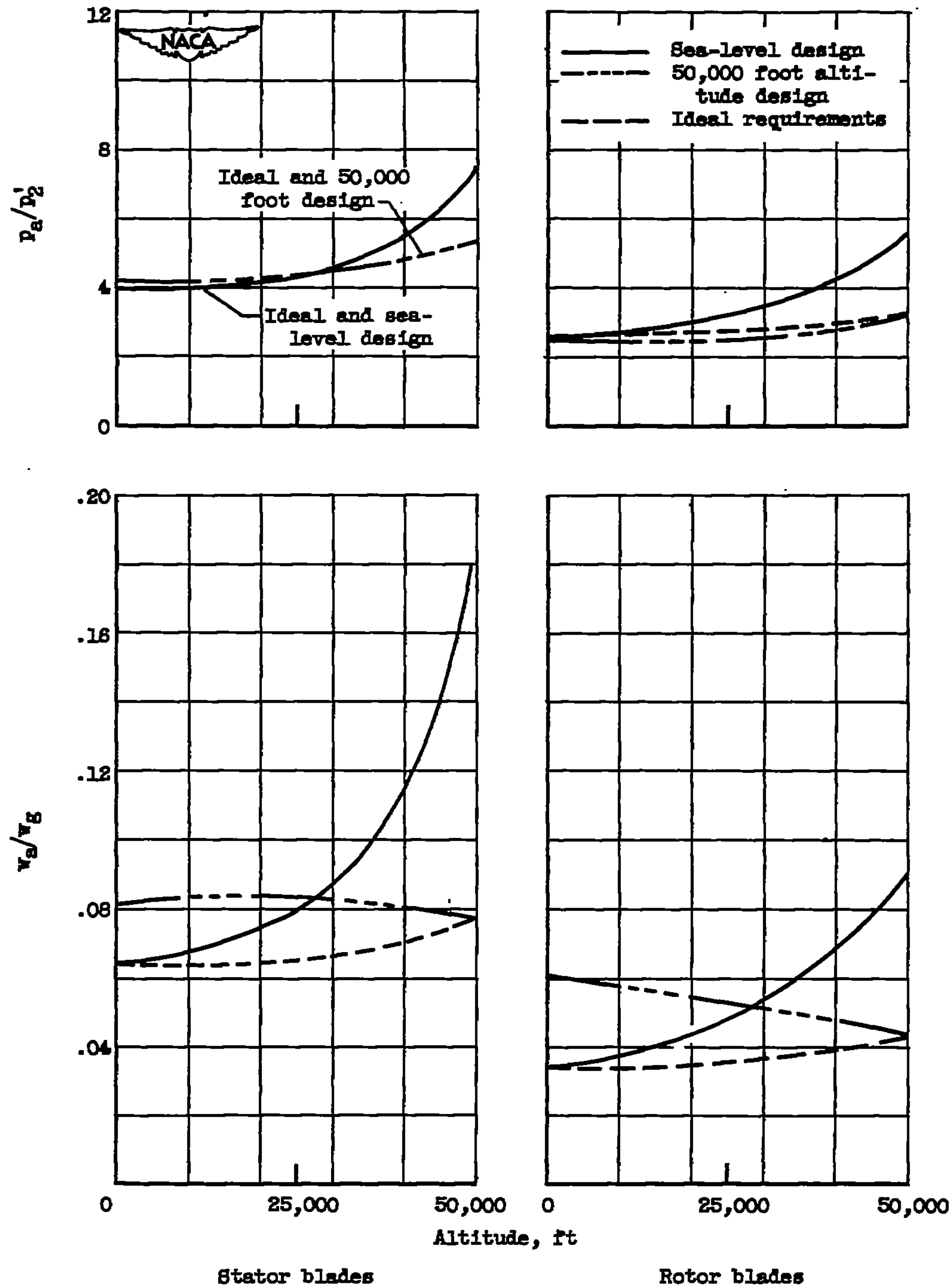
Figure 4. - Pressure distributions around stator and rotor blades.



2597

(a) Turbine-inlet temperature, 1600° F.

Figure 5. - Effect of design altitude on coolant flow and coolant compressor pressure ratio requirements at other altitudes. Maximum blade temperature, 1000° F; flight Mach number, 1.0; engine compressor pressure ratio, 4.



(b) Turbine-inlet temperature, 2500° F.

Figure 5. - Concluded. Effect of design altitude on coolant flow and coolant compressor pressure ratio requirements at other altitudes. Maximum blade temperature, 1000° F; flight Mach number, 1.0; engine compressor pressure ratio, 4.

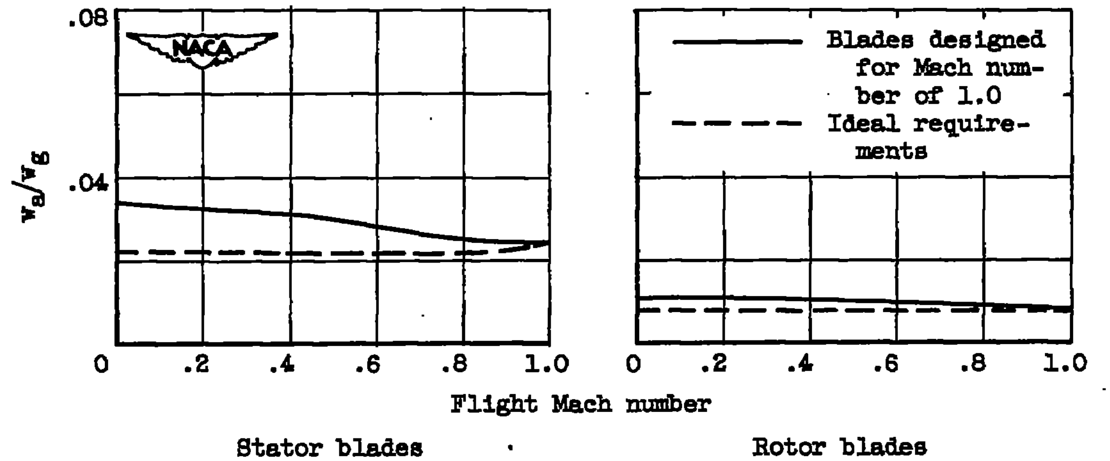


Figure 6. - Effect of design flight Mach number on coolant flow requirements at other Mach numbers. Maximum blade temperature, 1000° F; altitude, sea-level; engine compressor pressure ratio, 4; turbine-inlet temperature, 1600° F.

