

1. Report No. NASA TM X-3177		2. Government Accession No.		3. Recipient's Catalog No.	
4. Title and Subtitle AERODYNAMIC PERFORMANCE OF A FULLY FILM COOLED CORE TURBINE VANE TESTED WITH COLD AIR IN A TWO-DIMENSIONAL CASCADE				5. Report Date January 1975	
				6. Performing Organization Code	
7. Author(s) Roy G. Stabe and John F. Kline				8. Performing Organization Report No. E-8131	
9. Performing Organization Name and Address Lewis Research Center National Aeronautics and Space Administration Cleveland, Ohio 44135				10. Work Unit No. 505-04	
				11. Contract or Grant No.	
12. Sponsoring Agency Name and Address National Aeronautics and Space Administration Washington, D. C. 20546				13. Type of Report and Period Covered Technical Memorandum	
				14. Sponsoring Agency Code	
15. Supplementary Notes					
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17. Key Words (Suggested by Author(s)) Turbine Aerodynamic performance Cooled vanes Cascades			18. Distribution Statement Unclassified - unlimited STAR category 02 (rev.)		
19. Security Classif. (of this report) Unclassified		20. Security Classif. (of this page) Unclassified		21. No. of Pages 19	22. Price* \$3.25

* For sale by the National Technical Information Service, Springfield, Virginia 22151

AERODYNAMIC PERFORMANCE OF A FULLY FILM COOLED CORE TURBINE VANE TESTED WITH COLD AIR IN A TWO-DIMENSIONAL CASCADE

by Roy G. Stabe and John F. Kline

Lewis Research Center

SUMMARY

The aerodynamic performance of a fully film cooled core turbine vane was investigated experimentally in a two-dimensional cascade of 10 vanes. Three of these vanes were cooled; the others were solid (uncooled) vanes with the same profile as the cooled vanes. Cold air was used for both the primary and coolant flows. The cascade tests covered a range of exit velocity ratios and coolant flow rates. The principal measurements were cross-channel surveys of the exit flow conditions. Similar tests were performed with a single cooled vane in the 10-vane cascade. The results of this test were compared with the results obtained with three cooled vanes to determine the feasibility of using only one cooled vane. The performance of the cooled vane is also compared to the performance of an uncooled vane with the same profile.

At comparable conditions, that is, an ideal exit critical velocity ratio of 0.8 and coolant supply pressure equal to primary inlet total pressure, the thermodynamic loss of the cooled vane was 0.073. This is about three times the loss of the uncooled vane, 0.023. The high loss of the cooled vane was attributed to coolant flow maldistribution and pressure losses in the vane internal cooling passages. These pressure losses reduced the coolant energy available to the mixing process of the primary and coolant flows, which resulted in a lower aftermix velocity.

The injection of coolant flow into the primary flow stream also reduced the primary flow. The primary flow decreased at a rate of about 1 percent per percent coolant flow. Consequently, the combined coolant and primary flows were very nearly constant.

The pattern of the exit survey results from the three-cooled-vane configuration was repeated for the adjacent vanes. But with a single cooled vane the flow angle pattern was not repeated. This resulted in a maldistribution of the flow and some differences in performance. These differences were small, however; and it was concluded that the use of a single cooled vane in a 10-vane cascade may be satisfactory where loss comparisons of vanes with essentially similar cooling designs are all that is required.

INTRODUCTION

The use of higher turbine inlet temperatures in advanced aircraft engines has necessitated increasingly comprehensive and sophisticated turbine cooling schemes. The general method of cooling the turbine is to direct relatively cool air bled from the compressor through the turbine blading. This cooling air is then ejected into the primary gas stream. Such ejection can adversely affect the aerodynamic performance of the turbine. In addition, the various methods of ejecting the cooling air have significantly different effects on turbine performance.

