TURBINE COOLING - ITS LIMITATIONS AND ITS FUTURE

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SUMMARY

The relative merits of convection, transpiration, and full coverage film cooling methods were analytically investigated for local turbine inlet temperatures from 2000° to 3500° F (1367 to 2200 K), gas pressures from 5 to 40 atmospheres (50.7 to 405.3 N/cm^2) and cooling air temperatures from 600° to 1200° F (589 to 922 K). Effects of wall thickness and material temperature were also investigated. Transpiration or full coverage film cooling will probably be necessary to permit operation at local turbine inlet temperatures on the order of 3000° F (1922 K) and compressor pressure ratios of 20 or higher. Full coverage film cooling is often superior to transpiration cooling because oxidation problems with transpiration cooled materials reduce allowable metal operating temperatures. Increasing allowable metal temperature 100° F (56 K) or reducing cooling air temperature 200° F (111 K) can do more to improve convection cooling than is possible by improvements in current advanced state-of-the-art convection cooled blade or vane design.

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INTRODUCTION

It is necessary to periodically review the expected future trends for gas turbine engines and then determine how these trends will affect the necessary research and development work in the various engine components. The trend in both military and commercial gas turbine engines is towards turbofan engines having a compact, high temperature gas generator. To make such engines compact, lightweight, and with superior specific fuel consumption, there is the need to simultaneously increase both compressor pressure ratio and turbine inlet temperature. Compressor pressure ratios to 40 or higher appear to be looming in the future and turbine inlet temperatures may go to those corresponding to stoichiometric fuel-air mixtures if materials and cooling designs can be developed that will tolerate such temperatures reliably.

Up to the present time convection cooling has been the primary means of cooling gas turbine engines, with some augmentation of this cooling using film cooling slots in critical regions. At the severe cooling conditions expected in future engines, it is likely that convection cooling will be inadequate, and more advanced cooling schemes such as film and transpiration cooling will have to be utilized. In Ref. 1 it was shown that the potentials of convection cooling could be determined in a relatively simple manner by considering the blades as heat exchangers and evaluating the cooling requirements on a basis of the heat capacity of the cooling air flowing through the blades and vanes. Research being conducted on transpiration and film cooling methods is reported in Refs. 2 to 10.

In order to evaluate the relative potentials of convection, film, and transpiration cooling for future engines, analyses were made of expected heat fluxes that will be encountered in future engines for ranges of gas pressure from 5 to 40 atmospheres (50.7 to 405.3 N/cm^2) and for local turbine inlet temperatures from 2000° to 3500° F (1367 to 2200 K). Required coolant flows were then calculated for turbine rotor blades and stator vanes and compared to those needed for advanced convection cooled vanes and blades at a local turbine inlet temperature of 2500° F (1644 K). Parameters investigated include cooling air temperature from 600° to 1200° F (589 to 922 K), blade and vane maximum metal outside surface temperatures from 1400° to 2400° F (1033 to 1588 K), and blade and vane wall thicknesses from 0.020 to 0.050 inch (0.51 to 1.27 mm).

ANALYTICAL PROCEDURE

In order to make the results of this investigation be of a general nature rather than for a specific

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application, very few assumptions were made that would restrict the results to applications for any specific engine, but some specific assumptions were required. These assumptions are listed below:

Condition	Stator	Rotor
Gas approach Mach number	0.24	0.5
Average gas channel Mach number	0.60	0.65
Airfoil chord	2.0 in. (5.08 cm)	1.5 in. (3.81 cm)
Ratio of relative gas total pressure to stator inlet total pressure	1.0	0.7
Ratio of relative gas total temperature to stator inlet total temperature	1.0	0.9

Heat Transfer Models

Figure 1 illustrates the three heat transfer models that were used in analyzing the coolant requirements for convection, full coverage film, and transpiration cooled turbine blades and vanes. Figure 1(a) illustrates in schematic form one of the many heat transfer models that can be used for convection cooling. With this cooling method, all of the heat transferred from the gas to the blade or vane must be conducted through the wall before it can be transferred to the cooling air by convection. As a result the temperature drop through the wall, due to conduction, decreases the driving temperature difference between the inner wall surface and the coolant. The thermal effectiveness of the cooling air is then defined as

$$\eta_{conv} = \frac{T_{c,o} - T_{c,i}}{T_{B,i} - T_{c,i}} \tag{1}$$

(All symbols are defined in the appendix.) For the analyses of this report, calculations were made for values of thermal effectiveness, η_{conv} equal to 0.5, 0.7, and 1.0.

Figure 1(b) illustrates the heat transfer model used for the transpiration cooling analysis. With transpiration cooling, the coolant is still in contact with the wall on the outer (hottest) surface. As a result, this cooling scheme has the potential of heating the cooling air to a higher temperature than is possible with convection cooling, therefore, for transpiration cooling the equation for thermal effectiveness of the cooling air

$$\eta_{trans} = \frac{T_{c,o} - T_{c,i}}{T_{B,o} - T_{c,i}} \tag{2}$$

has the value $T_{B,o}$ in the denominator instead of $T_{B,i}$ as was in Eq. (1) for convection cooling. The value of $\eta_{trans} = 0.8$ is an assumed value that will be discussed in more detail later.

In addition for transpiration cooling the gas-to-blade heat transfer coefficient is reduced relative to convection cooling by the layer of boundary layer air that is developed by transpiration as discussed in Refs. 6 to 10. A transpiration cooled blade or vane will therefore have less heat transferred to it from the gas than a convection cooled blade or vane.

Figure 1(c) illustrates the full coverage film cooling heat transfer model. This type of film cooling resembles transpiration cooling except there are fewer (and larger) holes for the cooling air to pass through to the outside surface. Full coverage film cooling provides a layer of air on the outside surface of the blade or vane that reduces the gas-to-blade heat transfer coefficient relative to convection cooling, but to a lesser degree than for transpiration cooling (due to a more uniform cool air layer with transpiration cooling). In the analysis conducted in this paper, the full coverage film cooling model was considered to approach maximum usefulness of film cooling by providing a very large number of small film cooling passages. The equation for thermal effectiveness η_{film} for film cooling is the same as Eq. (2) for transpiration cooling, but the value of thermal effectiveness ($\eta_{film} = 0.6$) was assumed to be lower, as will be discussed in more detail later. With full coverage film cooling, depending upon the configuration, the cooling air heating could all occur within the film cooling passages, or part of the heating could result from convection cooling on the inside surface prior to the cooling air entering the film cooling passages.

Convection Cooling Analysis

For convection cooling under steady state conditions the following one-dimensional heat transfer equations are applicable:

Hot gas to outside surface

$$\frac{Q}{A} = h_{conv}(T'_g - T_{B,o}) \tag{3}$$

Conduction through the wall

$$\frac{Q}{A} = \frac{k_B}{t}(T_{B,o} - T_{B,i}) \tag{4}$$

Heat picked up by cooling air

$$\frac{Q}{A} = \frac{c_{p,c}w_c}{A}(T_{c,o} - T_{c,i}) \tag{5}$$

Combining Eqs. (1), (3), (4), and (5) results in

$$\frac{w_c c_{p,c}}{A} = \frac{h_{conv}}{\eta_{conv}}\left[\frac{T'_g - T_{B,o}}{T_{B,o} - \frac{h_{conv}t}{k_B}(T'_g - T_{B,o}) - T_{c,i}}\right] \tag{6}$$

Eqs. (3) and (6) contain the term for relative total gas temperature T'_g. To be more exact, this term should be the adiabatic wall temperature, sometimes called the effective gas temperature. For subsonic relative velocities the effective gas temperature is within 1 or 2 percent of the relative total gas temperature. For the purposes of this investigation, use of the relative total gas temperature was accurate enough.

It was assumed that aft of the leading edge the boundary layer was turbulent and the average heat transfer coefficient could be calculated by the flat plate equation of Ref. 11.

$$\bar{h}_{conv} = 0.037\,Re^{0.8}Pr^{1/3}\frac{k}{L} \tag{7}$$

where the characteristic dimension L is the surface length of the blade or vane from the leading edge. Values of L used in the calculations were 2 inches (5.08 cm) for stator vanes and 1.5 inches (3.81 cm) for rotor blades. The fluid properties in Eq. (7), including the temperature effect on density were evaluated at the reference temperature given in Ref. 11.

The fluid properties, Pr, thermal conductivity, ratio of specific heats and viscosity were obtained from the charts in Ref. 12 for ASTM-A-1 fuel and a pressure of 10 atmospheres.

The gas-to-surface heat transfer coefficient at the leading edge was obtained for a cylinder in cross flow from Ref. 13:

$$h_{conv} = 1.14\frac{k}{D}Re^{1/2}Pr^{0.4}\left[1 - \left(\frac{\varphi}{90}\right)^3\right] \tag{8}$$

where $\varphi = 0$ at the stagnation point.

Data from Ref. 1 are shown in Fig. 2 to illustrate the range of cooling air thermal effectiveness

$$\eta_{conv,avg} = \frac{T_{c,o} - T_{c,i}}{T_{B,avg} - T_{c,i}}$$

that can be expected for a range of relatively sophisticated convection cooled turbine blades designed for a supersonic aircraft engine at cruise conditions (gas pressure of 3.5 atm or 35.4 N/cm^2). It is seen that these blades have thermal effectiveness values ranging from slightly more than 0.5 to about 0.8. If

the inside wall temperature $T_{B,i}$ had been used in defining this effectiveness (as in Eq. (1)) instead of the average wall temperature $T_{B,avg}$, the values of η_{conv} would have been slightly higher (less than 5 percent). The effect is small in this case since the heat fluxes in Ref. 1 were low enough that the difference between $T_{B,i}$ and $T_{B,avg}$ was low (on the order of 35° F (19 K)).

The value of η_{conv} is not a constant for a given convection cooling configuration. It decreases as the cooling airflow (Reynolds number) increases. Therefore, η_{conv} will be smaller for the more difficult cooling conditions of high gas pressure and temperature. The dependence of η_{conv} on Reynolds number is shown from the following simple analysis. The heat transferred to the cooling air is

$$\frac{Q}{A_i} = h_c(T_{B,i} - T_{c,i})$$

Combining this equation with Eqs. (1) and (5) results in

$$\eta_{conv} = \frac{h_c A_i}{w_c c_{p,c}}$$

which can be written

$$\eta_{conv} = \frac{Nu_c}{Re_c Pr_c} \frac{A_i}{A_f}$$

For turbulent heat transfer and a constant cooling air Prandtl number

$$Nu_c \propto Re_c^{0.8}$$

therefore,

$$\eta_{conv} \propto \frac{1}{Re_c^{0.2}}$$

In the analysis of this paper the cooling air Reynolds number could range up to 16 times the value for the conditions of Fig. 2. Such Reynolds numbers could result in a reduction of η_{conv} by a factor of as much as 1.74. The maximum value of η_{conv} on Fig. 2 was about 0.8. Dividing this number by 1.74 could result in η_{conv} values as low as 0.46.