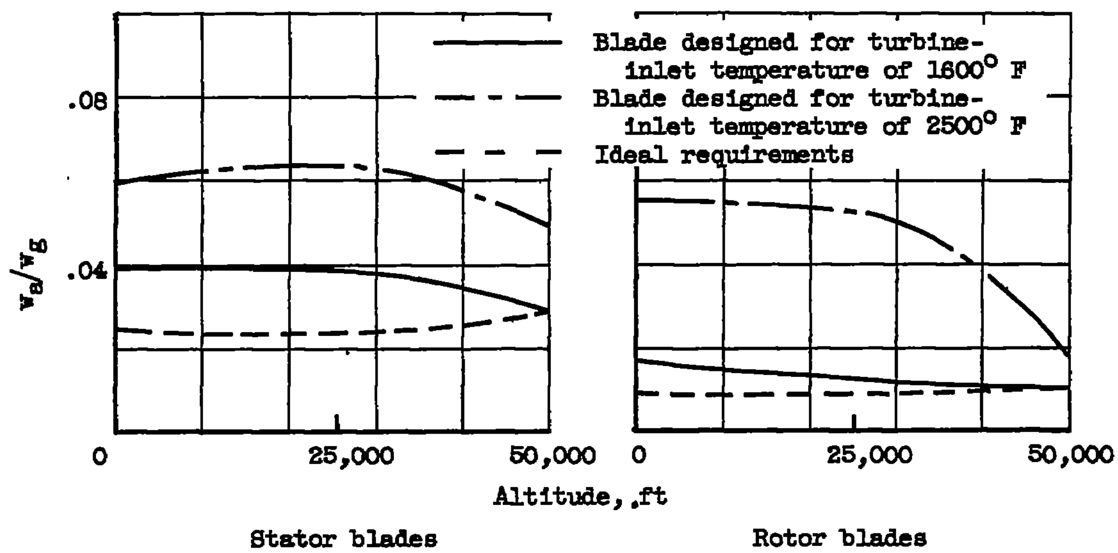
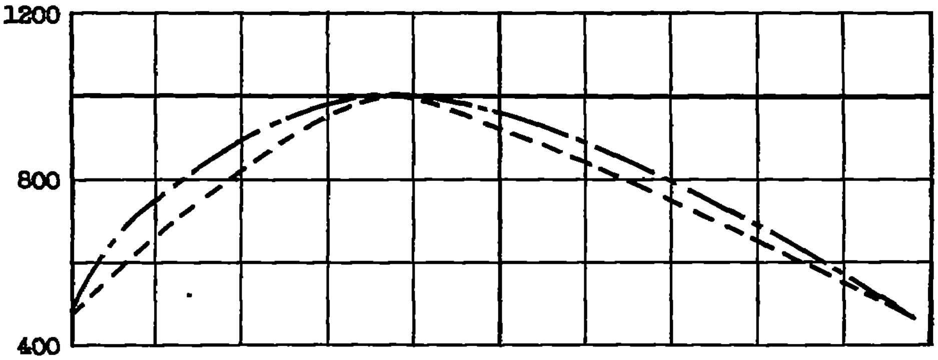
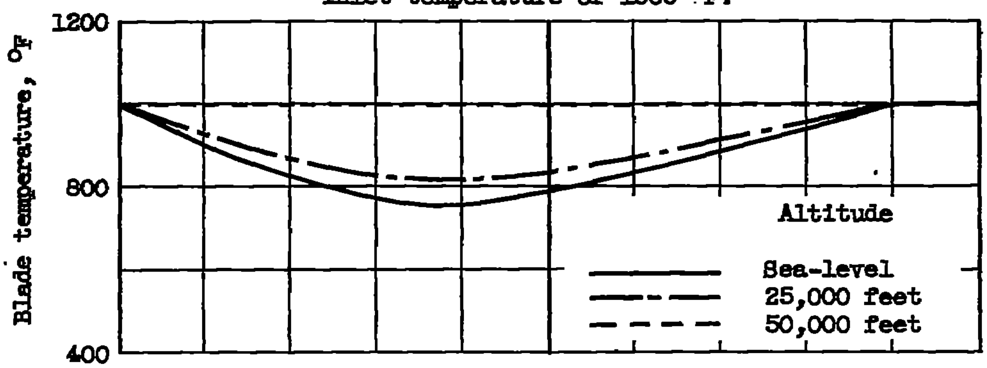


Figure 7. - Effect of design gas temperature on coolant flow requirements for blades operating at turbine-inlet temperature of 1600° F. Maximum blade temperature, 1000° F; flight Mach number, 1.0; engine compressor pressure ratio, 4; blades designed for altitude of 50,000 feet.

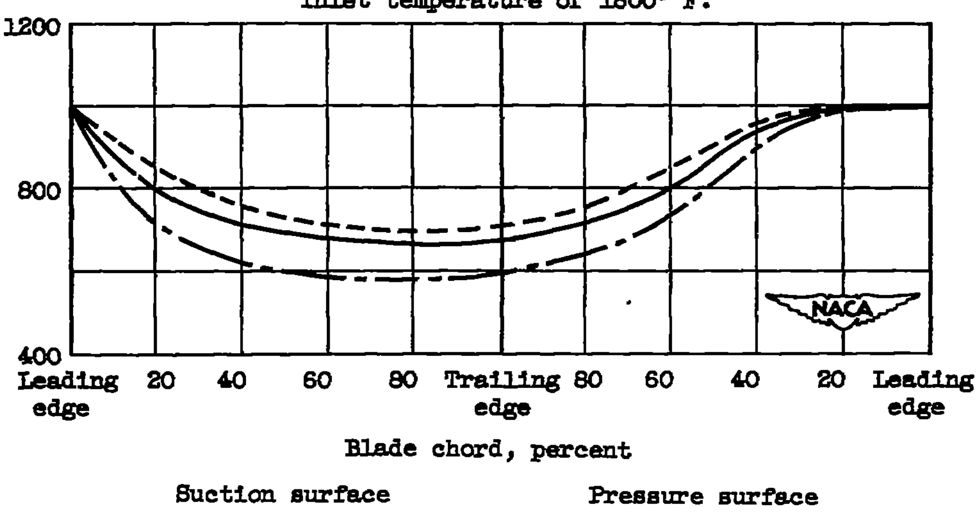
2597



(a) Blade designed for sea-level altitude and turbine-inlet temperature of 1600° F.