A comprehensive research program is in progress at the NASA Lewis Research Center to investigate the effects that the different methods of cooling have on turbine and turbine blade performance. Several experimental investigations of these effects on turbine performance have been performed and the results reported. Reference 1 summarizes much of this work. An analytical study of the effects of several cooling variables on turbine performance is presented in reference 2. The effects of cooling methods on turbine vane performance have also been investigated in cascades. The aerodynamic performance of two types of cooled turbine vanes tested at gas- to coolant-temperature ratio to 2.75 in an annular-sector cascade is presented in reference 3. Three-dimensional effects for vanes similar to those of reference 3 were then investigated in a full-annular cascade (ref. 4). References 5 to 7 investigated the effects of suction-surface, pressure-surface, and full-film coolant ejection on the aerodynamic performance of turbine vanes in two-dimensional cascades.

The subject investigation is a continuation of the effort to determine the effects of different types of cooling on turbine vane performance. The test vane was a fully film cooled core turbine vane. This test vane differed from those used for references 5 to 7 in that both the aerodynamic and cooling designs were representative of an advanced high-temperature-core engine application. The cooling design of the test vane included an internal liner. The function of this liner was to distribute the cooling flow to the various regions of the vane and also to provide impingement cooling on the inside surface of the vane shell. It was expected that this liner would cause some performance penalty.

Testing for this investigation was conducted in a simple two-dimensional cascade tunnel of 10 vanes. A two-dimensional cascade tunnel provides a convenient and inexpensive method of determining the relative performance of different types of cooled blading. For these tests, three of the vanes near the center of the cascade were cooled test vanes. The other vanes were solid (uncooled) vanes of the same size and profile as the cooled vanes. We believe that the use of three cooled vanes in a 10-vane cascade adequately simulates the flow field that would be obtained with 10 cooled vanes. This is the same arrangement that was used for references 5 to 7 with satisfactory results. The use of only three cooled vanes resulted in a considerable savings in the cost of test vanes.

However, in investigations where a number of different cooled vanes are tested and their performance compared, further savings could be realized if only one cooled vane of each type were necessary. Accordingly, a cascade configuration with a single cooled test vane was also tested to determine if the results obtained were comparable to the results from tests with three cooled vanes. A third cascade configuration of 10 solid (uncooled) vanes of the same size and profile as the cooled vanes was also tested. These tests were made to provide a basis for comparing the performance of the cooled vanes.

The cascade tests covered a range of pressure ratios corresponding to ideal exit critical velocity ratios from about 0.6 to 0.95 at a coolant- to primary-gas-pressure ratio of 1.0. At approximately design exit critical velocity ratio the coolant- to primary-gas-pressure ratio was varied from about 0.85 to 1.5. This resulted in a range of coolant flow to about 7.5 percent of the primary flow. Atmospheric air was used as the primary gas. The temperatures of the primary and coolant air were very nearly the same. The principal measurements were cross-channel surveys of exit total pressure, static pressure, and flow angle. The results of the experimental investigation include exit survey results and overall performance in terms of flow angle, weight flow, and kinetic energy loss. The performances of the cooled test vane for the three- and single-cooled-vane cascade configurations are compared. Data for a solid (uncooled) vane with the same profile as the cooled vane are also included to illustrate the effect of cooling on the performance of the test vanes.

SYMBOLS

a	distance along axial chord from leading edge, cm
c_a	vane axial chord, cm
$\bar{e}_{3,p}$	primary kinetic energy loss coefficient for cooled vanes, $1 - \left[\frac{W_3 V_3^2}{W_p (V_{id,3})_p^2} \right]$
$\bar{e}_{3,t}$	thermodynamic kinetic energy loss coefficient for cooled vanes, $1 - \left\{ \frac{W_3 V_3^2}{\left[W_p (V_{id,3})_p^2 + W_c (V_{id,3})_c^2 \right]} \right\}$
$\bar{e}_{3,uc}$	kinetic energy loss coefficient for uncooled vanes, $1 - (V_3/V_{id,3})^2$
m	mass flow, kg/(sec)(cm ²)
p	absolute pressure, N/cm ²
s	vane spacing, cm
t	tangential distance from vane trailing edge, cm
V	velocity, m/sec