Most relative coolant flow ratios presented in this paper are for a value of $\eta_{conv} = 0.7$. The value of relative coolant flow ratio for convection cooling corresponding to any other value of η_{conv} can be obtained very simply by multiplying the relative coolant flow ratio read from the curves of this paper by the ratio $0.7/\eta_{conv}$.

For the analyses of this paper, values of 0.5, 0.7, and 1.0 were taken for cooling air thermal effectiveness to show the range of interest with convection cooling, while 0.7 was taken as a value that represents advanced, but possible, cooling designs. A value of $\eta_{conv} = 1.0$ is the ideal case and represents a limit to the amount of cooling that is possible with convection.

Transpiration Cooling Analysis

With transpiration cooling the heat transfer analysis approach was similar to that for convection cooling except the inside wall temperature does not provide a restriction on the temperature rise of the cooling air. The following one-dimensional heat transfer equations are applicable:

Hot gas to outside surface

$$\frac{Q}{A} = h_{conv} \frac{h_{trans}}{h_{conv}} (T'_g - T_{B,o}) \tag{9}$$

Heat picked up by cooling air

$$\frac{Q}{A} = \frac{c_{p,c} w_c}{A} (T_{c,o} - T_{c,i}) \tag{5}$$

Combining Eqs. (2), (5), and (15) results in

$$\frac{w_c c_{p,c}}{A} = \frac{h_{conv}}{\eta_{trans}} \left(\frac{h_{trans}}{h_{conv}}\right) \left(\frac{T'_g - T_{B,o}}{T_{B,o} - T_{c,i}}\right) \tag{10}$$

The reduction in gas-to-wall heat transfer coefficient with transpiration cooling has been determined experimentally in Refs. 6 to 10 and the results agree with the following relationship developed in Refs. 14 and 15.

$$\frac{St_{trans}}{St_{conv}} = \frac{B}{e^B - 1} \tag{11}$$

where

$$B = \frac{\dfrac{\rho_c V_c}{\rho_g V_g}}{St_{conv}} \tag{12}$$

$$B = \frac{\rho_c V_c c_{p,g}}{h_{conv}} \tag{13}$$

The Stanton number ratio (Eq. (11)) is equal to the ratio of heat transfer coefficients, and noting that

$$\rho_c V_c = \frac{w_c}{A}$$

Eqs. (11) and (13) can be rewritten

$$\frac{h_{trans}}{h_{conv}} = \frac{B}{e^B - 1} \tag{11a}$$

$$B = \left(\frac{w_c c_{p,c}}{A}\right) \left(\frac{c_{p,g}}{c_{p,c}}\right) \left(\frac{1}{h_{conv}}\right) \tag{13a}$$

The required coolant flow for a transpiration cooled surface can be obtained by combining Eqs. (10), (11a), and (13a), to yield

$$\frac{w_c c_{p,c}}{A} = h_{conv} \frac{c_{p,c}}{c_{p,g}} \ln\left[\frac{1}{\eta_{trans}} \frac{c_{p,g}}{c_{p,c}} \left(\frac{T'_g - T_{B,o}}{T_{B,o} - T_{c,i}}\right) + 1\right] \tag{14}$$

The value of η_{trans} in Eq. (14) was assumed to be 0.8. It is more frequently assumed that the cooling air discharges from a transpiration cooled surface at the same temperature as the outside wall temperature, which would result in η_{trans} = 1.0. Based on the fact that transpiration cooled surfaces in gas turbine engines will be thin (about 0.030 to 0.050 in. or 0.76 to 1.27 mm) with a limited amount of internal (within the wall) surface area in contact with the coolant, it seems unlikely that η_{trans} = 1.0 is a good assumption. The value of 0.8 appears to offer some conservatism to the analysis.

Full Coverage Film Cooling Analysis

To obtain maximum cooling effectiveness from film cooling, the holes should be closely spaced over the entire surface in order to generate complete film coverage over the surface. Such a cooling scheme approaches transpiration cooling, but it is less effective than transpiration cooling because (1) a finite number of film cooling holes will not provide as uniform an insulating film of cooling air on the outside surface as the almost infinite number of holes obtained by transpiration cooling, and (2) transpiration cooling can utilize more of the heat capacity in the cooling air for reducing wall temperatures by internal

convection within the walls due to the very large surface area in contact with the coolant relative to film cooling.

Most film cooling analyses, such as those in Refs. 2 and 5, are for isolated holes or for rows of holes. These investigations and others have utilized several different methods of accounting for the method of heat transfer by film cooling. Frequently, it is assumed that the gas-to-surface heat transfer coefficient is the same as with convection, but the effective gas temperature is reduced by the insulating effect of the film. Some investigators have considered a reduction in gas-to-surface heat transfer coefficient due to film cooling. Since the model assumed in the present investigation has a large number of holes, which approaches transpiration cooling, a modified transpiration cooling analysis was used herein to calculate the cooling air requirements for full coverage film cooling.

It was assumed that the gas-to-surface heat transfer coefficient for full coverage film cooling would be a value midway between that for transpiration cooling and convection cooling and that the film cooling thermal effectiveness η_{film} would be equal to 0.6. From limited information available both of these assumptions are conservative based on the potential of full coverage film cooling techniques. The calculation procedure for full coverage film cooling was similar to that for transpiration cooling. Equation (10) was replaced by

$$\frac{w_c c_{p,c}}{A} = \frac{h_{conv}}{\eta_{film}} \left(\frac{h_{film}}{h_{conv}}\right) \left(\frac{T'_g - T_{B,o}}{T_{B,o} - T_{c,i}}\right) \tag{15}$$

and Eq. (11a) was replaced by

$$\frac{h_{film}}{h_{conv}} = \frac{1}{2} \left(1 + \frac{h_{trans}}{h_{conv}}\right) \tag{16}$$

where h_{trans}/h_{conv} was calculated from Eq. (11a) for the value of coolant flow from Eq. (15). This procedure required iteration of Eqs. (11a), (13a), (15), and (16).

Relative Coolant Flow Requirements

An approximate analysis, such as the one in this paper, does more to indicate trends than absolute values of coolant flow that are required for various cooling methods. The analysis compares convection, full coverage film, and transpiration cooling based on cooling air thermal effectiveness η. Such an approach can be useful, but its limitations must also be considered. As previously discussed, it is probably not possible to obtain the same values of cooling air thermal effectiveness for all cooling conditions due to variations in cooling air Reynolds number resulting from variations in temperature and pressure level of the gas and coolant, and the temperatures of the blade or vane material.

This paper compares the various cooling conditions and cooling methods on the basis of "relative coolant flow ratio." Coolant flow ratio is defined as the ratio of cooling airflow to compressor airflow. Relative coolant flow ratio is the coolant flow ratio relative to a "base" flow for a specific convection cooled blade or vane. In this paper the coolant flows are per unit of blade or vane surface area. The base flows have a compensation for gas pressure level. This compensation accounts for variations in compressor airflow with pressure level. No attempt was made to apply a similar compensation for variations in compressor airflow resulting from gas temperature level since such a compensation would be small relative to the compensation for pressure.

For comparison of effects of gas, coolant, and material temperature, and material thickness at a constant gas pressure of 20 atmospheres (202.6 N/cm^2) the "base" coolant flow used to calculate the "relative coolant flow ratio" is that required for the same location on a blade or vane of an advanced state-of-the-art convection cooled blade or vane (η_{conv} = 0.7) at the following conditions: (1) total gas pressure at the stator inlet of 20 atmospheres (202.6 N/cm^2), (2) turbine inlet temperature of 2500° F (1644 K), (3) cooling air temperature of 1000° F (811 K), (4) blade or vane wall thickness of 0.050 inch (1.27 mm), and (5) material temperatures consistent with present practice (1700° F or 1200 K for aft sections of rotor blades, 1800° F or 1255 K for aft sections of stator vanes and rotor blade leading edges, and 2000° F or 1367 K for stator vane leading edges).

For gas pressures different from 20 atmospheres (202.6 N/cm^2) the "base" coolant flow in the relative coolant flow ratio at each pressure level has been assumed to be that which would result if the coolant side heat transfer coefficient varied with gas pressure level in a manner exactly proportional to that for the gas side coefficient. All other conditions are the same as those listed above for 20 atmospheres (202.6 N/cm^2). This method of comparison is believed to give a reasonable approximation of how the coolant flow ratio would have to be varied in an engine as design gas pressure is varied.

CONVECTION COOLING

In this paper, turbine inlet temperature is defined as the total gas temperature at the stator inlet. The relative total gas temperature for the rotor was assumed to be 90 percent of the turbine inlet temperature. This temperature reduction results from both hot gas dilution from stator cooling air and the relative temperature reduction due to rotation of the rotor. The turbine inlet temperatures for stator vanes should be considered as local temperatures since the vanes must be designed for combustor outlet hot spots. Local variations in turbine inlet temperature are smaller for rotor blades than for stator vanes due to the averaging effect of blade rotation, but there are radial variations. Rotor blade turbine inlet temperatures are therefore considered to be circumferentially averaged temperatures measured at the stator inlet for the radial position of interest.

Cooling Air Thermal Effectiveness

The relative coolant flow requirements per unit of surface areas are shown in Fig. 3 for the section aft of the leading edge of turbine stator vanes. Figure 3(a) illustrates the effect of the cooling air thermal effectiveness η_{conv} on cooling air requirements. As previously discussed, advanced convection cooled configurations can have a cooling air thermal effectiveness η_{conv} in the range between 0.5 and 0.7. The ideal, but unobtainable, value of thermal effectiveness equal to 1.0 would occur if the cooling air temperature could be heated to the inner wall surface temperature of the blade or vane. Figure 3(a) shows that for the entire range of cooling air thermal effectiveness shown coolant flows may become excessive for local turbine inlet temperatures much above 2500° F (1644 K) with convection cooling for the typical values of cooling air temperature, blade material temperature, wall thickness, and gas pressure listed in the figure.

Cooling Air and Material Temperature Effects

Figures 3(b) and (c) show how the allowable turbine inlet temperature can be increased by reducing cooling air temperature or using materials having a higher temperature capability. These results are shown for a cooling air thermal effectiveness η_{conv} equal to 0.7. A material in stator vanes that could operate at 2200° F (1478 K) would permit the use of convection cooling to local gas temperatures up to 3500° F (2200 K).

