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**COMPARISON OF
COOLING EFFECTIVENESS
OF TURBINE VANES WITH
AND WITHOUT FILM COOLING**

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SUMMARY

The cooling effectiveness of three film-cooled vanes, each with different internal cooling configurations, was investigated. The cooling effectiveness of two film-cooled vanes was compared to the cooling effectiveness of two non-film-cooled vanes of similar internal cooling geometry. The cooling effectiveness of the third vane was compared to a non-film-cooled vane which had significantly different internal cooling geometry. The cooling effectiveness of the non-film-cooled vanes was analytically adjusted to account for the geometrical differences that did exist between the vanes that were compared.

The cooling effectiveness values used in the comparisons were obtained from experimental correlations, most of which have been published in the literature. The test conditions from which the correlations were obtained were: gas temperatures from 1370 to 1644 K (2500^o to 2980^o F), a gas pressure of about 31 N/cm² (45 psia), coolant temperatures from 310 to 811 K (590^o to 1500^o F), and coolant- to gas-flow ratios from 0.02 to 0.12.

The results of the comparison indicated that, for the vane configurations and test conditions examined, film cooling had an adverse effect near the suction-surface trailing edge of the vanes. It was postulated that the addition of film cooling caused a laminar or transitional boundary layer to become transitional or turbulent, and be reflected in an increased heat transfer coefficient. Film cooling was found to be beneficial to the cooling of the pressure surface of the vanes.

INTRODUCTION

The relative benefits of adding film cooling to a turbine vane were investigated by comparing the temperature data of vanes without film cooling to vanes of similar internal geometry but with film cooling added.

Advances in turbojet engine design require increasing turbine inlet temperatures. As turbine inlet temperatures increase above 1644 K (2500^o F) more attention is being given to various types of film cooling for turbine vanes and blades. Local film cooling and full-coverage film cooling are two possible ways of maintaining acceptable wall temperature levels on turbine vanes. To adequately predict the temperatures of film-cooled vanes exposed to high turbine inlet temperatures, it is necessary to know what the film-cooling effectiveness will be in an actual engine environment. Reference 1 presented test data and correlation equations for film-cooled vanes tested in an engine environment. This reference also indicated some adverse effects of film cooling on the suction-surface trailing edge regions of the vanes investigated.

In the present report, the cooling effectiveness of three film-cooled vanes was compared with the cooling effectiveness of two non-film-cooled vanes. On two of the film-cooled vanes, film-cooling air was ejected aft of the leading edge, near the "gill" region. Film-cooling air was ejected through a row of holes in one vane, and through a row of short slots in the other vane. The third vane had a row of short slots near the suction-surface trailing edge. In two cases, the internal cooling geometries of the film-cooled and the non-film-cooled vanes being compared were similar. In the third case, the internal geometry of the vanes being compared was significantly dissimilar. A calculation procedure was used to correct for the geometrical differences between the vanes being compared.

Experimentally obtained correlation equations, most of which have been published in references 1 to 6, were used to obtain the cooling-effectiveness values used in these comparisons. The experimental data were obtained from heat transfer tests conducted in a turbojet engine and a four-vane cascade. The test conditions from which the correlations were obtained were: gas temperatures from 1370 to 1644 K (2007^o to 2500^o F), gas pressures about 31 N/cm² (45 psia), coolant temperatures from 310 to 811 K (98^o to 1000^o F), and coolant- to gas-flow ratios from 0.02 to 0.12.

SYMBOLS

A	flow area
a, b	constants in eq. (3)
D _e	equivalent diameter
h	heat transfer coefficient
k	thermal conductivity
L	length along suction or pressure surface
Pr	Prandtl number

S surface area
T temperature
 \dot{w} local mass flow rate
x distance from stagnation point
 α, β constants in eq. (1)
 λ $h_g S_g / h_c S_c$
 μ viscosity
 ϕ cooling effectiveness, $(T_{ge} - T_w) / T_{ge} - T_{ci}$

Subscripts:

A vane A
B vane B
c coolant side
ci coolant inlet
g gas side
ge gas, effective
w wall
x local conditions at distance **x**

APPARATUS

A modified J-75 turbojet engine and a four-vane cascade, designed for J-75 sized vanes, were utilized to obtain the results presented in this report. Both of these facilities are capable of being operated at gas temperatures up to about 1644 K (2500° F). These facilities are discussed in detail in reference 7.

Research Engine

The research engine incorporated a modified high pressure spool and combustor assembly from a two-spool J-75 turbojet engine. A major feature of this engine was the provision for two separate and distinct cooling air systems in the stator assembly. The turbine stator consisted of 72 vanes. A cluster of five of these vanes served as test

vanes. Cooling air was supplied to these vanes from a laboratory air system independent of the system which supplied cooling air to the remaining 67 slave vanes.

Cascade

The cascade test section was an annular sector of a vane row and contained four vanes and five flow channels. The two central vanes served as the test vanes. Cooling air was supplied to these two vanes by a common laboratory air system.

Test Vane Description

The vane configurations discussed in this report had the same external profile which is described in detail in reference 8. The vane span was 9.78 centimeters (3.85 in.), and the midspan chord length was 6.27 centimeters (2.47 in.).

Vane A. - A cross-sectional schematic of vane A is shown in figure 1(a). The major features of this vane were film cooling on the suction and pressure surfaces starting just aft of the leading edge, an impingement cooled leading edge, and convection cooling augmented by chordwise fins in the midchord region.

Cooling air entered the vane at the vane tip, and was ducted into a central plenum. From this central plenum, part of the air passed through a row of impingement holes to impingement cool the vane leading edge. The cooling air then flowed in a chordwise direction and exited through the film-cooling holes located at the "gill" region on the suction and pressure sides of the vane. The remaining cooling air flowed in a chordwise direction and exited at the vane trailing edge, as shown in figure 1(a).