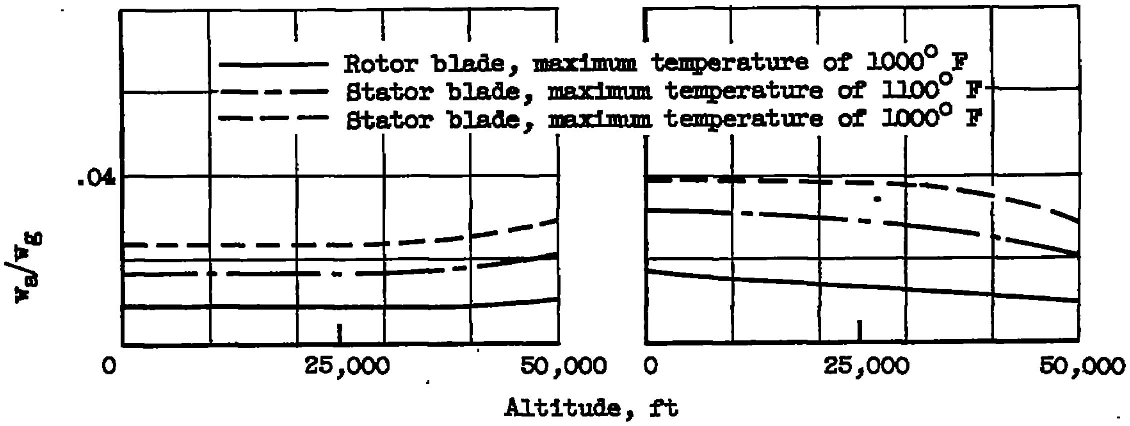


(b) Blade designed for altitude of 50,000 feet and turbine-inlet temperature of 1600° F.



(c) Blade designed for altitude of 50,000 feet and turbine-inlet temperature of 2500° F.

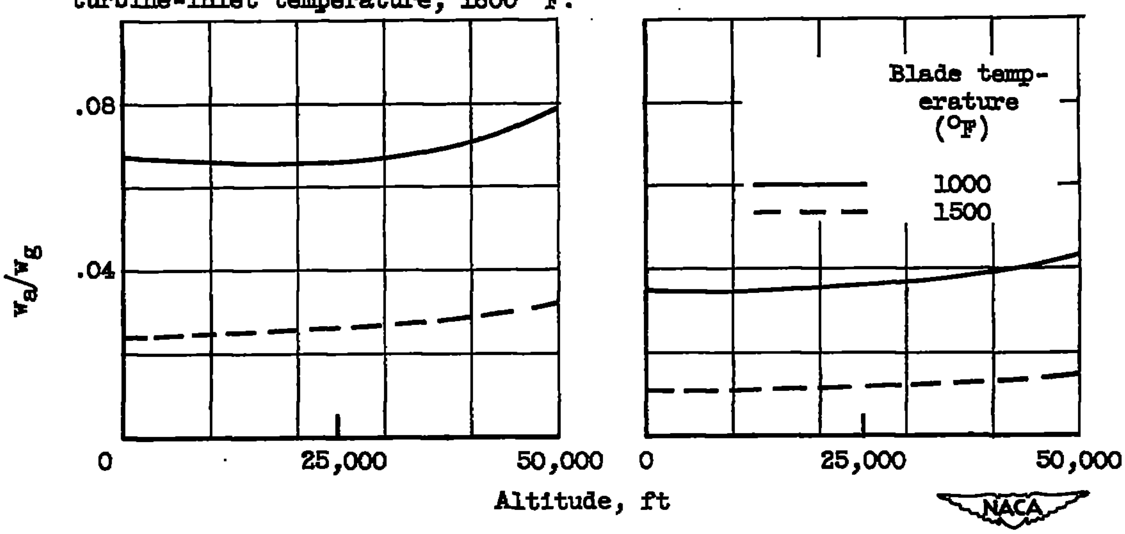
Figure 8. - Effect of blade design on off-design temperature distributions for stator blades without metering orifices. Flight Mach number, 1.0; engine compressor pressure ratio, 4; turbine-inlet temperature, 1600° F.



(a) Ideal coolant flow requirements for uniform blade temperature.

(b) Coolant flow requirements for blades designed for altitude of 50,000 feet.

Figure 9. - Comparison of rotor and stator blade coolant flow requirements for two stator blade temperatures. Maximum rotor blade temperature, 1000° F; flight Mach number, 1.0; engine compressor pressure ratio, 4; turbine-inlet temperature, 1600° F.