For comparative purposes, the curves from Fig. 3(a) are superimposed to show η_{conv} values of 0.5 and 1.0 for a cooling air temperature of 1000° F (811 K) and an external surface metal temperature of 1800° F (1255 K). These figures show that decreasing cooling air temperature or finding materials that can operate at higher metal temperatures are far more effective in permitting either higher turbine inlet temperatures, or lower coolant flow ratios, than will ever be possible by improving the cooling effectiveness of convection cooled turbine blades and vanes. Figure 3(b) shows that reducing cooling air temperature about 150° to 200° F (83 to 111 K) (depending upon turbine inlet temperature) is approximately equivalent to increasing η_{conv} from 0.7 to the ideal value of 1.0 for turbine inlet temperatures in excess of 2500° F (1644 K). Figure 3(c) shows that a 100° F (56 K) increase in allowable metal temperature is as effective as increasing η_{conv} from 0.07 to 1.0.

By taking partial derivatives of Eq. (6) for constant h_{conv} we find that

$$\frac{\left(\dfrac{\partial w_c}{\partial T_{B,o}}\right)}{\left(\dfrac{\partial w_c}{\partial T_{c,i}}\right)} = -\frac{T'_g - T_{c,i}}{T'_g - T_{B,o}}$$

Since $T_{c,i}$ is always less than $T_{B,o}$ it follows that the reduction in coolant flow from increasing blade or vane metal temperature is larger than that obtained for decreasing cooling air temperature by the same amount. The same result can be observed by inspection of Figs. 3(b) and (c).

Wall Thickness Effects

Figure 4(a) shows the beneficial effects of reducing wall thickness in gas turbines. The curves from Fig. 3(a) on the effect of η_{conv} for a wall thickness of 0.05 inch (1.27 mm) are also shown. For the high heat flux conditions, which occur at the higher gas temperature levels, reducing wall thickness can substantially reduce the coolant flow requirements. Thinner walls result in a smaller temperature drop through the walls as illustrated in Fig. 4(b). For a constant outside wall surface temperature, which may be determined by oxidation limitations, a smaller temperature drop through the wall results in a larger temperature difference between the wall temperature and the cooling air temperature. This larger temperature difference permits better utilization of the cooling air by allowing it to be heated to a higher temperature. Figure 4(a) shows that reducing the wall thickness from 0.05 to 0.03 inch (1.27 to 0.76 mm) can do more to increase the turbine inlet temperature capability for gas temperatures above about 2800° F (1810 K) than is possible by increasing η_{conv} from 0.7 to 1.0. It can be concluded, therefore, that for high temperature applications the wall should be made as thin as possible consistent with structural considerations such as foreign object damage, gas pressure forces, and oxidation effects. It will be shown in the next section that thin walls are even more important for gas pressures greater than the value of 20 atmospheres (202.6 N/cm^2) shown in Fig. 4.

Gas Pressure Effects

Figure 5 illustrates the effect of gas pressure (compressor pressure ratio) on coolant flow requirements and wall temperature drops for convection cooled stator vanes. For convection cooling, high gas pressures substantially increase coolant flow requirements and limit the maximum attainable turbine inlet temperature. At the high heat fluxes accompanying high gas pressures and the higher turbine inlet temperatures, the temperature drop through the wall becomes so great that the inside wall temperature approaches cooling air temperature and convection cooling is not possible for typical cooling air and metal temperatures.

FULL COVERAGE FILM AND TRANSPIRATION COOLING

The cooling effectiveness of full coverage film and transpiration cooling is superior to convection cooling and can be explained with the help of Fig. 6. This figure compares the gas-to-surface heat transfer coefficients, heat flux (Q/A), and cooling air temperature rise for the three methods of cooling. With convection cooling the gas-to-surface heat transfer coefficient gradually increases (Fig. 6(a)) as turbine inlet temperatures increase due primarily to increased gas thermal conductivity with increasing temperature. With transpiration cooling, however, the heat transfer coefficient decreases with increasing turbine inlet temperature. This decrease results from steadily increasing coolant flow rates which provide a thicker and thicker insulating layer of air over the cooled surface. The heat transfer coefficient for full coverage film cooling lies between that for convection and transpiration cooling, but it is not an average of the other two values. For a given turbine inlet temperature and material temperature, the coolant flow requirement is higher for full coverage film cooling than for transpiration cooling. This higher coolant flow causes the film cooling heat transfer coefficient to be closer to the transpiration coefficient than to the convection coefficient.

The heat fluxes shown in Fig. 6(b) are exactly as one would expect based on the heat transfer coefficients since the driving temperature difference between the gas and the surface is identical for all three

methods of cooling. The temperature rise plotted in Fig. 6(c) illustrates the second reason for the superiority of transpiration and film cooling over convection cooling. The solid curves illustrate the temperature difference between the hottest metal in contact with the coolant ($T_{B,c}$) and the coolant supply temperature ($T_{c,i}$). This temperature difference remains constant as turbine inlet temperature is increased for transpiration and full coverage film cooling since $T_{B,c}$ is the outside wall temperature. For convection cooling, however, the higher heat fluxes at high turbine inlet temperature result in an increasing temperature drop through the wall. Since the driving temperature difference ($T_{B,c} - T_{c,i}$) on the coolant side of the wall for convection cooling is the difference between the inside wall temperature ($T_{B,c}$ for convection) and the cooling air temperature ($T_{c,i}$), the heat that can be transferred to the cooling air decreases as the turbine inlet temperature increases.

The temperature rise of the cooling air, shown by the dashed lines in Fig. 6(c), is a direct measure of the heat absorbed per unit weight of cooling air flow rate. This temperature rise is influenced by both air thermal effectiveness η_{conv} and the maximum surface temperature in contact with the cooling air. This temperature rise is constant for transpiration and full coverage film cooling over the entire range of turbine inlet temperatures, but it decreases for convection cooling as turbine inlet temperature is increased. The result of these combined effects is shown in Fig. 6(d) as a plot of relative coolant flow ratio versus turbine inlet temperature.

Comparison With Convection Cooling

Relative coolant flow ratios are shown in Fig. 7 for both stator vanes and rotor blades for full coverage film cooling, transpiration cooling, and convection cooling. In Figs. 7(a) to (c), it will be observed that similar trends are shown for both stator vanes and rotor blades and for both the leading edge and aft section of vanes. The metal temperatures shown in Fig. 7 are generally consistent with present practice and vary according to location on the vane or whether there are centrifugal stresses such as in rotor blades. All of the coolant flow ratios are relative to those for the local area of a convection cooled vane or blade. It will be observed that the cooling conditions vary in Figs. 7(b) and (c) relative to Fig. 7(a). In Fig. 7(c) there is a better cooling capability for convection cooling at the vane leading edge because of a higher metal temperature which permits a greater temperature difference between vane wall and coolant. In Fig. 7(b) there is a lower relative gas temperature encountered on the rotor blades compared to the vane. As a result, the relative coolant flow ratios for transpiration and full coverage film cooling are higher in Figs. 7(b) and (c) compared to Fig. 7(a). This effect can be explained by the fact that transpiration and full coverage film cooling show their greatest superiority to convection cooling under the more difficult cooling conditions. As metal temperatures are raised or relative gas temperatures are reduced, the cooling problem becomes less severe.

The comparison shown in Fig. 7(c) is somewhat unrealistic in that at the present time a material temperature as high as 2000° F (1367 K) is not possible for transpiration cooling. At such high temperature levels oxidation becomes severe in porous materials and clogging occurs. At the present state-of-the-art, a more realistic permissible porous material temperature is 1600° F (1145 K). Additional discussion of this oxidation problem will be given in a later section of the paper. Figure 7(d) compares convection, full coverage film and transpiration cooling coolant flow requirements for the case where the material temperature for convection and full coverage film cooling is 2000° F (1367 K) and the material temperature for transpiration cooling is 1600° F (1145 K). With this material temperature limitation on transpiration cooling, it can be seen that the superiority of transpiration cooling relative to convection cooling disappears for gas temperatures less than 2600° F (1700 K). The superiority over full coverage film cooling disappears for gas temperatures less than about 3200° F (2030 K). Although not shown in Fig. 7, similar trends exist for the aft sections of turbine blades and vanes, but the gas temperature levels at which transpiration cooling becomes superior to convection and full coverage film cooling are lower in the aft section of the blades and vanes.

The results shown in Fig. 7 are for a gas pressure of 20 atmospheres (202.6 N/cm^2). It will be pointed out later that coolant flow requirements are substantially increased for convection cooling as gas pressures are increased, but not for film and transpiration cooling. Therefore transpiration cooling will show a greater superiority over convection cooling for gas pressures in excess of 20 atmospheres (202.6 N/cm^2).

Cooling Air Temperature Effects

The effects of cooling air temperature on convection coolant flow requirements were shown in Fig. 3(b). Figure 8 shows a replot of some of the data from Fig. 3(b) and similar results for full coverage film and transpiration cooling to illustrate the sensitivity of the three methods of cooling to cooling air temperature. The results for convection cooling are for a thermal effectiveness η_{conv} of 0.7. Figure 8 shows that cooling air temperature has a very marked effect on cooling airflow requirements for convection cooling. It shows that about the only hope for exceeding a turbine inlet temperature of 3000° F (1922 K) with convection cooling with an external wall surface temperature of 1800° F (1255 K) and a wall thickness of 0.050 inch (1.27 mm) without excessive coolant flows is to reduce the cooling air temperature by several hundred degrees below the compressor discharge temperature (which will usually be on the order of 1000° to 1200° F or 811 to 922 K). Such an approach would require rejecting heat from the cooling air to fuel or engine by-pass air. With transpiration or full coverage film cooling, however, cooling air temperatures up to 1200° F (922 K) could probably be tolerated. Although it would be beneficial to reduce cooling air temperature from the standpoint of cooling air requirements, the mechanical complexity may not be warranted.

Material Temperature Effects

The effects of material temperature on the cooling airflow requirements for stator vanes are shown in Fig. 9 for convection, full coverage film, and transpiration cooling. The material temperatures shown may not in all cases be feasible for cooled turbines, but the calculations were made to illustrate the benefits that might be obtained through material improvements. The figure shows that increasing stator vane material temperature to 2200° F (1478 K) makes convection cooling possible up to local turbine inlet temperatures of 3500° F (2200 K) for a gas pressure of 20 atmospheres (202.6 N/cm^2) and wall thicknesses of 0.050 inch (1.27 mm). Although 2200° F (1478 K) is approaching the melting point of nickel base alloys, which are the primary materials in use and under development for turbine blades and vanes, there are indications that some materials can operate for a few hours at this temperature in stator vanes. Transpiration and film cooling show a lower sensitivity to material temperature than convection cooling. Local turbine inlet temperatures up to 3500° F (2200 K) are feasible with considerably lower material temperatures for transpiration and advanced film cooling. In previous figures the material temperature for transpiration cooled stators was considered to be 1800° F (1255 K). With presently available transpiration cooled materials and very long time operation the material temperature may in some cases have to be reduced to as low as 1400° F (1033 K) in order to eliminate oxidation problems. If the material temperature had to be reduced from 1800° to 1400° F (1255 to 1033 K), Fig. 9 shows that the transpiration cooling coolant flow ratio would have to be almost doubled at a local turbine inlet temperature of 2500° F (1644 K), and increased about 50 percent at a local turbine inlet temperature of 3500° F (2200 K). A potential transpiration material with a high temperature capability will be discussed later.