The film-cooling holes on the suction side of the vane consisted of a single row of 58 holes, having a diameter of 0.064 centimeter (0.025 in.), and a centerline spacing of 0.157 centimeter (0.062 in.). The holes were located at a surface distance of 2.26 centimeters (0.89 in.) from the leading-edge stagnation point, and were inclined at a 28° angle to the vane suction surface. The film-cooling holes on the pressure side of the vane consisted of a double row of 58 and 59 holes, having a diameter of 0.071 centimeter (0.028 in.) and a centerline spacing of 0.157 centimeter (0.062 in.). The holes were located at a surface distance of 1.78 centimeters (0.70 in.) and 2.16 centimeters (0.85 in.), respectively, from the leading-edge stagnation point, and were inclined at a 40° angle to the vane pressure surface. The second row of holes was offset in the spanwise direction by one-half the centerline spacing.

The width of the split trailing edge was 0.051 centimeter (0.020 in.), and was maintained by five spacers, which formed six equally spaced channels 1.384 centimeters

(0.545 in.) in the spanwise direction, 0.64 centimeter (0.25 in.) in the chordwise direction, and 0.051 centimeter (0.020 in.) wide.

Vane B. - A cross-sectional schematic of vane B is shown in figure 1(b). The internal configuration of this vane was similar in design to vane A. The notable differences were the elimination of film cooling, the extension of the midchord fins forward towards the leading edge, and the increased width of the split trailing edge from 0.051 to 0.076 centimeter (0.020 to 0.030 in.). The cooling-air flow within vane B was similar to that of vane A with the exception that, since no film-cooling holes were provided, the cooling air for the vane leading edge flowed into collection chambers, and exited at the vane hub. A more detailed description is contained in references 1 and 4.

Vane C. - A cross-sectional schematic of vane C is shown in figure 1(c). The major features of this vane were film-cooling slots on the suction and pressure surfaces, starting aft of the leading edge, an impingement-cooled leading edge, and convection cooling, augmented by spanwise fins in the midchord region. This is in contrast to the chordwise fins used in vanes A and B.

Cooling air entered the vane at the vane tip. From a plenum chamber contained within the vane tip platform (not shown in fig. 1(c)), part of the air was distributed into a leading edge plenum, and passed through a row of impingement holes to impingement cool the leading edge. The cooling air then exited through film-cooling slots located at the "gill" region on the suction and pressure sides of the vane. The remaining cooling air flowed radially inward through the passages formed by the spanwise fins and the midchord plenum walls. Near the vane hub the cooling air from the passages was collected in the midchord plenum; it then flowed in a chordwise direction through the split trailing edge.

The film-cooling slots on the suction side of the vane consisted of a row of ten identical spanwise slots, each 0.394 centimeter (0.155 in.) in the spanwise direction, and 0.056 centimeter (0.022 in.) in the chordwise direction. The slots were located at a surface distance of 2.67 centimeters (1.05 in.) from the leading edge stagnation point, and were inclined at a 28° angle to the vane suction surface. The film-cooling slots on the pressure side of the vane consisted of a row of fifteen spanwise slots, which were separated into three groups by size. The five slots at the vane tip were 0.427 centimeter (0.168 in.) long by 0.094 centimeter (0.037 in.) wide. The five midspan slots were 0.427 centimeter (0.168 in.) long by 0.061 centimeter (0.024 in.) wide. The five hub slots were 0.437 centimeter (0.172 in.) long by 0.056 centimeter (0.022 in.) wide. The slots were located at a surface distance of 2.43 centimeter (0.955 in.) from the leading-edge stagnation point, and were inclined at a 30° angle to the vane pressure surface.

The width of the split trailing edge was 0.076 centimeter (0.030 in.), and was maintained by twelve spacers, which formed thirteen equally spaced channels, 0.577

centimeter (0.227 in.) in the spanwise direction, and varied from 1.11 to 1.60 centimeters (0.438 to 0.630 in.) in the chordwise direction.

Vane D. - A cross-sectional schematic of vane D is shown in figure 1(d). The internal configuration of this vane was similar in design to vane C. The notable differences were the elimination of film cooling, and the increased width of the split trailing edge from 0.076 to 0.102 centimeter (0.030 to 0.040 in.). The cooling-air flow within vane D was similar to that of vane C with the exception that, since no film-cooling slots were provided, the cooling air for the vane leading edge flowed through the chordwise passages at the "gill" region, rejoined the midchord-cooling air in the midchord plenum, and exited through the split trailing edge.

Vane E. - A cross-sectional schematic of vane E is shown in figure 1(e). The major features of this vane were impingement-cooled leading edge and midchord regions, and a film-cooled split trailing edge, which contained pin fins.

Cooling air entered the vane at the vane tip. From a plenum chamber contained within the vane tip platform (not shown in fig. 1(e)), part of the air was distributed into the leading-edge impingement tube. The leading edge cooling air then flowed through slots to impingement cool the leading edge. This air emptied into a collection chamber, was ducted radially inward, and exhausted at the vane hub. The remaining cooling air entered the midchord plenum, flowed through impingement orifices in the impingement insert, and then left the vane through film-cooling slots and the split trailing edge.

The film-cooling slots on the suction surface consisted of a row of eight identical slots, each 1.11 centimeters (0.438 in.) in the spanwise direction, by 0.051 centimeter (0.020 in.). The slots were located at a surface distance of 4.93 centimeters (1.94 in.) from the leading-edge stagnation point, and were inclined at a 12° angle to the vane suction surface. The film-cooling slot on the pressure side of the vane consisted of a continuous slot 9.27 centimeters (3.65 in.) in the spanwise direction, by 0.064 centimeter (0.025 in.). The slot was located at a surface distance of 4.04 centimeters (1.59 in.) from the leading-edge stagnation point, and was inclined at an 18° angle to the vane pressure surface.