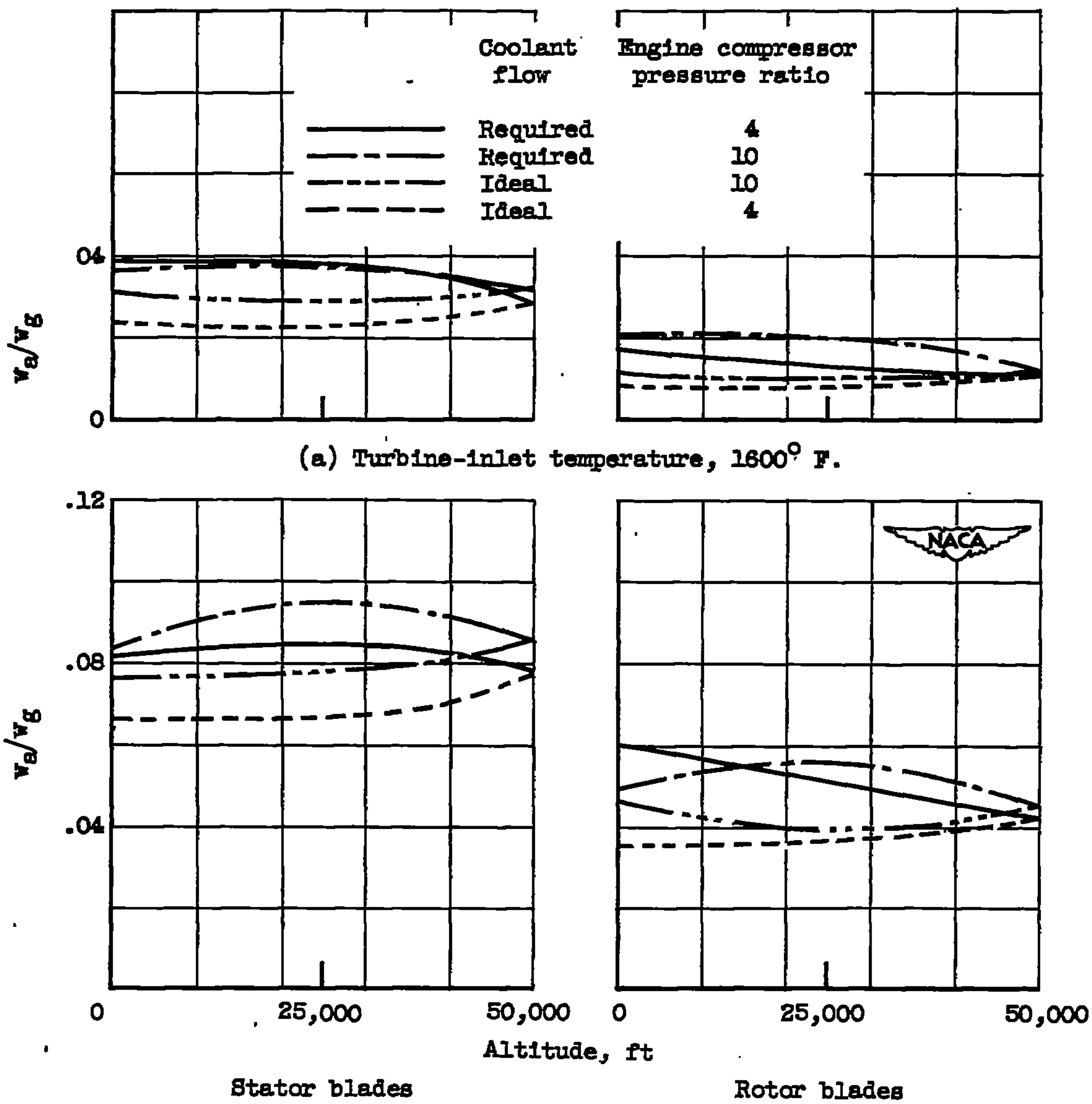


Stator blades

Rotor blades

Figure 10. - Effect of blade metal temperature on ideal coolant flow requirements. Turbine-inlet temperature, 2500° F; flight Mach number, 1.0; engine compressor pressure ratio, 4.

2597



(a) Turbine-inlet temperature, 1600° F.

(b) Turbine-inlet temperature, 2500° F.

Figure 11. - Effect of engine compressor pressure ratio on coolant flow requirements. Maximum blade temperature, 1000° F; flight Mach number, 1.0; blades designed for altitude of 50,000 feet.