W	flow rate per unit of vane span, kg/(sec)(cm)
y	coolant flow, percent of primary flow
α	exit flow angle, deg from axial
δ	ratio of inlet total pressure to U. S. standard sea-level atmospheric pressure, $p_1'/10.132 \text{ N/cm}^2$
$\sqrt{\theta_{cr}}$	ratio of inlet critical velocity to critical velocity of U. S. standard sea-level air, $V_{cr, 1}/310.6 \text{ m/sec}$

Subscripts:

c	coolant flow
cr	flow conditions at Mach 1
id	ideal or isentropic process
p	primary flow
s	vane surface
1	station at vane inlet
2	station at vane exit survey plane
3	station downstream of vane exit, where flow conditions are assumed to be uniform

Superscript:

'	total state conditions
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APPARATUS AND PROCEDURE

Test Vanes

The test vane is shown in figure 1. The vane had 44 rows of cooling holes. The spanwise spacing of the holes was 2.54 millimeters. Alternate rows were offset 1.27 millimeters to provide better film coverage. The axis of the holes was parallel to the cascade end walls and inclined in the direction of flow. The angle between the axis of the holes and the vane surface was about 35° for all cooling holes except those at the leading edge. There were no holes at the trailing edge of the vane. The diameters of the cooling holes ranged from 0.33 to 0.84 millimeter. The area-averaged cooling hole diameter was about 0.5 millimeter. The test vane also had an internal liner. The function of this liner was to distribute the coolant to the various regions of the vane and also to provide impingement cooling to the inside surface of the vane shell. The profiles and

relative positions of the test vanes in the cascade are shown in figure 2. The vane coordinates, other significant dimensions, and the velocity diagrams are also shown in this figure.

Cascade Tunnel

The vanes were tested in the simple two-dimensional cascade tunnel shown in figure 3. This cascade tunnel has 10 vanes with a span of 10.16 centimeters. However, only three of the vanes near the center of the cascade were cooled. Two other configurations were also tested - one with a single cooled vane and another with all solid (uncooled) vanes.

In operation, room air was drawn through the cascade tunnel, the vanes, and the exhaust control valve into the laboratory exhaust system. Cooling air for the film-cooled vanes was supplied to both ends of the vanes from manifolds on the tunnel end walls. An ASME flat-plate orifice was used to measure the cooling flow.

All three configurations were tested over a range of inlet-total- to exit-static- pressure ratios corresponding to ideal exit critical velocity ratios of about 0.6 to 0.95. For these tests with the two cooled-vane configurations the coolant supply manifold pressures were maintained at a value equal to the cascade inlet total pressure p_1' . The coolant flow was about 3.4 percent of the primary flow. At approximately design exit critical velocity ratio the cooled vanes were also tested over a range of coolant- to primary-gas-pressure ratios p_c/p_1' from about 0.85 to 1.5. This resulted in a range of coolant flows to about 7.5 percent of the primary flow. The total temperatures of the primary and cooling air were very nearly equal for all tests with the cooled vanes.

Instrumentation

In the solid (uncooled) vane configuration the two vanes that formed the center passage of the 10-vane cascade were instrumented at midspan with static pressure taps. The location of these taps is shown in figure 2. Vane surface pressure taps were not used on the cooled vanes. The cascade tunnel also had wall static pressure taps in the exit section. These taps were used to set the exit static pressure. The vane surface and wall static pressures were measured with mercury-filled manometers. The pressure data were recorded by photographing the manometer board.

The vane-to-vane variations of exit total pressure, static pressure, and flow angle were surveyed simultaneously with the rake shown in figure 4. The total pressure was measured with a simple square-ended probe. The static pressure was measured with a wedge probe that had an included angle of 15° . The angle probe was a two-tube type with

the tube ends cut at 45° . The probe measured a differential pressure which was related to flow angle. Strain-gage transducers were used to measure these pressures.