There were four rows of oblong pin-fins in the trailing edge, whose major and minor axes were 0.38 and 0.25 centimeter (0.15 and 0.10 in.), respectively. Since the trailing edge of vane E was tapered, the height of these oblong pin-fins varied from 0.152 to 0.076 centimeter (0.060 to 0.030 in.). There was also a single row of round pin-fins with a diameter of 0.20 centimeter (0.08 in.), and a height of 0.064 centimeter (0.025 in.). Reference 6 discusses additional features of this vane.

INSTRUMENTATION

The discussion of instrumentation will be limited to the gas temperature measurements and the vane temperature measurements. Details on the other aspects of the instrumentation are discussed in reference 1 for the engine, reference 6 for the cascade, and reference 7 for operational instrumentation common to both facilities.

Gas Temperature Measurements

The research engine contained eight circumferentially spaced, radially traversing gas temperature probes located at the turbine stator inlet. Each probe was programmed to stop automatically at nine radial positions, where temperatures were recorded. The eight radial temperature profiles generated were averaged to form a single radial temperature profile. Since the principle plane of instrumentation on the test vanes is at the midspan, the temperature used in the correlation of the data is based on the average of the middle three of the nine radial positions.

The total inlet temperature profile in the cascade was measured at 12 radial locations by a similar radially traversing probe located in front of the channel formed by the two test vanes. The average gas temperature used in the correlation of the data was obtained by averaging the temperatures at the two middle positions of the traversing probe.

Vane Temperatures

The vane temperatures were measured by 0.051-centimeter - (0.020-in. -) diameter sheathed, Chromel-Alumel thermocouples embedded in slots in the vane wall. A more detailed discussion of these type of thermocouples and their installation is made in reference 9. The thermocouple locations of interest for each vane considered are shown in figures 1(a) to (e) for vanes A to E, respectively. Table I gives the dimensionless chord-wise surface distance of each thermocouple relative to the leading-edge stagnation point. Vanes A and B had eight similar thermocouple locations. Vanes C and D had 13 similar thermocouple locations. Vane E had two thermocouple locations of interest.

ANALYSIS METHODS

The effectiveness of film cooling can be shown by a comparison of the local cooling effectiveness of a film-cooled and a similar non-film-cooled vane. Two methods of presenting the local cooling effectiveness are examined in reference 6. A correlation of the

temperature difference ratio ϕ (defined as $(T_{ge} - T_w)/(T_{ge} - T_{ci})$) against the coolant-to gas-flow ratio \dot{w}_c/\dot{w}_g was found satisfactory for film-cooled and non-film-cooled regions of the vane reported. This correlation takes the following form:

$$\phi = \left[1 + \alpha \left(\frac{\dot{w}_c}{\dot{w}_g} \right)^\beta \right]^{-1} \quad (1a)$$

where α and β are constants obtained from a least-squares fit of the experimental data, \dot{w}_g is the gas flow per vane channel, and \dot{w}_c is the local coolant flow per vane. For those thermocouple locations of interest that were both internally convection cooled and film cooled, reference 1 shows that the data are correlated by using only the internal coolant flow.

Reference 6 shows that, from a one-dimensional heat balance, ϕ can also be related to a ratio of gas-to-coolant heat transfer coefficients.

$$\phi = \left(1 + \frac{h_g S_g}{h_c S_c} \right)^{-1} \quad (1b)$$

Comparing equation (1a) and equation (1b), it can be seen that the heat transfer coefficient ratio $h_g S_g/h_c S_c$ is equal to $\alpha(\dot{w}_c/\dot{w}_g)^\beta$.

This experimental correlation (eq. (1a)) is a useful tool for comparing the cooling effectiveness of different vane designs tested in the same facility. In this report, the equations obtained from the curve fit of experimental data are used in comparing the film-cooling effectiveness of the vane designs.

Since the purpose of this investigation is to examine the effects of film cooling by comparing a film-cooled and a non-film-cooled vane, the differences in those parameters not directly associated with film cooling must be taken into account. If the data being compared were for similar gas and coolant conditions, then the major parameter that can differ between the two vanes is the internal coolant passage geometry.

Since, from equations (1a) and (1b), $h_g S_g/h_c S_c$ equals $\alpha(\dot{w}_c/\dot{w}_g)^\beta$, a correction needs to be applied to the coolant-side heat transfer coefficient to account for the internal geometry differences. For example, to calculate a modified ϕ for vane B, which is to be compared with vane A, then the term $\alpha(\dot{w}_c/\dot{w}_g)^\beta$ for vane B must be multiplied by the heat transfer coefficient ratio $h_{c,B}/h_{c,A}$; that is,

$$\left(\frac{h_g S_g}{h_c S_c} \right)_B \frac{h_{c,B}}{h_{c,A}} = \alpha_B \frac{h_{c,B}}{h_{c,A}} \left(\frac{\dot{w}_c}{\dot{w}_g} \right)^{\beta_B} \quad (2)$$

where the comparison is made for equal local coolant- to gas-flow ratios for vane A and vane B. The ratio of coolant-side heat transfer coefficients can be obtained from the appropriate Nusselt number relationship. In a simplified case, this ratio would be

$$\frac{h_{c,B}}{h_{c,A}} = \frac{\left[\frac{k}{D_e} \left(\frac{\dot{w}_c}{A} \frac{D_e}{\mu} \right)^a \text{Pr}^b \right]_B}{\left[\frac{k}{D_e} \left(\frac{\dot{w}_c}{A} \frac{D_e}{\mu} \right)^a \text{Pr}^b \right]_A} = \left(\frac{D_{e,A}}{D_{e,B}} \right)^{1-a} \left(\frac{A_A}{A_B} \right)^a \quad (3)$$

For the analysis herein, the exponent a was assumed to be 0.8. The right term of equation (2), $\alpha_B (h_{c,B}/h_{c,A}) (\dot{w}_c/\dot{w}_g)^{\beta_B}$, can then be substituted for the term $\alpha_B (\dot{w}_c/\dot{w}_g)^{\beta_B}$ in equation (1a) to obtain a modified ϕ for vane B.