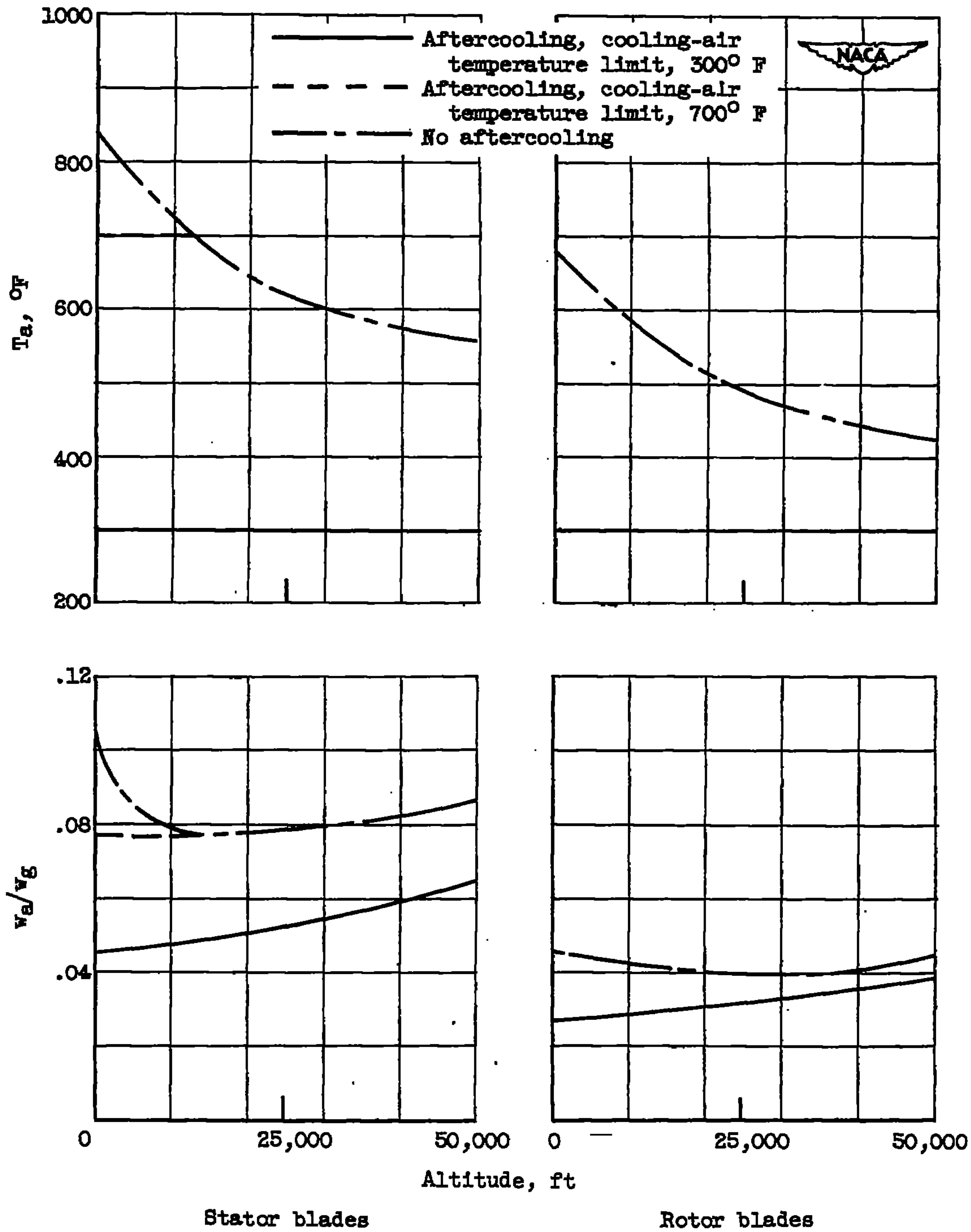
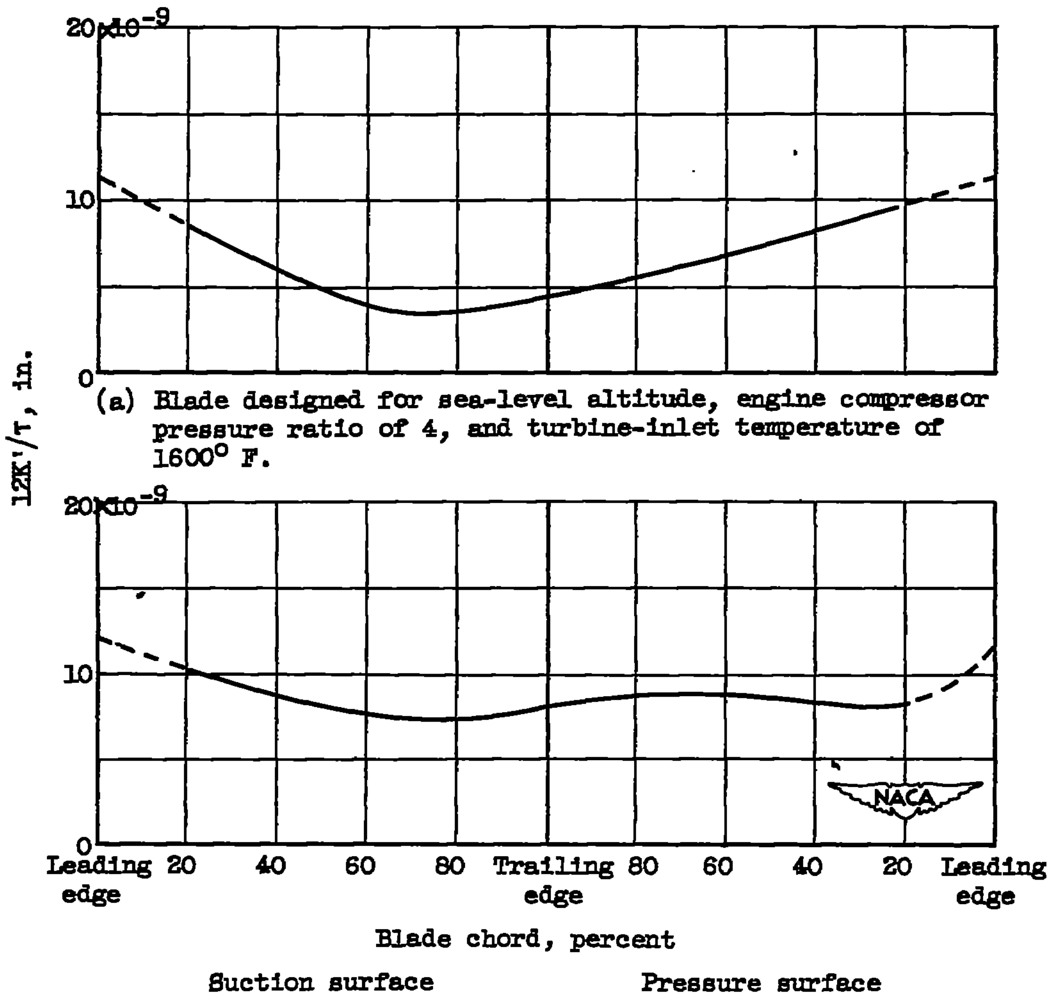


Figure 12. - Effect of aftercoolers on ideal cooling-air requirements. Uniform blade temperature, 1000° F; flight Mach number, 1.0; engine compressor pressure ratio, 10; turbine-inlet temperature, 2500° F.

2597

2597

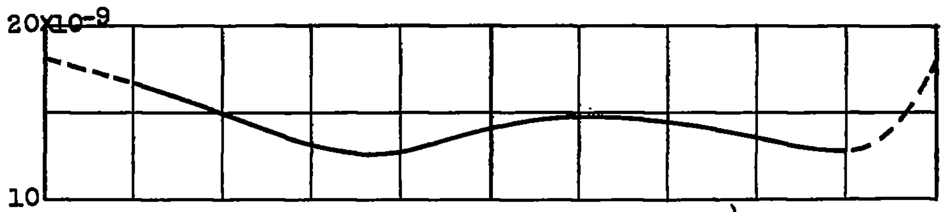


(a) Blade designed for sea-level altitude, engine compressor pressure ratio of 4, and turbine-inlet temperature of 1600° F.

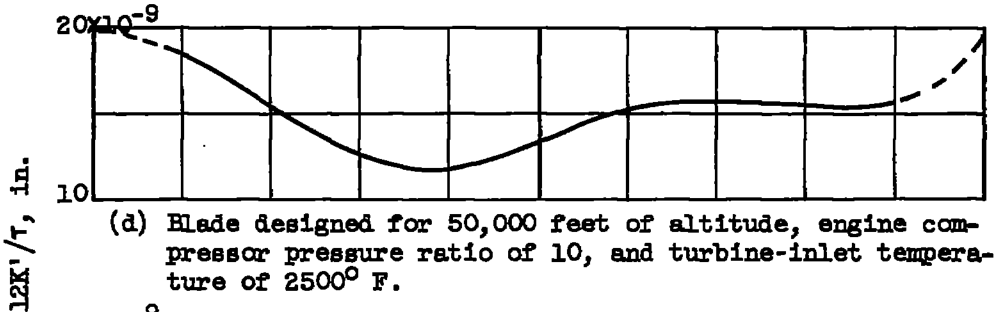
(b) Blade designed for 50,000 feet of altitude, engine compressor pressure ratio of 4, and turbine-inlet temperature of 1600° F.