The survey rake was installed in the cascade with the rake stem parallel to the vane trailing edges. The sensing elements of the rake were aligned with the design flow angle and fixed. This angle was not changed during the surveys. The sensing elements were located at the midspan region of the vane with the element tips at the survey plane, station 2 in figure 2. Station 2 was 16.6 percent of the vane axial chord axially downstream of the vane trailing edges. The rake was traversed tangentially over a distance of about two vane spaces behind the vane bounding the center channel of the 10-vane cascade. The traverse speed was about 2.5 centimeters per minute. An actuator-driven potentiometer was used to provide a signal proportional to rake position. The output signals of the three pressure transducers and the rake position potentiometer were recorded on magnetic tape. The recording rate was 20 words per second.

Data Reduction

Vane surface static pressures were taken from photographs of the manometer board. These data were used to calculate the blade surface velocity ratios. A computer was used to reduce the vane exit survey data recorded on magnetic tape. These flow angle and pressure data were used to calculate velocity, mass flow, and the tangential and axial components of momentum as a function of rake position. These quantities were then integrated numerically over a distance equal to one vane space to obtain overall values at the plane of the rake, station 2. The continuity and conservation of momentum and energy relations were then used to calculate the flow angle, velocity, and pressures at a hypothetical location where the flow conditions were assumed to be uniform. This location was designated station 3. For these calculations a constant-area process and conservation of the tangential component of momentum were assumed between stations 2 and 3. The details of these calculations are given in reference 4.

RESULTS AND DISCUSSION

In this section the overall performance of the fully film cooled core turbine vane as determined with three cooled vanes in a two-dimensional cascade is presented for the range of exit velocity ratios and coolant flows investigated. These results are compared with the results of similar tests with a single cooled vane and also with the results of the solid (uncooled) vane tests.

Performance Comparison

Survey results. - Typical exit survey data taken at near-design conditions, that is, an ideal exit velocity ratio of 0.8 and coolant supply total temperature and pressure equal to primary inlet total conditions, are shown in figure 5 for the three vane configurations tested. For these conditions the coolant flow was about 3.4 percent of the primary flow for both cooled-vane configurations. The survey results are shown for one vane space. However, survey data were taken for two or more vane spaces to determine if the pattern of the vane-to-vane variation repeated for the adjacent vanes.

The total pressure and static pressure traces are shown in figure 5(a). The total pressure wakes for the cooled vanes were much larger than the wake for the solid (uncooled) vane. And the wake for the three-cooled-vane configuration was slightly larger than the wake for the single-cooled-vane configuration. A larger wake indicates a larger loss. There were also some small differences in the static pressure variation. These differences will result in corresponding differences in the vane-to-vane variation in velocity and mass flow for the three configurations. The static pressures at the end points of the survey were equal, and the pattern of the vane-to-vane variation of static pressure repeated fairly well for all three configurations.

The flow angle data are shown in figure 5(b). The flow angle for the uncooled vanes was generally higher than it was for the cooled vanes. That is, the uncooled vanes turned the flow more than the cooled vanes did. The pattern of the vane-to-vane variation of flow angle repeated for the adjacent vanes for both the uncooled-vane and three-cooled-vane configurations. This pattern did not repeat for the single-cooled-vane configuration. In this case the flow angle varied in a manner similar to that for the uncooled-vane configuration on the pressure-surface side of the wake and in a manner similar to that for the three-cooled-vane configuration on the suction-surface side of the wake.

The mass flow, or flow per unit area, which was calculated from the pressure and angle data is shown in figure 5(c). Consequently, the vane-to-vane variation in mass flow was repeated for both the uncooled-vane and three-cooled-vane configurations but not for the single-cooled-vane configuration. The flow angle has a greater influence on the mass flow in the free stream, and the velocity is more important in the wake region. This results in a larger mass flow in the free stream and a smaller mass flow in the wake for cooled vanes as compared with solid vanes. The mass flow deficit in the wake of the cooled vanes indicates that there was a fairly large amount of low momentum fluid in this region. The results of reference 3 further indicate that much of this low momentum fluid was unmixed coolant flow.