RESULTS AND DISCUSSION

The results presented herein for the five vanes are based on vane metal temperature correlations that were originally presented in references 1 to 6. All of the correlation data were obtained from experimental investigations made in an engine and/or a four-vane cascade. The test conditions from which the correlations were obtained were:

Gas temperature, K ($^{\circ}$ F)	1370 to 1644 (2007 to 2500)
Gas pressure, N/cm ² (psia)	31 (45)
Coolant temperature, K ($^{\circ}$ F)	310 to 811 (98 to 1000)
Total coolant- to gas-flow ratio	0.02 to 0.12

For convenience in comparing the effects of film cooling, the vanes were grouped into "sets," according to their general configuration. The sets were formed as follows:

Set number	1	2	3
Vane configuration	A and B	C and D	B and E
Facility in which data were generated	Engine and four-vane cascade	Engine	Engine and four-vane cascade

The vanes within sets 1 and 2 had internal cooling configurations that were similar. However, the internal configurations of the vanes in set 3 were significantly dissimilar. Each set contained one vane that incorporated film cooling, and one vane that did not incorporate film cooling. The effects of film cooling were evaluated by comparing the temperature difference ratio ϕ between the film-cooled and the non-film-cooled vanes within each set. In this report, the suction surface and the pressure surface data are treated separately. The overall results of the study are presented in the following paragraphs.

Suction Surface

Cooling effectiveness (engine). - The local cooling effectiveness on the suction surface of vanes A, B, C, D, and E were obtained from equation (1a) and the experimental correlation constants listed in table II(a). Some representative curves showing the variation in cooling effectiveness as a function of the coolant- to gas-flow ratio \dot{w}_c/\dot{w}_g for the vanes in each set are shown in figures 2 to 4. The curves shown in these figures were for thermocouple location 1 located near the trailing edge (see fig. 1). The cooling effectiveness shown in figures 2 to 4 was determined for vanes that had been investigated in an engine environment.

Figure 2 compares the local cooling effectiveness for set 1 vanes. The solid lines represent the results obtained directly from the correlation constants of table II(a) and equation (1a). At the location of thermocouple 1, the internal coolant passage geometries of vanes A and B are the same except that the width of the trailing edge coolant passage of vane A is less than that of vane B. For equal quantities of coolant flow rate, the coolant flow per unit area through vane A's split trailing edge would be greater than the coolant flow per unit area through vane B's split trailing edge. Therefore, for equal quantities of coolant flow rate, the coolant side heat transfer coefficient for vane A at the trailing edge would be greater than that of vane B. This reasoning would lead to the expectation that, for a given set of gas-side conditions, vane A would have a higher cooling effectiveness than vane B. Observation of the solid line curves in figure 2 indicate the opposite result; that is, the cooling effectiveness of vane B (non-film-cooled) was higher than that of vane A (film-cooled) throughout the entire range of coolant flow ratios. The

dashed line in figure 2 shows the cooling effectiveness of vane B when analytically modified to account for the difference in trailing edge passage widths. The modified cooling effectiveness curve of vane B is shown to be improved over the uncorrected curve. From the results shown in figure 2, it is apparent that at thermocouple location 1, vane B (non-film-cooled) is cooled more effectively than vane A (film-cooled). Similar cooling effectiveness comparisons were made for vanes A and B at thermocouple locations 2, 3, and 4. These curves were not shown in order to avoid repetitiveness. The results at thermocouple locations 2 and 3 were similar to those discussed for thermocouple 1. The results for thermocouple location 4 indicated that the cooling effectiveness of vane A (film-cooled) was higher than the cooling effectiveness of vane B (non-film-cooled) after the cooling effectiveness of vane B was modified to account for the presence of chordwise fins in vane B.

Figure 3 compares the cooling effectiveness of set 2 vanes. The internal coolant passage geometries of vanes C and D are the same except that the width of the split trailing edge of vane C is less than that of vane D. The solid lines represent the results obtained using the correlation constants of table II(a), and show that the cooling effectiveness of vane C was higher than that of vane D throughout the range of the coolant flow ratios. The dashed curve in figure 3 shows the cooling effectiveness of vane D when analytically modified to account for the difference in trailing edge passage widths. Comparison of vane C results with the modified vane D (dashed curve) results now indicate that, at thermocouple location 1, vane D (non-film-cooled) was cooled more effectively than vane C (film-cooled).

Although not shown, cooling effectiveness comparisons were made for thermocouple locations 2 through 8 for vanes C and D. The results for thermocouple location 2 were similar to those obtained for location 1. The results for thermocouple location 3 showed no appreciable difference in cooling effectiveness between vanes C and D. The results for thermocouple locations 4 to 8 indicate that film cooling was beneficial at these locations.

A comparison of the trailing edge cooling effectiveness was also made for set 3 vanes. These vanes had considerably different trailing edge coolant side geometries than did the vanes in set 1 or set 2. Because of the cooling effectiveness results obtained in the previous vane sets, it was thought that an attempt to compare the cooling effectiveness of set 3 vanes would be of interest. Because the trailing edge geometry of vane E was so much different than that of vane B, the analytical modifications applied to vane B cooling effectiveness might be more questionable than the corrections made for the relatively minor geometry differences between vanes of sets 1 and 2. The trailing edge passage width of vane B was 0.076 centimeter (0.030 in.) compared to a width of 0.086 centimeter (0.034 in.) for vane E. The trailing edge passage of vane B contained rectangular spacers that were continuous in the chordwise direction, whereas the trailing edge coolant passage of vane E contained pin-fins that were either oblong or circular in

cross section. The increase in h_c due to the presence of pin-fins in vane E was calculated by using equation (D-2) in reference 10 which was used to modify the cooling effectiveness of vane B. Figure 4 compares the cooling effectiveness of vanes B and E at thermocouple location 1. The solid curves again represent results obtained using the correlation constants of table II(a) in equation (1a). The solid curves indicate that the cooling effectiveness of vane E was significantly higher than that of vane B for values of \dot{w}_c/\dot{w}_g greater than 0.015. The dashed curve in figure 4 shows the cooling effectiveness of vane B when it was analytically modified to account for the differences in trailing edge geometry between vanes B and E. Comparison of vane E results with that of the modified vane B results (dashed curve) now indicate that, at thermocouple location 1, vane B (non-film-cooled) is cooled more effectively than vane E (film-cooled). This result is consistent with the results observed for the set 1 and set 2 vanes discussed perviously.