Figure 13. - Permeability variations around stator blades without metering orifices for various design conditions. Maximum blade temperature, 1000° F; flight Mach number, 1.0. Permeabilities calculated for pressure drop through wall of 10 pounds per square inch discharging to NACA standard sea-level conditions.

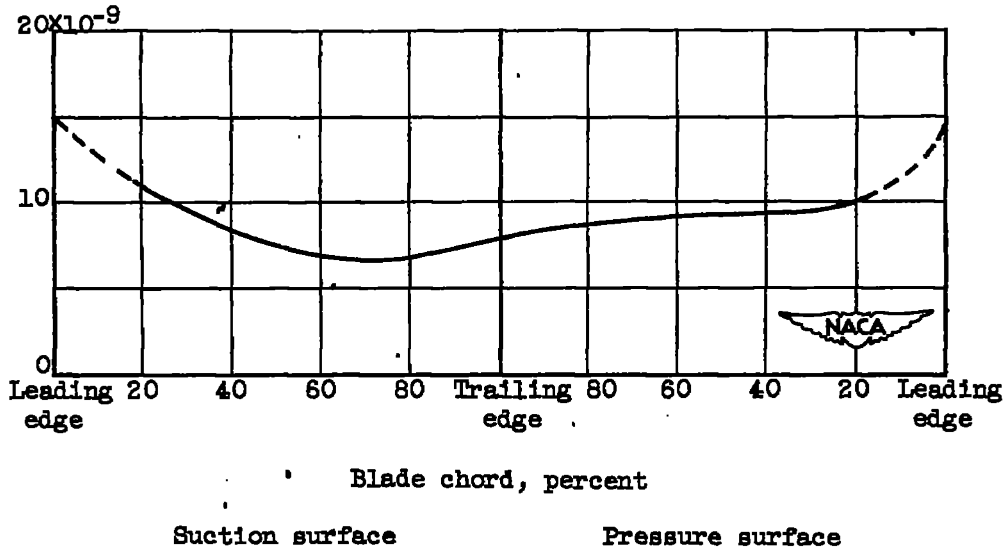
2597



(c) Blade designed for 50,000 feet of altitude, engine compressor pressure ratio of 4, and turbine-inlet temperature of 2500° F.



(d) Blade designed for 50,000 feet of altitude, engine compressor pressure ratio of 10, and turbine-inlet temperature of 2500° F.



(e) Blade designed for 50,000 feet of altitude, engine compressor pressure ratio of 10, and turbine-inlet temperature of 1600° F.

Figure 13. - Concluded. Permeability variations around stator blades without metering orifices for various design conditions. Maximum blade temperature, 1000° F; flight Mach number, 1.0. Permeabilities calculated for pressure drop through wall of 10 pounds per square inch discharging to NACA standard sea-level conditions.