Gas Pressure Effects

Figure 10 shows how gas pressure affects the cooling air requirements per unit surface area for convection, full coverage film, and transpiration cooling for stator vanes. It will be observed that gas pressure (compressor pressure ratio) has a significant effect on cooling air requirements with convection cooling but no effect with film cooling or transpiration cooling for the assumptions used in this analysis.

Figure 10 shows that as gas pressure is increased above 20 atmospheres (202.6 N/cm^2) convection cooling requires excessive cooling air at gas temperatures much above 2500° F (1644 K) for the aft section of stator vanes having a thickness of 0.050 inch (1.27 mm). The reason for this rapid rise in cooling airflow with increasing compressor pressure ratio is the high heat flux that results from increased gas-to-surface heat transfer coefficients at high pressures. These increased heat fluxes cause a large temperature drop through the wall from the conduction process. As a result, the inside wall temperature begins to approach the cooling air temperature for a constant maximum outside wall temperature. This low inside wall temperature can make convection cooling impractical.

With transpiration and full coverage film cooling, the maximum temperature difference between the cooling air and wall temperature remains constant regardless of heat flux (see Fig. 6(c)). As a result,

compressor pressure ratio does not influence the coolant flow ratio for film and transpiration cooling.

TURBINE BLADE LIFE

Turbine blade life is influenced by stress level, temperature level of the blade material, material strength at high temperatures, and thermal stresses that develop as a result of temperature variations in the wall. For high heat fluxes the temperature drop through the wall may become as large or larger than the variations around the blade periphery, therefore, these temperature drops become of considerable importance.

Wall Temperature Drops

The method of calculating the wall temperature drops for convection, advanced film, and transpiration cooling are presented in Ref. 17. The resulting temperature drop through a wall having a thickness of 0.050 inch (1.27 mm) is shown in Fig. 11 for convection cooling, full coverage film cooling, and transpiration cooling for ranges of turbine inlet temperature and gas pressure. Figure 11(b) shows the temperature drop for convection cooling. For these calculations, the temperature difference between the outside wall surface temperature and the cooling air was 800° F (444 K). When the temperature drop through the convection cooled wall becomes a significant portion of this 800° F (444 K) maximum possible temperature difference, convection cooling becomes very difficult. The very high wall temperature drops in Fig. 11(b) illustrate this difficulty.

The temperature drops through the wall for full coverage film cooling (Fig. 11(a)), are smaller than those for convection cooling. Temperature drops for two modes of transferring the heat with full coverage film cooling are shown. The dash-dot lines labeled "no inside wall convection" assume that the entire temperature rise of the coolant takes place within the wall in a manner somewhat similar to transpiration cooling. In order for heat transfer to occur in this manner, a large heat transfer surface is required within the wall as could possibly be obtained by having sharply slanted passages similar to those shown in Fig. 1(c). In some cases, however, shorter passages may be desired from the standpoint of fabrication and structural considerations. For such cases internal convection cooling would be required in conjunction with the film cooling in order to effectively use a significant portion of the potential temperature rise of the cooling air $(T_{B,o} - T_{c,i})$. The dashed curves in Fig. 11(a) illustrate the case where convection on the inside wall provides half of the temperature rise of the cooling air and the other half occurs due to convection within the film cooling holes. This mode of heat transfer results in higher temperature drops throughout the wall than for the case where all of the heat transfer to the cooling air occurs inside the film cooling passages, but, these temperature drops are still significantly lower than those occurring with convection cooling.

The temperature drops through the wall for transpiration cooling (Fig. 11(c)), are just slightly higher than those for full coverage film cooling where there is no inside wall surface convection. The difference between these temperature drops results from transpiration cooling materials having a lower effective thermal conductivity than full coverage film cooling. The calculations in Fig. 11(c) are based on a thermal conductivity of 8 Btu/(hr)(ft)($^\circ$F) (13.84 J/(sec)(m)(K)) which corresponds to a porosity of the material of about 15 percent.

Life

The results shown in Fig. 11 show that the temperature drop through the wall can vary up to 600° F (333 K) or higher. To determine how these temperature drops might affect turbine blade life, calculations were made of the lives that would be obtained. These life calculations considered both steady state creep damage and fatigue damage from thermal cycling for two arbitrarily chosen centrifugal stress levels - 26,000 pounds per square inch (179 N/mm^2) and a 50 percent greater stress (39,000 psi or 269 N/mm^2). For an advanced nickel base alloy, IN-100 (15Co-10Cr-5.5Al-4.7Ti-3Mo-Ni), and an outside wall surface temperature of 1800° F (1255 K), the following blade lives were obtained:

Wall temperature drop, °F (K)	Centrifugal stress, psi (N/mm²)	Calculated life, hr Without holes	With holes
48 (27)	26,000 (179)	117	117
300 (167)	26,000 (179)	2357	2328
480 (267)	26,000 (179)	462	421
600 (333)	26,000 (179)	1929	1341
48 (27)	39,000 (269)	12	12
300 (167)	39,000 (269)	231	228
480 (267)	39,000 (269)	76	80
600 (333)	39,000 (269)	575	399

The method of calculation and a more detailed description of some of the significance of these lives is given in Ref. 17. The trends shown are illustrative only, and actual lives would be dependent upon the application. The results show, however, that for a constant outer surface temperature (such as an oxidation limit) that high wall temperature drops are not necessarily detrimental. The higher temperature drops result in lower average wall temperatures. At lower temperatures the material is stronger and the resulting life is longer. The reduction in life in the above table for a temperature drop through the wall of 480° F (267 K) relative to the life for a temperature drop of 300° F (167 K) results from the fact that in that particular case life was not appreciably improved by stress relaxation as it was for the other cases. The stress relaxation at a wall temperature drop of 480° F (267 K) occurred primarily in the portion of the wall in compression from thermal stress. At other temperature drops the relaxation occurred in a portion of the wall that was in tension.

TURBINE MATERIALS

In previous discussions in this paper, material temperatures have been assumed with little justification given for the choice of values. The majority of the calculations were made at conservative temperature values for the present state-of-the-art. Calculations were also made to indicate how coolant flow requirements would vary if the required material temperatures were increased or decreased.

Blade and Vane Materials

Figure 12 illustrates the present and expected future temperature limitations for several classes of rotor blade and stator vane materials. At the present time, rotor blades are made exclusively of superalloys (nickel or cobalt base). Strength capabilities of the best of these superalloys limits their use temperature to slightly over 1800° F (1255 K) for rotor blades. Oxidation begins at lower temperatures (about 1700° F or 1200 K). As a result oxidation-corrosion protective coatings are required when long-life is required. Based upon research at NASA and other organizations, it is expected that the use temperature of superalloys for rotor blades can be extended to about 1900° F (1310 K) within the next few years.

At the present time dispersion and fiber strengthened materials have not found use in turbine rotor blades. Although TD-Ni and TD-NiCr (materials containing a thoria dispersoid) have been successful in increasing the use temperature for low stress applications, their stress carrying capabilities have been too low to use in rotor blades. Research on adding inert dispersoids or refractory fibers to superalloys has given indication that within the next few years such materials might find use in turbine rotor blades with metal temperatures up to 2200° F (1478 K). These materials will still require some form of surface protection from oxidation and corrosion for temperatures above 1700° F (1200 K). This surface protection may determine the maximum operating temperature as shown by the "coating durability limits" on Fig. 12.

At the present time two classes of the newer materials have the strength to be used in stator vanes at material temperatures above 2100° F (1422 K). These materials include superalloys and the dispersion strengthened nickel and nickel-chromium alloys, TD-Ni and TD-NiCr. The life of coatings, used to protect stator vane alloys from oxidation, limits the upper use temperature, particularly for long time operation. Within the next few years it is expected that the use temperature of the superalloys can be increased to about 2250° F (1506 K) and the dispersion or fiber strengthened superalloys to about 2350° F (1561 K).

It is unlikely that the use temperature can be increased very much higher because the melting point of some of the alloy constituents is being approached. Coatings to provide oxidation protection may also limit the upper temperature of these materials. To go to higher material operating temperatures will necessitate switching to refractory alloys. Stator vane material temperatures up to approximately 2400° F (1590 K) may be possible using the refractory columbium alloys. All refractory materials are extremely prone to oxidation, therefore, durable oxidation resistant coatings will be required. It is expected, therefore, that refractory alloys will be used only in applications that do not require long life because a coating failure from foreign object damage could cause a rapid oxidation failure of a refractory alloy.

Transpiration and Film Cooling Material Oxidation

Photographs of typical transpiration cooling and full coverage film cooling materials are shown in Fig. 13. In these materials, the coolant flow passages are small. Oxide films can block these passages and reduce the coolant flow. This blockage causes the material to become hotter due to poorer cooling, and further oxidation is accelerated. As a result, transpiration and full coverage film cooling are much more sensitive to oxidation than convection cooling. NASA research is being conducted on these oxidation problems. Figure 14 compares the effects of oxidation on the coolant flow rate through transpiration and full coverage film cooling materials. Conventional wire-form transpiration cooling material such as illustrated in Fig. 13(a) is compared in Fig. 14 for two alloys for a range of exposure times at 1800° F (1255 K). Alloy A is a conventional nickel-base alloy (Hastelloy X) commercially used in high temperature applications. At 1800° F (1255 K) oxidation is severe and wire-form transpiration cooling material becomes completely blocked in less than 400 hours of exposure time. Experimental tests on alloy B, an iron-chromium-aluminum-yttrium heating element alloy (GE 1541), show some data scatter, but do not indicate any blockage from oxidation after exposures up to 600 hours at 1800° F (1255 K). Research is continuing on methods for best fabricating this material into a wire-form transpiration cooling material.

Figure 14 also compares a full coverage film cooling material with the wire-form transpiration cooling material when both materials are made from the nickel-base alloy A (Hastelloy X). These two material forms initially had the same coolant flow rate per unit surface area for the same pressure drop across the material surface. Because of the larger, but fewer coolant passages in the full coverage film cooling material, its clogging tendency due to oxidation was greatly reduced compared to the wire-form transpiration cooling material. The results of these investigations indicates that transpiration cooling materials can probably be operated at material temperatures on the order of 1800° F (1255 K) within the near future. Even higher temperatures may be possible for full coverage film cooling materials. It may be possible to provide oxidation resistant coatings inside the film cooling holes to further reduce blockage by oxides.