Although not shown, a comparison was also made at thermocouple location 2. The results for thermocouple location 2 indicate that the cooling effectiveness of vane E (film cooled) was higher than the cooling effectiveness of vane B (non-film-cooled) after the cooling effectiveness of vane B was modified to account for the differences in the trailing edge geometries. This might be expected because thermocouple 2 for vane E was located immediately downstream of the film-cooling slot.

The results presented herein indicate that, under certain circumstances, film cooling results in higher vane wall temperatures at some locations downstream of the film injection region. It was not within the scope of the present investigation to determine the cause and conditions under which these adverse temperature conditions would result. It can be postulated, however, that if the gas-side boundary layer, without film cooling, was laminar or transitional, the injection of the film-cooling air may cause the boundary layer to become transitional or turbulent, respectively. This boundary layer transition would be reflected in an increased gas-side heat transfer coefficient. Thus it is possible that film cooling may be detrimental at locations where the benefits of film cooling are more than off-set by the increase in the gas-side heat transfer coefficient.

Cooling effectiveness (cascade). - A comparison was also made of the cooling effectiveness of set 1 and set 3 vanes tested in a four-vane cascade. The correlation constants are listed in table II(b). The results of these comparisons were similar to the results discussed previously. That is, film-cooling air ejection from the suction surface of a turbine vane results in higher wall temperatures at some downstream locations.

Pressure Surface

The local cooling-effectiveness values for the pressure surface were obtained for vanes A, B, C, and D using the same procedure discussed in the section ANALYSIS METHODS. Equation (1a) constants and exponents for each thermocouple location shown

in figure 1 are given in table III for engine and cascade tests. Figure 5 compares the cooling effectiveness of set 1 vanes at thermocouple location 7. The internal coolant passage geometries of vanes A and B are the same at this thermocouple location. Therefore, no analytical modification of the cooling effectiveness is required. Figure 5 shows that the cooling effectiveness of vane A (film-cooled) is higher than that of vane B (non-film-cooled) for values of \dot{w}_c/\dot{w}_g greater than 0.01.

Cooling effectiveness comparisons were made for other thermocouple locations for vane sets 1 and 2, with analytical modifications where necessary to account for internal geometrical differences. The results indicate that the local cooling effectiveness values downstream of the film coolant ejection point were always higher for the film-cooled vanes than for similar locations on the non-film-cooled vanes. That is, the downstream temperatures were lower for the film-cooled vanes. From the standpoint of the cooling effectiveness on the pressure surface it appears that, for the vanes considered herein, film cooling had no adverse effect on wall temperatures downstream of the point of film-cooling air ejection.

CONCLUDING REMARKS

The data presented herein indicate that, under certain conditions, film cooling is detrimental to effective cooling of turbine vanes. The experimental data examined in this report show that, for the test conditions specified, film cooling was detrimental on the suction-surface trailing edge region. It should be emphasized that the results obtained here were based on existing experimental data, and that the data had to be adjusted to account for the geometry differences that existed between the vanes being compared. The type of analytical adjustments that were made for the geometry differences between the film-cooled and the non-film-cooled vanes used in the comparisons are believed to be logical and reasonable. However, the analyses did not account for every uncertainty among the vanes, and several simplifying assumptions were made. Some of the assumptions and uncertainties associated with the analyses are:

- (1) The assumption of one-dimensional heat transfer
- (2) The uncertainty in the local coolant flow distribution within the vanes being compared
- (3) The uncertainty in the internal geometry between the vanes being compared
- (4) The uncertainty in the test conditions that may have existed when the experimental data were obtained for each of the vane configurations

Because of these factors, and the analytical adjustments that had to be made to the experimental correlations to account for geometrical differences, the results of the present investigation should be considered only as qualitative rather than quantitative.

SUMMARY OF RESULTS

The cooling effectiveness of three film-cooled vanes, each with a different internal cooling configuration, was investigated. Two of the vanes had film-cooling air ejected just aft of the leading edge. Film-cooling air was ejected through holes in one vane and through short slots in the other vane. The third vane had film-cooling air ejected through slots near the trailing edge. The cooling effectiveness of two film-cooled vanes was compared to the cooling effectiveness of two non-film-cooled vanes where the comparisons were made between vanes of similar internal cooling geometry. The cooling effectiveness of the third film-cooled vane was compared to one of the non-film-cooled vanes which had significantly different internal cooling geometry. The cooling effectiveness of the non-film-cooled vanes was analytically adjusted to account for the geometrical differences between the vanes being compared.

These comparisons showed that:

1. The effectiveness of film cooling was, from a qualitative viewpoint, detrimental on the suction-surface trailing edge region of the vanes tested. This phenomenon was shown to exist on the three film-cooled vanes tested in a turbojet engine and the two film-cooled vanes tested in a four-vane cascade. It was postulated that the addition of film cooling caused a laminar or transitional boundary layer to become transitional or turbulent, and be reflected in an increased gas side heat transfer coefficient.