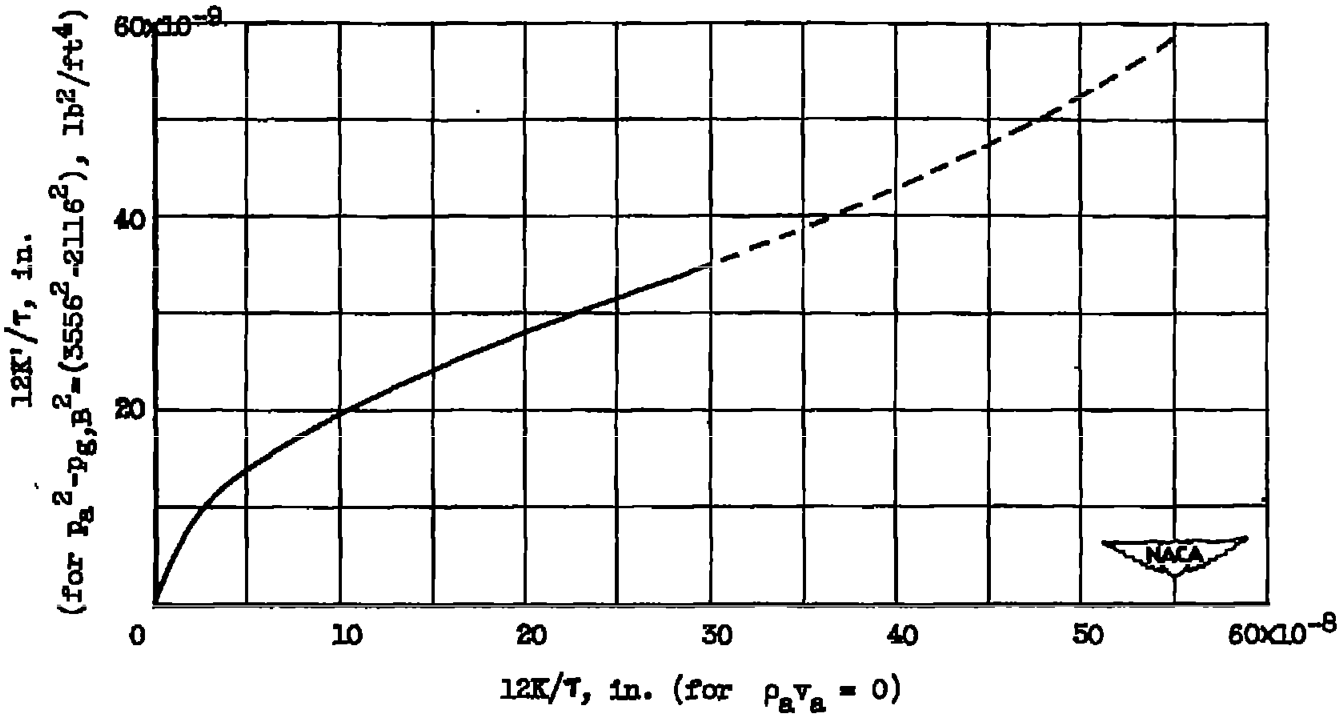
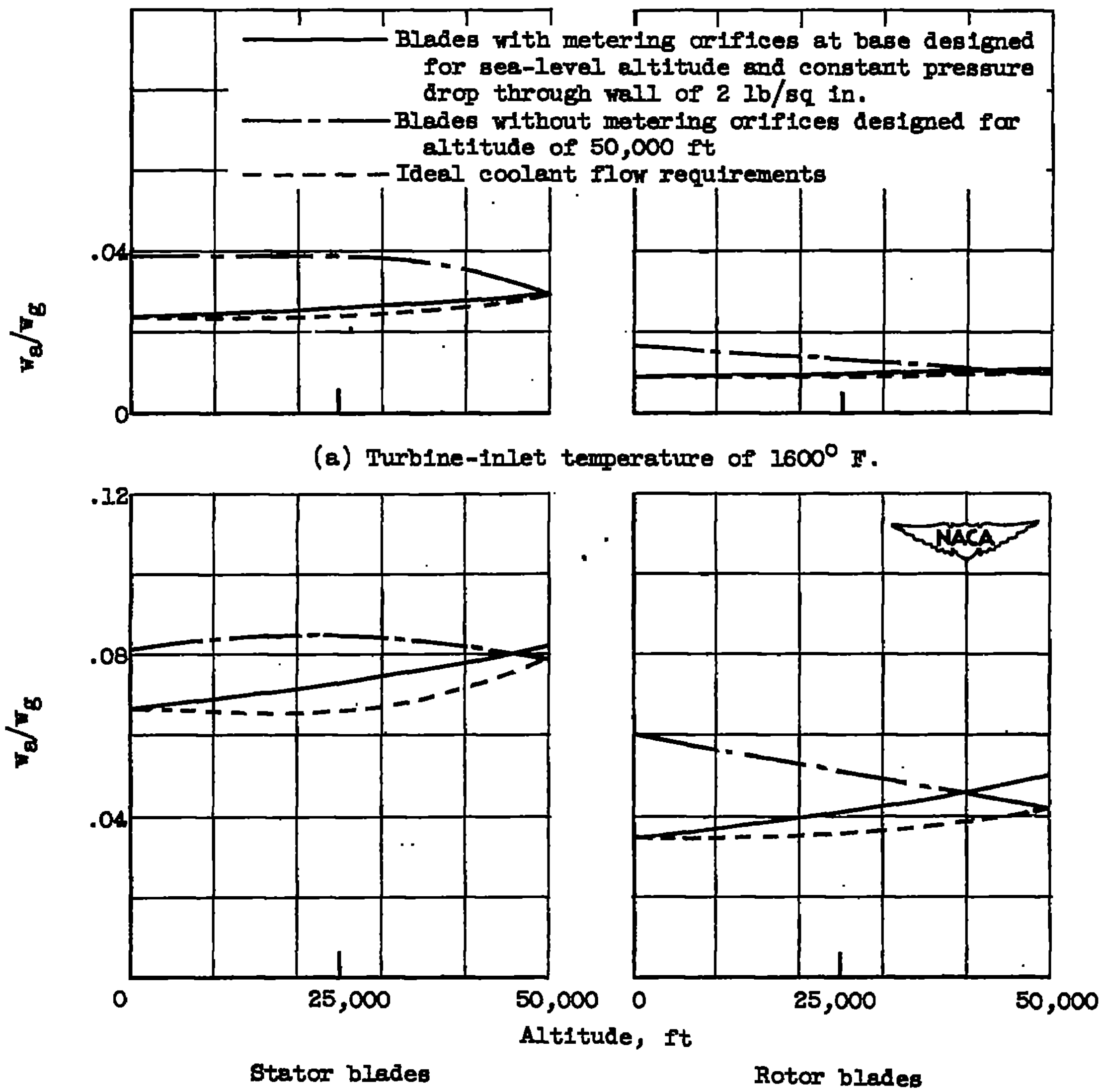


Figure 14. - Curve for conversion of permeabilities calculated for zero flow and permeabilities calculated for pressure drop through wall of 10 pounds per square inch discharging to NACA standard sea-level atmosphere. Data obtained for brazed and calendered stainless-steel wire meshes in references 9 and 10.



(b) Turbine-inlet temperature of 2500° F.

Figure 15. - Effect of orifices for metering coolant flow to local positions around blade periphery on coolant flow requirements for two turbine-inlet temperatures. Maximum blade temperature, 1000° F; flight Mach number, 1.0; engine compressor pressure ratio, 4.