OTHER FACTORS AFFECTING COOLED TURBINES

Air Cooling

This analysis considered only the cooling airflow requirements for first-stage turbine stator and rotor vanes and blades. It did not consider the problems of shroud cooling, nor the aerodynamic effects of the discharge of the cooling air into the gas stream.

Considering only the first stage of the turbine can be defended on the basis that cooling is most difficult for this stage due to the high temperature levels at this location. The results presented can also be used as an approximation for the second stage of the turbine if the turbine inlet temperatures are considered as temperatures at the inlet of the second stage.

An analysis was not made on the cooling requirements for the turbine shroud. Shroud cooling will become a very important requirement at high turbine inlet temperatures, and it must be considered. The general trends shown in this paper for blade and vane cooling should provide at least an indication of the relative merits of the three general air cooling schemes as applied to shroud cooling.

The aerodynamic effects of cooling air discharge on the turbine performance constitute a subject of too large a scope to include in this paper. In any cooled turbine design, however, the cooling airflow must be considered in the aerodynamic design. The cooling airflow effects naturally become larger and more important at the higher flow rates that will be experienced at very high turbine inlet temperatures.

In general, it can probably be concluded that for equal discharge of cooling air, convection cooled turbines will have superior aerodynamic performance than film or transpiration cooled turbines. The discharge of film or transpiration cooling air into the vane or blade boundary layer usually has deleterious effects on turbine performance, particularly if its effect was not considered properly in the aerodynamic design. The results of this present investigation, however, indicate that to achieve the very high turbine inlet temperatures and gas pressures that are expected for future engines, with air cooling, transpiration and/ or full coverage film cooling may be required. More research will be needed to evaluate the aerodynamic effects of transpiration and film cooling on turbine aerodynamic performance.

Liquid or Fuel Cooling

For most applications of turbine cooling, air cooling is much more practical than liquid cooling, particularly if the heat must ultimately be rejected to air anyway. If fuel can be used for cooling, the thermodynamic cycle of the engine is enhanced relative to air cooling. With fuel cooling the heat rejected from the turbine is put back into the cycle at the optimum location - in the combustor. Reference 18 investigated cooling stator vanes with either methane (for a supersonic transport application) or hydrogen (for a space shuttle application). The type of methane cooled design required for such an application is shown in Fig. 15. The design is aimed at keeping the surface temperature at a high value consistent with material oxidation and strength considerations to reduce the heat load to the coolant. Coolant wall surface temperature must be kept relatively low (on the order of 1000° F (811 K)) or lower to minimize fuel fouling or material carburization problems. These requirements necessitate use of a relatively low thermal conductivity thermal barrier between the coolant passage and the outside shell of the vane. Pressed metal fibers can be used for this purpose and they can have a range of thermal conductivities. For the vane design shown in Fig. 15, the required metal fiber material thermal conductivity was 1.4 Btu/(hr)(ft)($^\circ$F) (2.42 J/(sec)(m)(K)) for all of the vane except the trailing edge region. In this thinner trailing edge region, it was found that the thermal conductivity should be 0.52 Btu/(hr)(ft)($^\circ$F) (0.90 J/(sec)(m)(K)). A few of the resulting temperatures are shown for a vane designed for a hot spot gas temperature of 2490° F (1639 K) for Mach 3 flight cruise conditions. It was found that liquid or fuel cooling requires a relatively blunt trailing edge because this region of the vane must be cooled by conduction. At a distance of 1/4 inch (6.3 mm) from the cooling passage, the trailing edge temperature approaches 95 percent of gas temperature. It may be possible to utilize a heat pipe for cooling this region of the vane, but further research would be required to determine if such an approach would be feasible. Except for the cooling problems with long trailing edges, it was shown that relatively uniform circumferential temperature distributions could be obtained. The maximum difference in shell temperatures was less than 200° F (111 K).

These results indicate that for at least some applications liquid or fuel cooling can provide adequate cooling. The investigation revealed, however, that trailing edge cooling is critical unless a blunt trailing edge can be utilized. The fabrication of such blades is probably more complicated than for most air cooled blades. Problems such as coolant leakage could also be critical. It is probably safe to say that liquid or fuel cooling may find use in some special applications, but air cooling is much more flexible and is expected to be the primary method of cooling used in the foreseeable future.

CONCLUDING REMARKS

From the results of this investigation the following conclusions can be drawn:

1. Convection cooling of turbine blades and vanes becomes very difficult under the combination of high gas temperatures and pressures expected in future engines. High temperature drops through the blade or vane wall reduce the temperature difference between the inside wall surface and the cooling air and correspondingly reduce cooling effectiveness. Reducing wall thickness, developing materials with a higher temperature capability, or reducing cooling air temperature with heat exchangers will permit operation at higher gas temperatures and pressures, or reduce coolant flow requirements.

2. Reducing wall thickness from 0.050 to 0.020 inch (1.27 to 0.51 mm) could increase permissible turbine inlet temperature by as much as 500° F (278 K) for convection cooling.

3. Local turbine inlet temperatures up to 3500° F (2200 K) are possible for convection cooled stators if a material with a temperature capability of 2200° F (1478 K) can be used.

4. Reducing cooling air temperature from 1000° to 600° F (811 to 589 K) could permit increasing turbine inlet temperature as much as 500° F (278 K) for convection cooling.

5. Increasing allowable metal temperatures 100° F (56 K) or reducing cooling air temperature 200° F (111 K) can do more to improve the cooling capabilities of convection cooled blades and vanes than is possible by further improvements in the better convection cooling designs now available.

6. The coolant flow requirements for full coverage film and transpiration cooling are very much lower than for convection cooling at high turbine inlet temperatures and high compressor pressure ratios. These cooling methods can often permit turbine inlet temperatures 1000° F (556 K) higher than permitted with convection cooling at the same coolant flow rates. Cooling air temperature, and maximum allowable wall temperature have considerably less effect on the coolant flow requirements for transpiration and full coverage film cooling than for convection cooling.

7. Unless oxidation problems can be overcome in transpiration cooled materials to permit them to operate at temperatures approaching that possible with conventional turbine materials, transpiration cooling may not show advantages over full coverage film cooling except for extremely difficult cooling problems such as local turbine inlet temperatures in excess of about 3200° F (2030 K) at compressor ratios in excess of 20.

8. For a given maximum outside wall temperature (such as an oxidation temperature limit), high temperature drops through the wall thickness are not necessarily detrimental to blade life. Due to the lower average wall temperature with the high temperature drops, such blades or vanes may have longer life than the ones with lower temperature drops.

9. Recent research on both solid and porous turbine blade and vane materials show promise of increasing the permissible material temperatures for convection, film, and transpiration cooled turbine blades and vanes.

10. Air cooling appears more practical than liquid or fuel cooling for future engines.

APPENDIX - SYMBOLS

A	external blade surface area
A_f	coolant passage flow area
A_i	coolant passage surface area
B	defined by Eq. (12)
c_p	specific heat at constant pressure
D	blade or vane leading edge diameter
g	gravitational constant
h	gas-to-surface heat transfer coefficient
h_c	surface-to-coolant heat transfer coefficient
k	thermal conductivity
L	Reynolds number characteristic length (average blade or vane surface distance from leading edge)
M	Mach number
p	pressure
Pr	Prandtl number, $c_p \mu / k$
Q	heat transfer per unit time
Re	Reynolds number, $\rho V L / \mu$ or $\rho V D / \mu$
St	Stanton number, $h / \rho V c_p$
T	temperature
t	vane or blade wall thickness
V	velocity
w	coolant flow rate

η cooling air thermal effectiveness defined by Eqs. (1) and (2)

μ viscosity

ρ density

φ angle measured from stagnation point, deg

Subscripts:

avg   average

B   blade or vane wall

B, c   wall in contact with coolant

c   coolant

conv   convection cooling

e   effective or adiabatic

film   film cooling

g   gas

i   inside of blade or vane (coolant side), or inside surface

o   outside of blade or vane (gas side), or outside surface

ref   reference

s   at stator inlet

trans   transpiration cooling

Superscripts:

'   relative total

¯   average over dimension L

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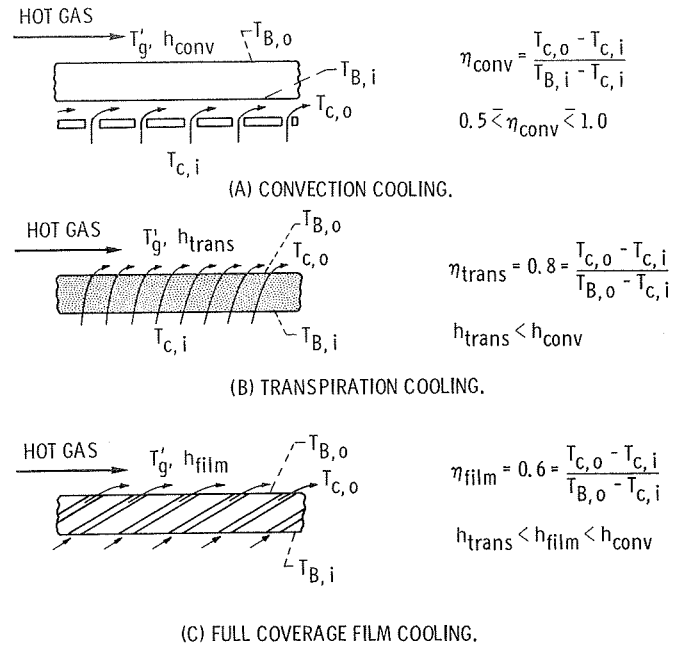
HOT GAS $\longrightarrow$ T'_g, h_{conv} $\ulcorner T_{B,o}$ $\urcorner T_{B,i}$

$T_{c,o}$

$T_{c,i}$

$$\eta_{conv} = \frac{T_{c,o} - T_{c,i}}{T_{B,i} - T_{c,i}}$$

$$0.5 \lesssim \eta_{conv} \lesssim 1.0$$

(A) CONVECTION COOLING.

HOT GAS $\longrightarrow$ T'_g, h_{trans} $\ulcorner T_{B,o}$ $T_{c,o}$

$T_{c,i}$ $\llcorner T_{B,i}$

$$\eta_{trans} = 0.8 = \frac{T_{c,o} - T_{c,i}}{T_{B,o} - T_{c,i}}$$

$$h_{trans} < h_{conv}$$

(B) TRANSPIRATION COOLING.