2. Film cooling was, from a qualitative viewpoint, effective on the pressure surface of the vanes tested.

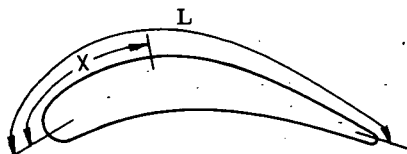
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TABLE I. - DIMENSIONLESS THERMOCOUPLE
LOCATIONS ON VANES A TO E



Thermocouple position number	Dimensionless surface location, x/L^a				
	Vane configuration				
	A	B	C	D	E
1	0.8808	0.8808	0.9158	0.9158	0.8563
2	.7955	.7955	.7899	.7899	.7420
3	.5292	.5292	.6711	.6711	-----
4	.4065	.4065	.6047	.6047	-----
5	.4208	.4208	.5540	.5540	-----
6	0.5406	0.5406	0.4963	0.4963	-----
7	.7476	.7476	.4345	.4345	-----
8	.8456	.8456	.3810	.3810	-----
9	-----	-----	.4045	.4045	-----
10	-----	-----	.4555	.4555	-----
11	-----	-----	0.5068	0.5068	-----
12	-----	-----	.6029	.6029	-----
13	-----	-----	.7818	.7818	-----

^a $L = 7.26$ cm (2.86 in.) on suction surface, and 6.51 cm (2.56 in.) on pressure surface; x is measured from the leading edge.

TABLE II. - CORRELATION CONSTANTS USED IN OBTAINING COOLING EFFECTIVENESS

$$\phi_x = \left[1 + \alpha \left(\frac{\dot{w}_c}{\dot{w}_g} \right)^\beta \right]^{-1} \quad \text{FOR THERMOCOUPLES LOCATED ON SUCTION SURFACE OF VANES}$$

(a) Correlation constants determined from engine test data

Thermocouple position number ^a	Vane A ^b		Vane B ^b		Vane C ^c		Vane D ^c		Vane E ^b	
	α	β	α	β	α	β	α	β	α	β
1	0.1812	-0.54	0.1403	-0.56	0.1019	-0.70	0.1033	-0.72	0.0580	-0.76
2	.1429	-.44	.0476	-.65	.0859	-.65	.1310	-.65	.0464	-.73
3	.0459	-.58	.0561	-.55	.0742	-.70	.0768	-.69		
4	.0321	-.80	.0731	-.51	.0529	-.71	.0717	-.67		
5					0.0477	-0.76	0.0659	-0.71		
6					.0336	-.81	.0858	-.65		
7					.0368	-.79	.0841	-.65		
8					.0539	-.73	.1061	-.65		

(b) Correlation constants determined from cascade test data

1	0.1335	-0.603	0.1325	-0.578					0.0344	-0.893
2	.0701	-.628	.0735	-.569					.0211	-.940
3	.0455	-.609	.0734	-.525						
4	.1272	-.505	.0874	-.445						

^aThermocouple position numbers refer to specific thermocouple positions on the vane suction surface given in table I and schematically shown in figure 1.

^bEngine test data from ref. 1.

^cEngine test data from ref. 5.

TABLE III. - CORRELATION CONSTANTS USED IN OBTAINING COOLING

$$\text{EFFECTIVENESS } \varphi_x = \left[1 + \alpha \left(\frac{\dot{w}_c}{\dot{w}_g} \right)^\beta \right]^{-1} \text{ FOR THERMOCOUPLES}$$

LOCATED ON PRESSURE SURFACE OF VANES

(a) Correlation constants determined from engine test data

Thermocouple position number ^a	Vane A ^b		Vane B ^b		Vane C ^c		Vane D ^c	
	α	β	α	β	α	β	α	β
5	0.0009	-1.65	0.0865	-0.57				
6	.0094	-.97	.0675	-.59				
7	.0169	-.94	.0784	-.61				
8	.0451	-.88	.1934	-.53				
9					0.0043	-1.43	0.165	-0.52
10					.0032	-1.51	.072	-.76
11					.0074	-1.29	-----	-----
12					.020	-1.06	.099	-.66
13					.023	-1.14	.154	-.71

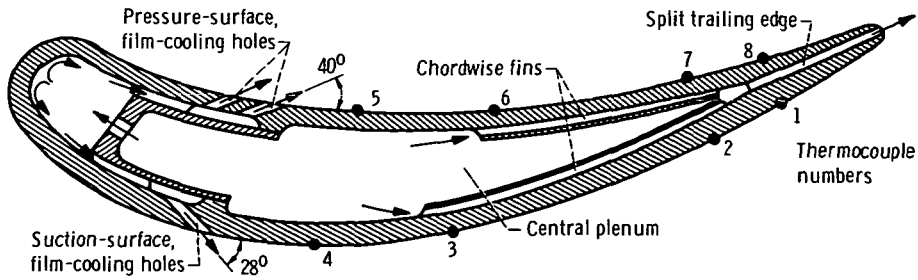
(b) Correlation constants determined from cascade test data

5	0.00226	-1.418	0.0637	-0.526				
6	.00378	-1.420	.0640	-.573				
7	.0181	-1.052	.0733	-.614				
8	.0212	-1.115	.128	-.635				

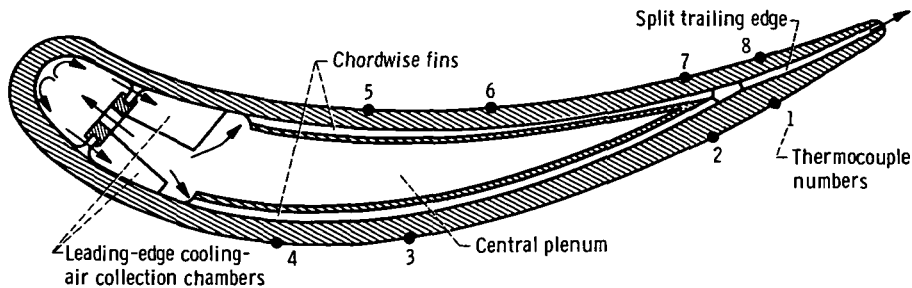
^aThermocouple position numbers refer to specific thermocouple positions on the vane pressure surface given in table I and schematically shown in figure 1.

^bEngine test data from ref. 1.