The repeating pattern of the survey results implies that the use of three cooled vanes in a 10-vane cascade should provide valid data. With a single cooled vane, however, the pattern of the angle data was not repeated for the adjacent vanes; this resulted

in a maldistribution of the flow. The differences between the single- and three-cooled-vane configurations are small, but this flow maldistribution did change the performance of the single cooled test vane. The differences in performance are reflected in the after-mix parameters which are presented in a later section.

Cascade tests using only one cooled vane may be satisfactory where loss comparisons for vanes with essentially similar cooling designs are all that is required. In this case the flow field would be similar for all test vanes and would probably not significantly affect the difference in loss due to cooling design. The use of only one cooled vane, however, may result in invalid data if vanes with radically different cooling designs are tested. Here the flow field would be different for each vane, and the performance of some vanes may be more sensitive to these differences than others.

Vane surface velocity distribution. - The two vanes forming the center channel of the 10-vane cascade were instrumented with static pressure taps. These taps were installed on the solid (uncooled) vanes only. The vane surface velocity distribution, which was calculated from the static pressure measurements at near-design operating conditions, is shown in figure 6. This is a typical velocity distribution for a core turbine vane with moderately high loading. The Zweifel loading coefficient for incompressible flow was 0.789. The loading coefficient for compressible flow, which was calculated by integrating the vane surface static pressure distribution, was 0.654. The surface velocity distributions for the cooled vane were probably similar to that shown for the uncooled vane.

Aftermix performance parameters. - The aftermix values of kinetic energy loss coefficient, flow angle, and total weight flow are shown in figure 7 as a function of the ideal exit critical velocity ratio for all configurations tested. The aftermix parameters were calculated from the exit survey measurements, as explained in the section Data Reduction. All data shown in figure 7 are for coolant supply total pressure and temperature equal to the primary inlet total conditions. The coolant flow was about 3.4 percent of the primary flow for both cooled-vane configurations over the range of exit velocity ratios investigated.

The kinetic energy loss coefficients are shown in figure 7(a). Both the primary and thermodynamic loss coefficients are shown for the cooled vanes. These terms are defined in the section SYMBOLS. Briefly, the primary loss coefficient is the ratio of the actual loss of the mixed flow to the ideal energy of the primary flow. The thermodynamic loss coefficient is the ratio of the actual loss of the mixed flow to the ideal energy of both the primary and coolant flows.

The loss for the three-cooled-vane configuration was slightly higher than the loss for the single-cooled-vane configuration. This difference was probably due to the differences in the exit survey measurements discussed previously. However, the variation of loss with exit velocity ratio was similar for both cooled-vane configurations. There was a trend of decreasing loss with increasing exit velocity. The variation in loss with exit

velocity for the uncooled vanes shows the loss increasing sharply at an ideal exit velocity ratio of about 0.9. This is an indication that separation was beginning to occur.

When the coolant supply total pressure and temperature are equal to the primary inlet total state conditions, as is the case for figure 7(a), the expression for the thermodynamic loss reduces to an identity with the expression for the loss of an uncooled vane. Thus, in figure 7(a), the thermodynamic loss can be compared directly with the loss for an uncooled vane. The thermodynamic loss for the cooled vanes at approximately design exit velocity ratio, 0.8, was 0.073. This is about three times the loss for the uncooled vanes, 0.023. The loss for the cooled vanes was also higher than the loss obtained from other film-cooled-vane tests (ref. 7). Some of the reasons for this are discussed in a following section.

The aftermix flow angle data are shown in figure 7(b). The highest flow angles were for the uncooled-vane-configuration. At approximately design exit velocity ratio, 0.8, this angle was close to the design value, 67° . As indicated by the exit survey results the cooled vanes did not turn the flow as well as the uncooled vanes. The flow angle for the three-cooled-vane configuration was about 1.5° less than design and almost 1° less than the flow angle for the single-cooled-vane configuration. The difference in flow angle for the various configurations decreased at the higher ideal exit critical velocity ratios.