HOT GAS $\longrightarrow$ T'_g, h_{film} $\ulcorner T_{B,o}$ $T_{c,o}$

$\llcorner T_{B,i}$

$$\eta_{film} = 0.6 = \frac{T_{c,o} - T_{c,i}}{T_{B,o} - T_{c,i}}$$

$$h_{trans} < h_{film} < h_{conv}$$

(C) FULL COVERAGE FILM COOLING.

Figure 1. - Heat transfer models used in analysis.

η_{conv}

1.0 .7 .5

$$\eta_{conv,avg} = \frac{T_{c,o} - T_{c,i}}{T_{B,avg} - T_{c,i}}$$

□ ○ ◇ DETAILED DESIGNS

COOLING AIR TO COMPRESSOR AIRFLOW RATIO

AVERAGE METAL TEMPERATURE, °F

AVERAGE METAL TEMPERATURE, K
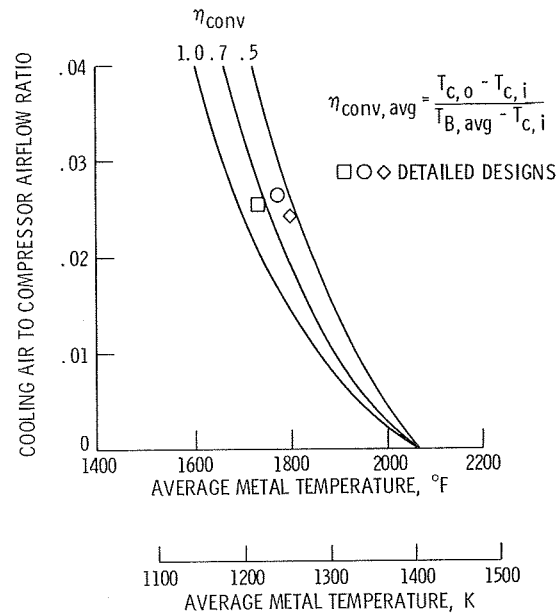
Figure 2. - Heat exchanger effectiveness of air
cooled turbine blades. Reference 1.

(A) COOLING AIR THERMAL EFFECTIVE-
NESS EFFECT $T_{c,i}$ = 1000° F (811 K).
$T_{B,o}$ = 1800° F (1255 K).

(B) COOLING AIR TEMPERATURE
EFFECT. $T_{B,o}$ = 1800° F (1255 K).

(C) METAL TEMPERATURE EFFECT
$T_{c,i}$ = 1000° F (811 K).
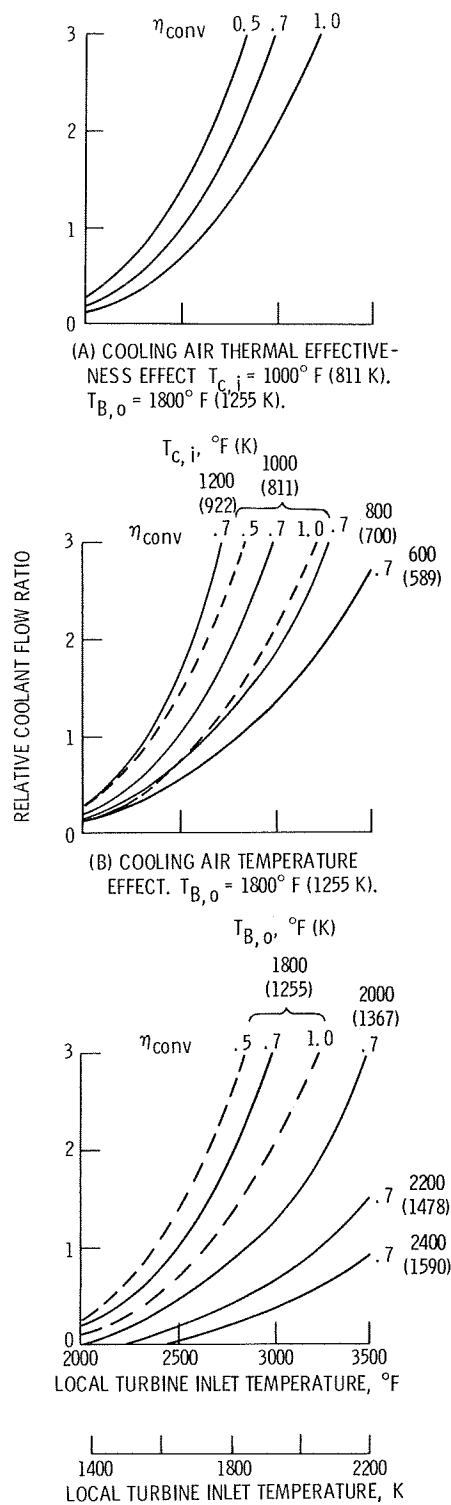
Figure 3. - Relative coolant flow requirements
for convection cooling of aft section of stator
vanes. $p'_{g,s}$ = 20 atmos. (202.6 N/cm²).

(A) COOLANT FLOW.

(B) WALL TEMPERATURE DROP
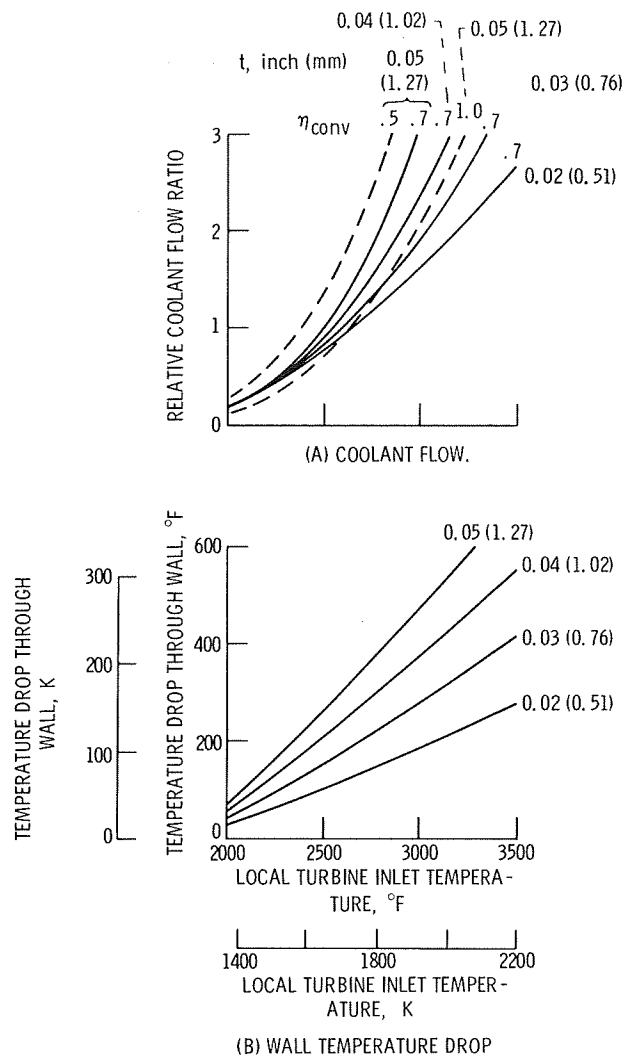
Figure 4. - Effect of wall thickness on relative coolant flow
requirements and wall temperature drop for convection
cooling aft section of stator vanes. $p'_{g,s}$ = 20 atmos.
(202.6 N/cm²), $T_{c,i}$ = 1000° F (811 K). $T_{B,o}$ = 1800° F
(1255 K).

PRESSURE, $p'_{g,s}$, ATMOS.

(A) COOLANT FLOW.

(A) HEAT TRANSFER COEFFICIENTS.

CONVECTION

FILM

TRANSPIRATION

CONVECTION

FILM

TRANSPIRATION

$T_{B,c} - T_{c,i}$
$T_{c,o} - T_{c,i}$

TRANSPIRATION AND FILM

TRANSPIRATION

FILM

CONVECTION

CONVECTION

(C) COOLING AIR TEMPERATURE RISE.

CONVECTION

FILM

TRANSPIRATION

(B) HEAT FLUX TO VANE.

CONVECTION

FILM

TRANSPIRATION

(D) RESULTING COOLANT FLOW REQUIREMENTS.
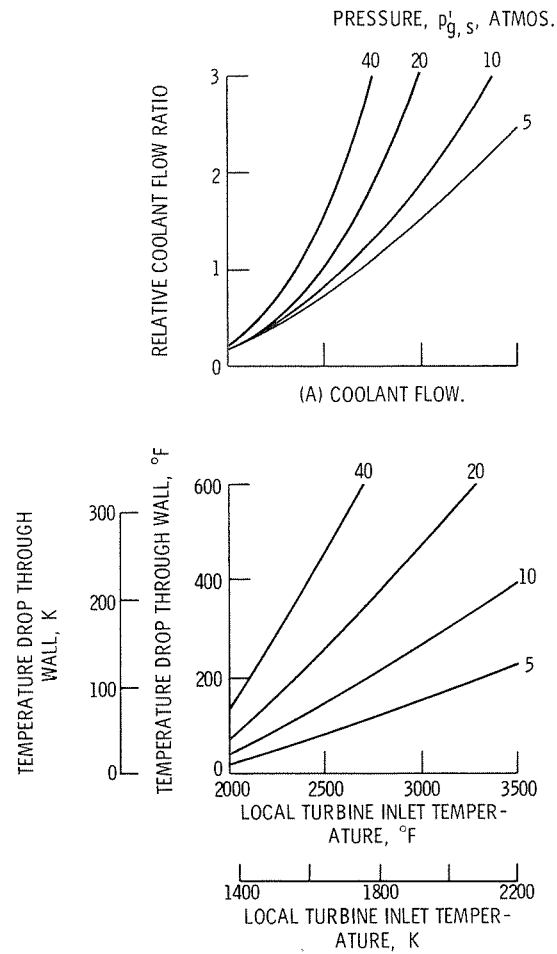
Figure 5. - Effect of gas pressure on relative coolant flow requirements and wall temperature drop for convection cooling aft section of stator vanes. $T_{c,i}$ = 1000° F (811 K), $T_{B,o}$ = 1800° F (1255 K), t = 0.050 inch (1.27 mm), η_{conv} = 0.7.
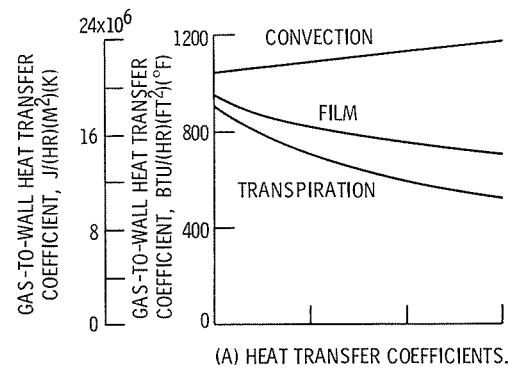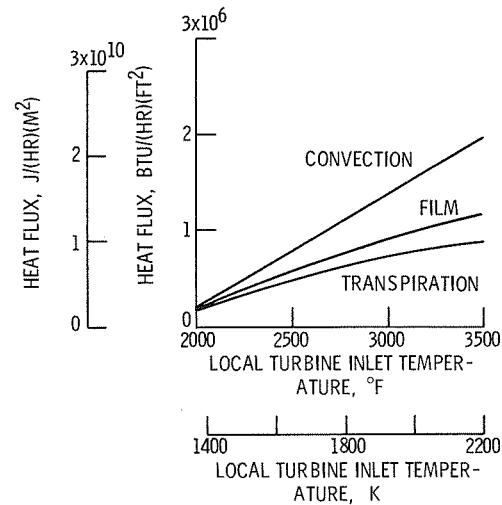
Figure 6. - Comparison of convection full coverage film, and transpiration cooling heat transfer effects for aft section of stator vanes. $T_{B,o}$ = 1800° F (1255 K), $T_{c,i}$ = 1000° F (811 K), η_{conv} = 0.7, t = 0.050 inch (1.27 mm) $p'_{g,s}$ = 20 atmos. (202.6 N/cm²) η_{film} = 0.6, η_{trans} = 0.8.