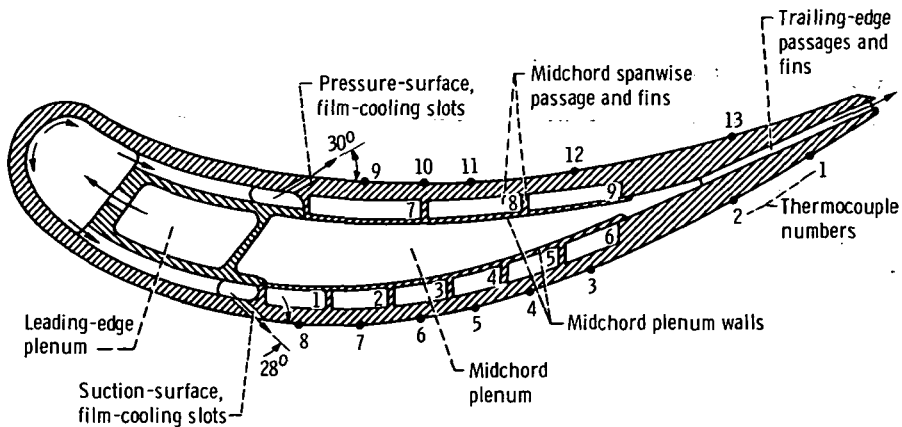
^cEngine test data from ref. 5.



(a) Vane A. Impingement cooled leading edge; convection and film cooled pressure and suction surfaces; chordwise fins in midchord region.

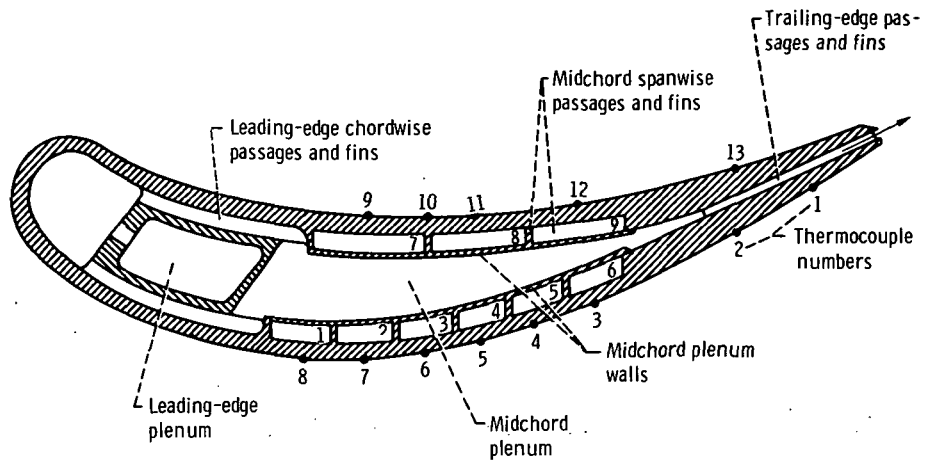


(b) Vane B. Impingement cooled leading edge; convection cooled pressure and suction surfaces; chordwise fins in midchord region.

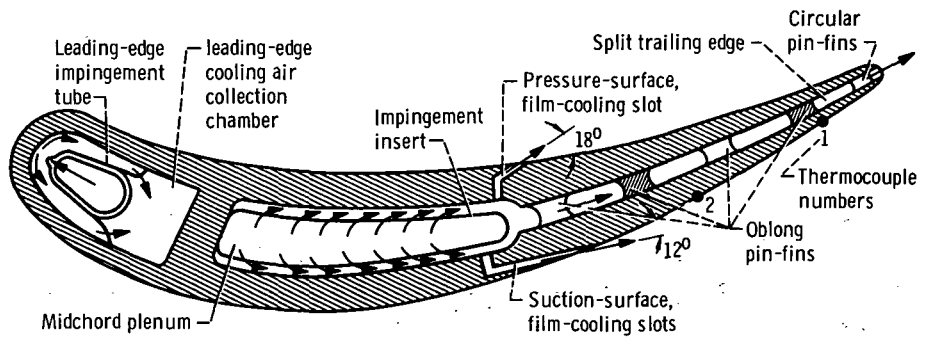


(c) Vane C. Impingement cooled leading edge; convection and film cooled pressure and suction surfaces; spanwise fins in midchord region.

Figure 1. - Typical cross sections (not to scale) showing cooling schemes and midspan thermocouple locations of the five test vanes.



(d) Vane D. Impingement cooled leading edge, spanwise fins, and split trailing edge.



(e) Vane E. Impingement cooled and chordwise finned leading edge, impingement cooled pressure and suction surfaces, and convection and film cooled trailing edge.

Figure 1. - Concluded.

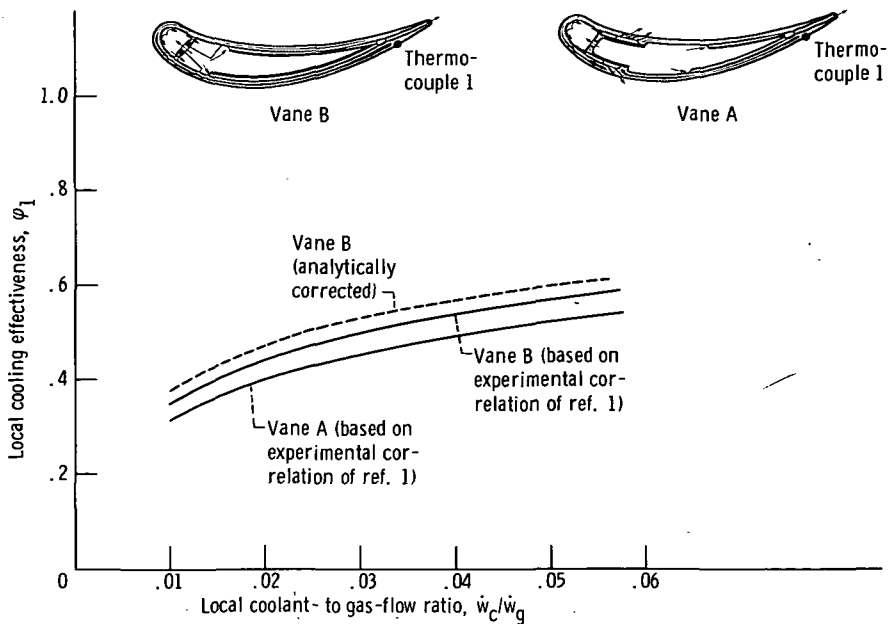


Figure 2. - Local cooling effectiveness for vanes A and B at thermocouple position 1 based on engine data.