The weight flow data are shown in figure 7(c). Weight flow is dependent on both loss and flow angle. The differences in flow between the three test configurations were due to the combination of loss and flow angle data shown in figures 7(a) and (b). At approximately design exit velocity ratio the flow for the three-cooled-vane configuration was about 1.5 percent higher than the flow for the uncooled-vane configuration. This was due primarily to the flow angle, which was closer to axial. The flow for the single-cooled-vane configuration was about 1 percent less than the flow for the uncooled-vane-configuration. And this was due primarily to the much higher loss.

Effect of Coolant Flow on Performance

The overall performance of the film-cooled vane over a range of coolant flow rates is presented in figure 8. These data were obtained with the three-cooled-vane configuration and are for an ideal exit critical velocity ratio of 0.8, which was near the design value. The coolant flow was varied by changing the coolant supply pressure. Consequently, the ideal energy of the coolant increased with coolant fraction because of the increasing coolant- to primary-pressure ratio. Data were taken for a range of coolant-supply- to primary-inlet-total-pressure ratios from 0.85 to 1.5 and also with the coolant supply shut off. The point where coolant supply and primary inlet total pressures were equal is indicated in figure 8. At this point the coolant flow was 3.4 percent of the

primary flow. The highest coolant flow was about 7.5 percent of the primary flow at a coolant- to primary -pressure ratio of 1.5.

Kinetic energy loss coefficient. - The primary and thermodynamic loss coefficients for the three-vane configuration are shown as a function of percent coolant flow in figure 8(a). The thermodynamic loss coefficient is a measure of the loss compared with the combined ideal energies of the primary and coolant flows. The primary loss coefficient is a measure of the loss per unit of primary flow only. With the coolant flow shut off the loss was 0.055 compared with a loss of 0.023 for the solid (uncooled) vane. This increased loss was caused by leakage of the primary flow through the vane internal cooling passages. This leakage occurs for all coolant- to primary-pressure ratios less than 1. The resulting loss is not associated with the normal flow process. For this reason the data shown in figure 8 are related to the performance of the uncooled vane.

The thermodynamic loss increased with increasing coolant flow. This increase was fairly rapid at coolant- to primary-pressure ratios greater than 1, where the ideal energy of the coolant was greater than the ideal energy of the primary flow. This rapid rise in thermodynamic loss indicates that the energy of the coolant was not used very effectively. The primary loss decreased gradually with increasing coolant flow but was higher than the loss for the uncooled vanes over most of the range of coolant flow. When the coolant supply and primary inlet total pressures were equal, the coolant flow was 3.4 percent of the primary flow. At this point the primary and thermodynamic loss coefficients were 0.042 and 0.073, respectively. As mentioned previously the losses for this cooled vane were considerably higher than the loss for the uncooled vane and also much higher than the losses obtained from other film-cooled-vane tests such as those of reference 7.

The reasons for these higher losses with the three-vane configuration are more apparent in figure 8(b), where the ratio of the aftermix- to ideal-exit velocity is plotted as a function of percent coolant flow. The aftermix velocity ratio was lower than it was for the uncooled vanes and decreased with percent coolant flow even though the ideal energy of the coolant flow was increasing. This occurred because pressure losses and maldistribution of the coolant flow in the vane internal cooling passages reduced the energy of the coolant available to the mixing process so that it was less than the energy of the primary flow. This low coolant energy detracts from the energy of the primary flow and may also interfere with the primary flow process, resulting in high loss. The vanes tested for reference 7 had no internal cooling liner and therefore no internal losses. The energy of the coolant at the point of injection into the primary flow was then higher than for the subject vanes, and the resulting aftermix losses were lower.

Reduction of primary flow. - When coolant is injected into the primary flow stream from vane surfaces, it usually reduces the primary flow. This reduction is effected through blockage - making part of the flow area unavailable to the primary flow - or through interference with the primary flow process and increased loss.

The reduction of primary flow, as a percentage of the flow with no coolant, is shown in figure 8(c) as a function of percent coolant flow. The total flow, primary plus coolant, is also shown in this figure. The primary flow decreases with percent coolant flow at a rate of almost 1 percent per percent coolant flow. Consequently, the total flow is very nearly constant. These results appear to be typical for film-cooled vanes.