Figure 6. - Concluded.

FULL COVERAGE FILM COOLING
TRANSPIRATION COOLING
CONVECTION COOLING, $\eta_{conv} = 0.7$

(A) AFT SECTION OF STATOR VANES $T_{B,o} = 1800°$ F (1255 K).

(B) AFT SECTION OF ROTOR BLADES. $T_{B,o} = 1700°$ F (1200 K).
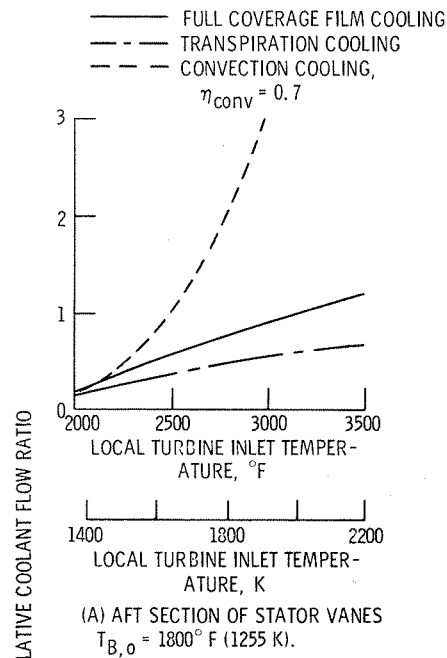
Figure 7. - Full coverage film and transpiration cooling as compared with convection cooling. $T_{c,i} = 1000°$ F (811 K), $p'_{g,s} = 20$ atmos. (202.6 N/cm²), $t = 0.50$ inch (1.27 mm).
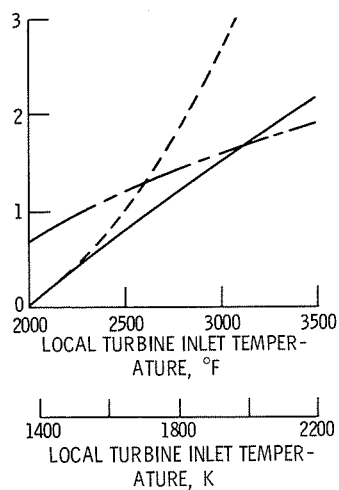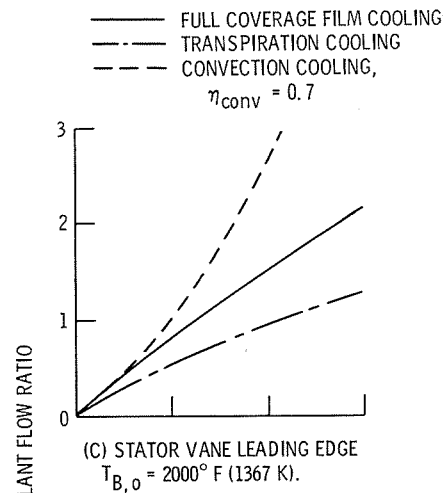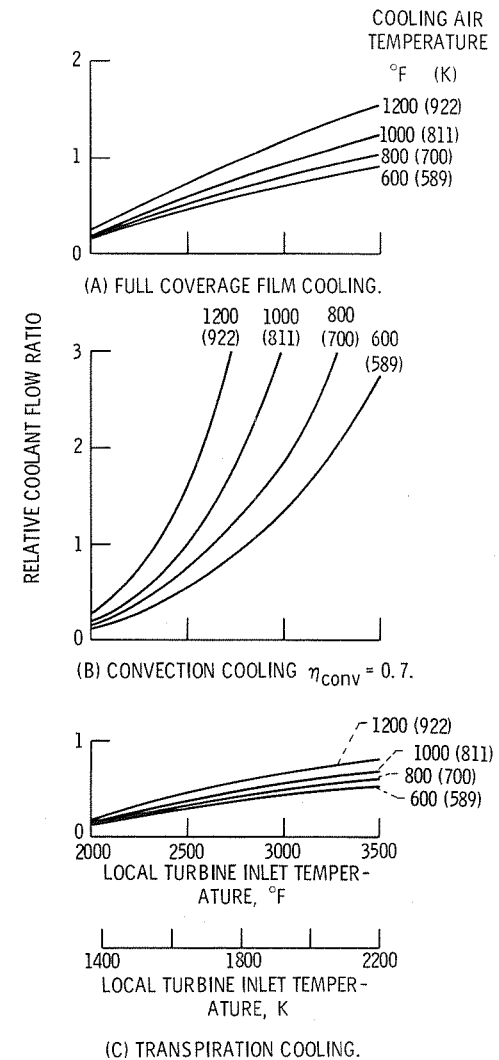
FULL COVERAGE FILM COOLING
TRANSPIRATION COOLING
CONVECTION COOLING, $\eta_{conv} = 0.7$

(C) STATOR VANE LEADING EDGE $T_{B,o} = 2000°$ F (1367 K).

(D) STATOR VANE LEADING EDGE WITH REDUCED METAL TEMPERATURE FOR TRANSPIRATION COOLING $T_{B,o} = 2000°$ F (1367) FOR CONVECTION AND FILM COOLING, $T_{B,o} = 1600°$ F (1145 K) FOR TRANSPIRATION COOLING.

Figure 7. - Concluded.

COOLING AIR TEMPERATURE
°F   (K)
1200 (922)
1000 (811)
800 (700)
600 (589)

(A) FULL COVERAGE FILM COOLING.

(B) CONVECTION COOLING $\eta_{conv} = 0.7$.

(C) TRANSPIRATION COOLING.

Figure 8. - Effect of cooling air temperature on relative coolant flow requirements for aft section of stator vanes. $T_{B,o} = 1800°$ F (1255 K), $t = 0.050$ inch (1.27 mm) $p'_{g,s} = 20$ atmos. (202.6 N/cm²).

(A) FULL COVERAGE FILM COOLING.

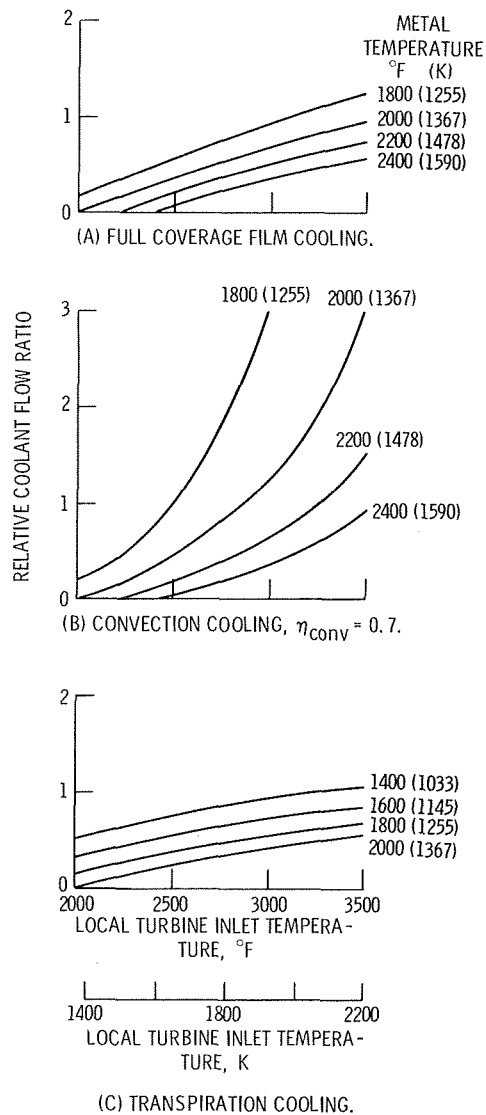(B) CONVECTION COOLING, $\eta_{conv} = 0.7$.

(C) TRANSPIRATION COOLING.

Figure 9. - Effect of stator vane aft section metal temperature on relative coolant flow requirements. $T_{c,i} = 1000^\circ$ F (811 K), t = 0.050 inch (1.27 mm) $p'_{g,s} = 20$ atmos. (202.6 N/cm²).
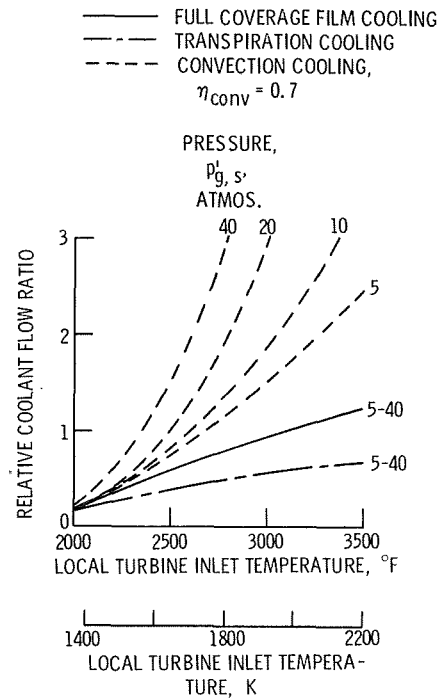