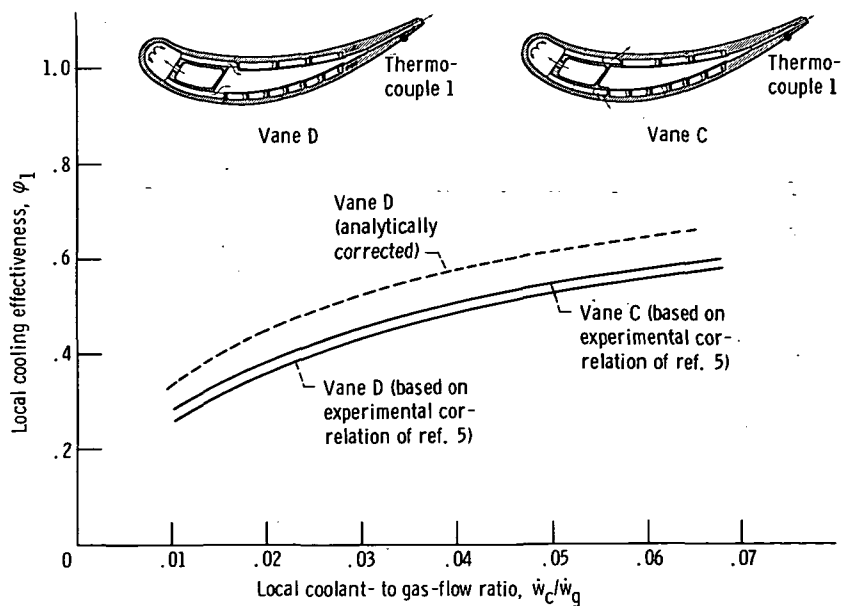


Figure 3. - Local cooling effectiveness for vanes C and D at thermocouple position 1 based on engine data.

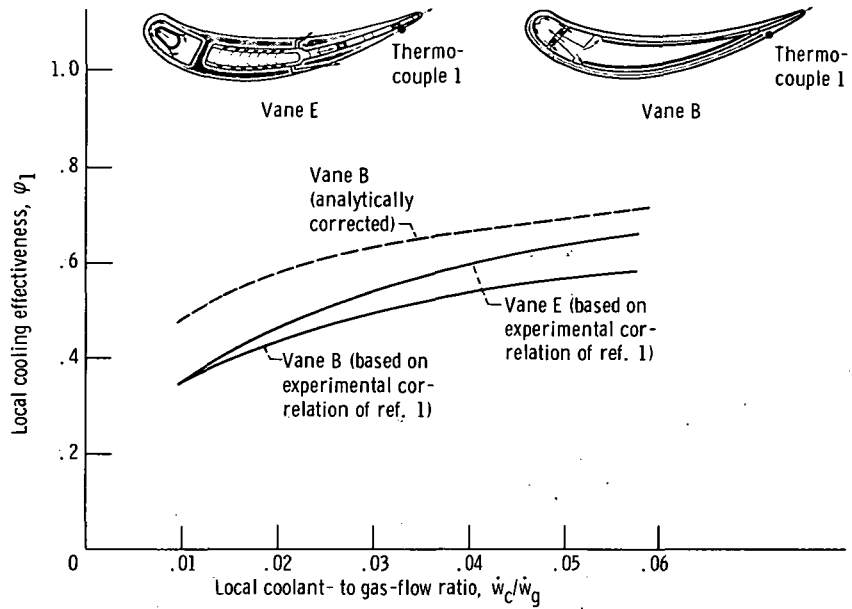


Figure 4. - Local cooling effectiveness for vanes B and E at thermocouple location 1 based on engine data.

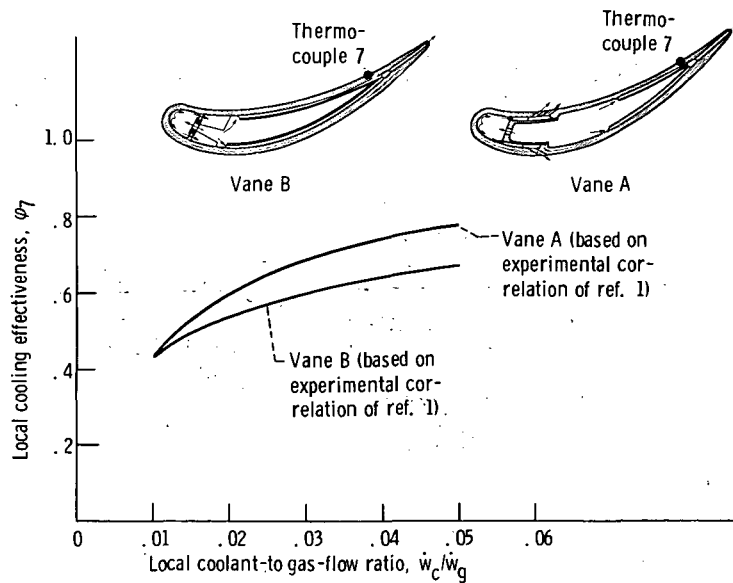


Figure 5. - Local cooling effectiveness for Vanes A and B at thermocouple location 7 based on engine data. (Thermocouple location 7 on pressure surface did not require analytical modification of geometry differences).



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