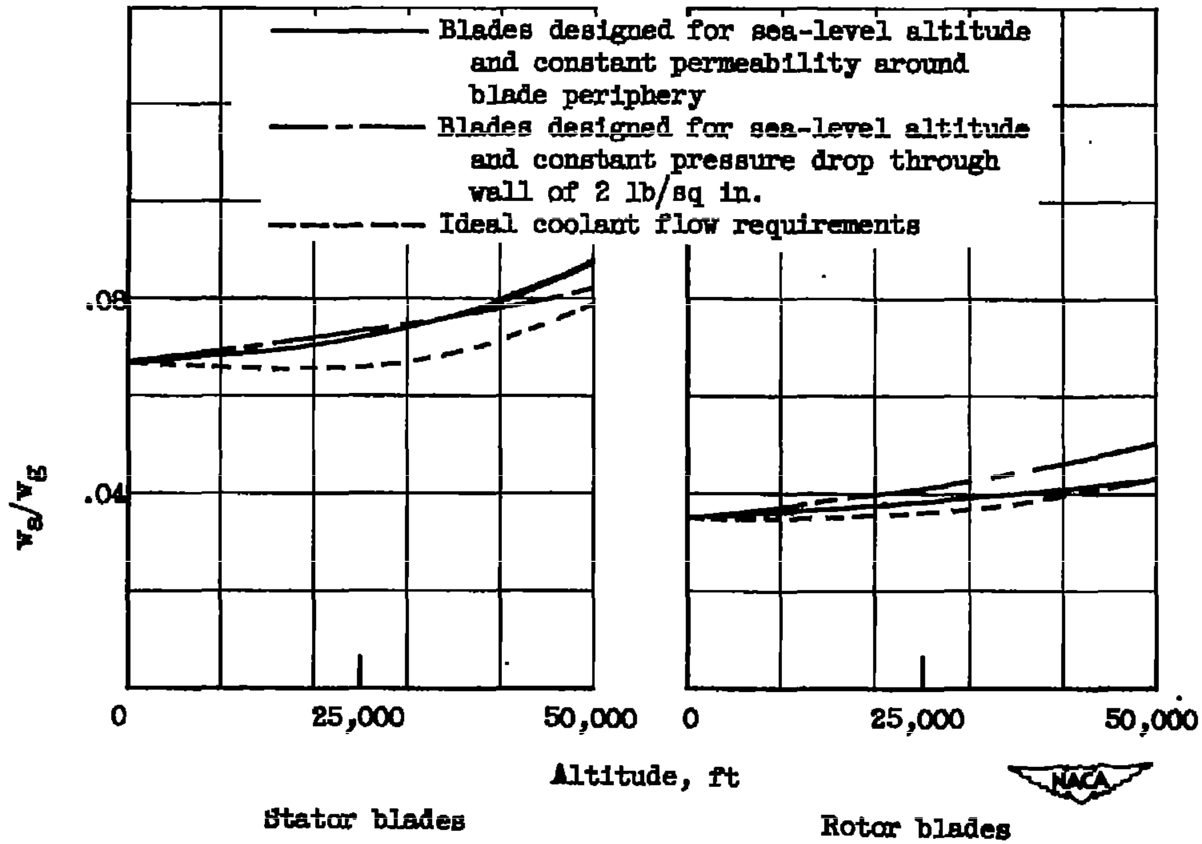


Figure 16. - Comparison of coolant flow requirements for two types of transpiration cooled blade with orifices at blade base. Turbine-inlet temperature, 2500° F; maximum blade temperature, 1000° F; flight Mach number, 1.0; engine compressor pressure ratio, 4.

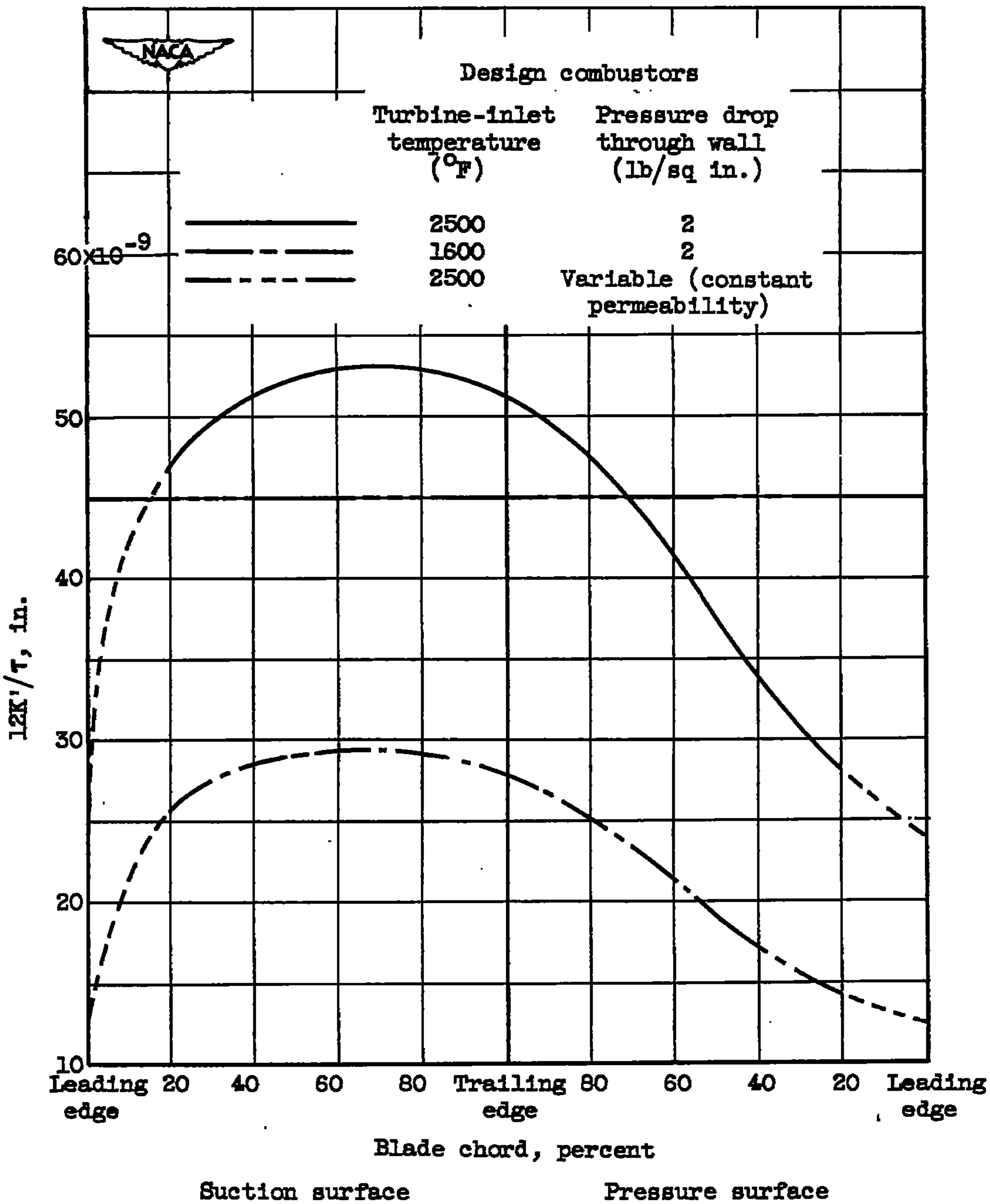


Figure 17. - Permeability variations around three stator blade designs utilizing metering orifices at blade base. Blades designed for sea-level altitude, blade temperature of 1000° F, engine compressor pressure ratio of 4, and flight Mach number of 1.0. Permeabilities calculated for pressure drop through wall of 10 pounds per square inch discharging to NACA standard sea-level conditions.

NACA-Langley - 9-8-52 - 350