Figure 10. - Gas pressure effects on full coverage film and transpiration cooling for aft section of stator vanes. $T_{c,i} = 1000^\circ$ F (811 K), $T_{B,o} = 1800^\circ$ F (1255 K), t = 0.050 inch (1.27 mm).
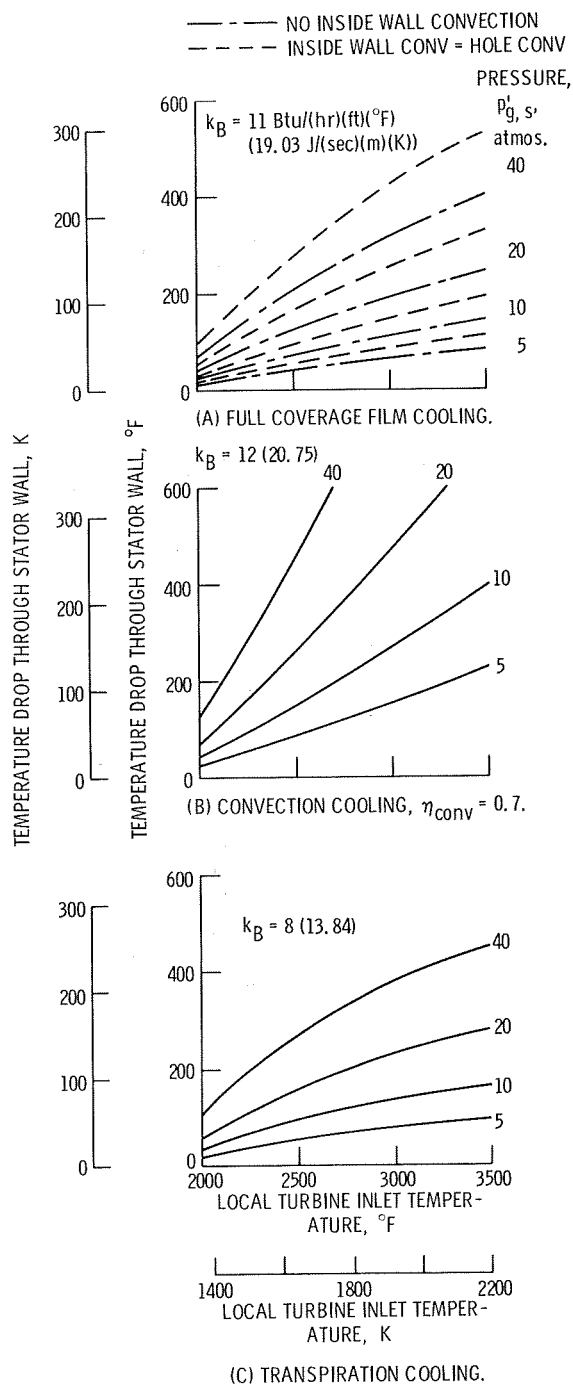
Figure 11. - Temperature drop through 0.050 inch (1.27 mm) wall of aft section of stator vanes. $T_{B,o} = 1800°$ F (1255 K), $T_{c,i} = 1000°$ F (811 K).
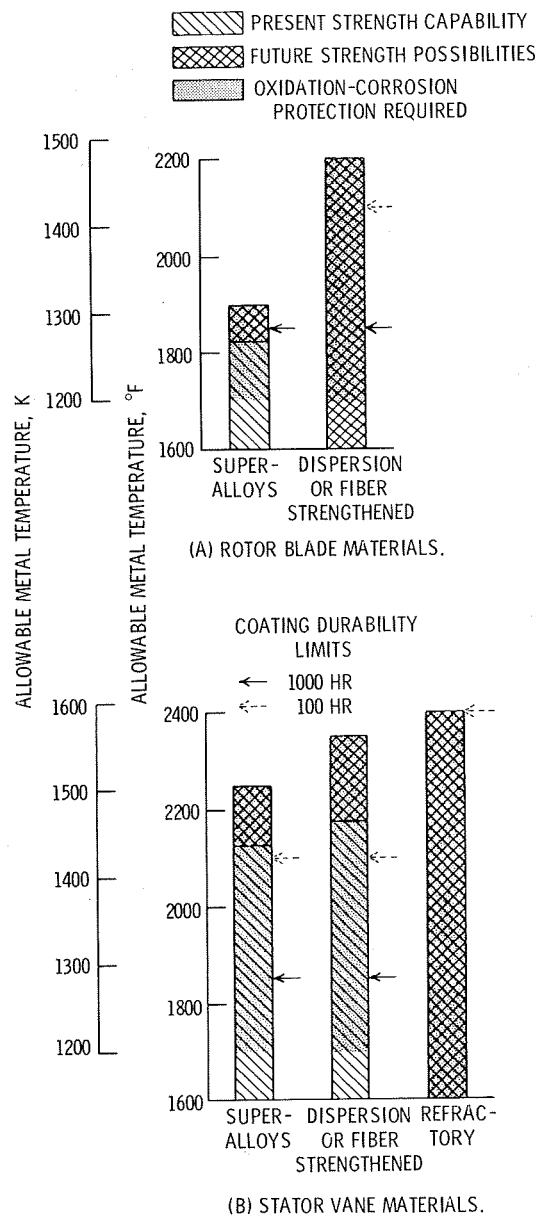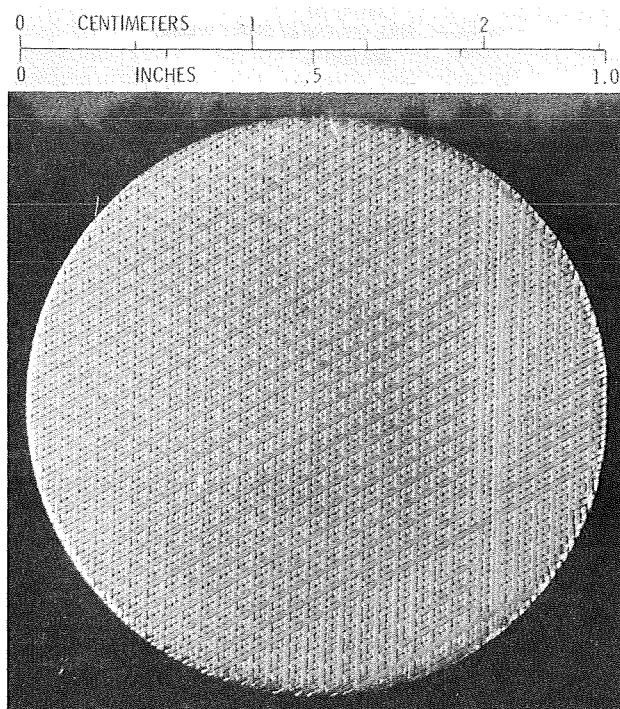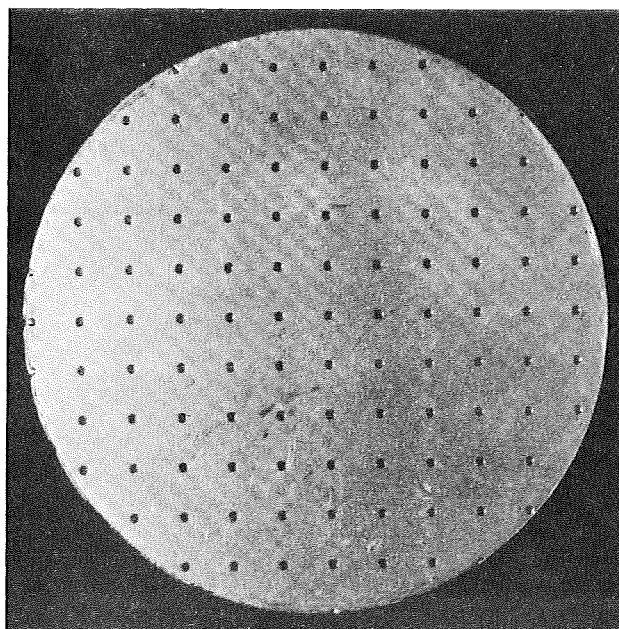

Figure 12. - Turbine material temperature capabilities.

(A) WIRE FORM TRANSPIRATION COOLING MATERIAL.

(b) FULL COVERAGE FILM COOLING MATERIAL.

Figure 13. - Photograph of surfaces of transpiration and film cooling materials.

————— WIRE FORM TRANSPIRATION COOLING
MATERIAL
— — — FULL COVERAGE FILM COOLING
MATERIAL

ALLOY A - 22Cr-18.5Fe-9Mo-1.5Co-Ni (HASTELLOY X)
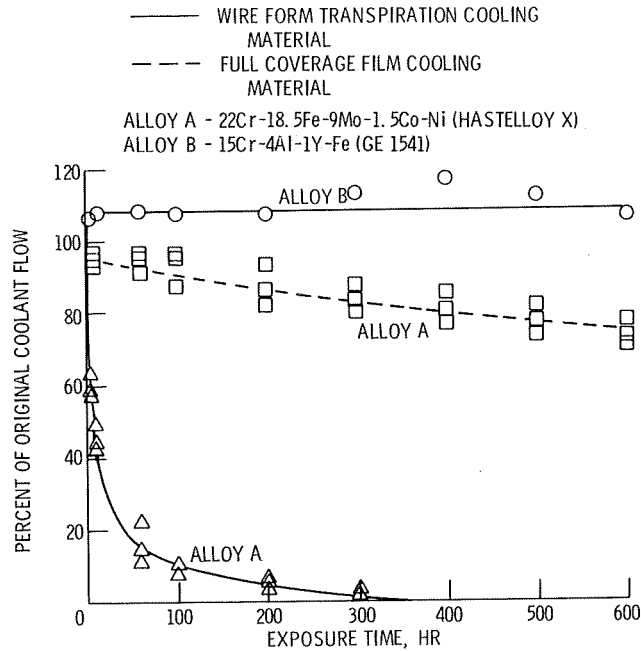ALLOY B - 15Cr-4Al-1Y-Fe (GE 1541)

Figure 14. - Coolant flow reduction from oxidation of transpiration and film cooling materials at 1800° F (1255 K).
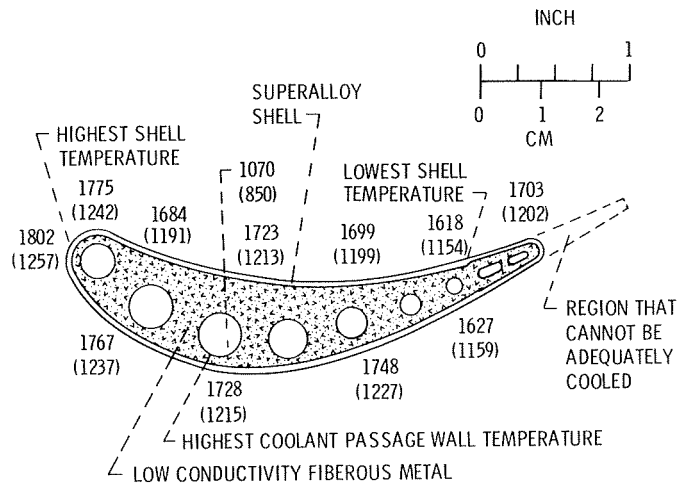

Figure 15. - Temperature distribution in a methane cooled vane under cruise conditions. Temperatures in °F, (K).