SUMMARY OF RESULTS

The aerodynamic performance of a fully film cooled core turbine vane was investigated experimentally in a two-dimensional cascade of 10 vanes. Three of these vanes were cooled. The others were solid (uncooled) vanes. The cooled vanes were tested over a range of ideal exit critical velocity ratios and coolant flow rates. The vane performance obtained from these tests was compared with the results of similar tests with a single cooled vane and also with the performance of an uncooled vane of the same profile. The results of this investigation are summarized as follows:

1. The pattern of the exit survey results from the three-cooled-vane configuration was repeated for the adjacent vanes, which implies that the use of three cooled vanes in a 10-vane cascade should provide valid data. With a single cooled vane the flow angle variation was not repeated. This resulted in maldistribution of the flow and some differences in the aftermix performance parameters. However, these differences were small, and it was concluded that the use of a single cooled vane in a 10-vane cascade may be satisfactory where loss comparisons of vanes with essentially similar cooling designs are all that is required.

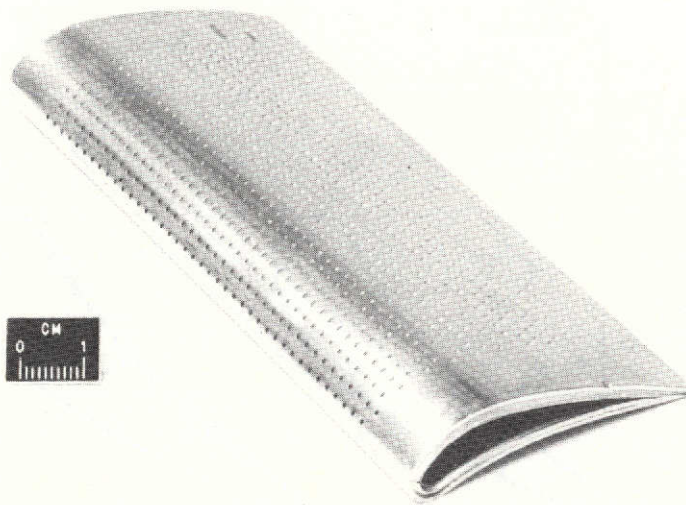
2. At approximately design conditions, that is, an ideal exit critical velocity ratio of 0.8 and coolant supply pressure equal to the primary inlet total pressure, the thermodynamic loss coefficient was 0.073 for the three-cooled-vane configuration. The loss coefficient for an uncooled vane of the same profile, which for the conditions stated is comparable to the thermodynamic loss coefficient, was only 0.023. This high loss for the cooled vanes was attributed to pressure losses and coolant flow maldistribution in the vane internal cooling passages. These pressure losses reduced the energy of the coolant available to the mixing process of the primary and coolant flows and therefore reduced the aftermix velocity.

3. Injection of coolant flow into the primary flow stream reduced the primary flow at a rate of about 1 percent per percent coolant flow. Consequently, the total flow was very nearly constant. These results appear to be typical of film-cooled vanes.

Lewis Research Center,
National Aeronautics and Space Administration,
Cleveland, Ohio, November 13, 1974,
505-04.

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C-73-2622

Figure 1. - Film-cooled test vane.

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Vane coordinates		
X	Y_L	Y_U
0	0.508	0.508
0.127	-----	.851
.254	-----	1.003
.381	-----	1.120
.508	-----	1.212
.635	-----	1.285
.762	.061	1.341
.889	.117	1.389
1.016	.163	1.425
1.143	.201	1.448
1.270	.236	1.463
1.397	.267	1.471
1.527	.292	1.476
1.778	.328	1.461
2.032	.358	1.427
2.286	.376	1.377
2.540	.384	1.321
2.794	.381	1.255
3.048	.368	1.191
3.302	.353	1.110
3.556	.328	1.026
3.810	.297	.942
4.064	.262	.848
4.318	.221	.747
4.572	.178	.635
4.826	.130	.521
5.080	.084	.399
5.334	.025	.267
5.552	.089	.089

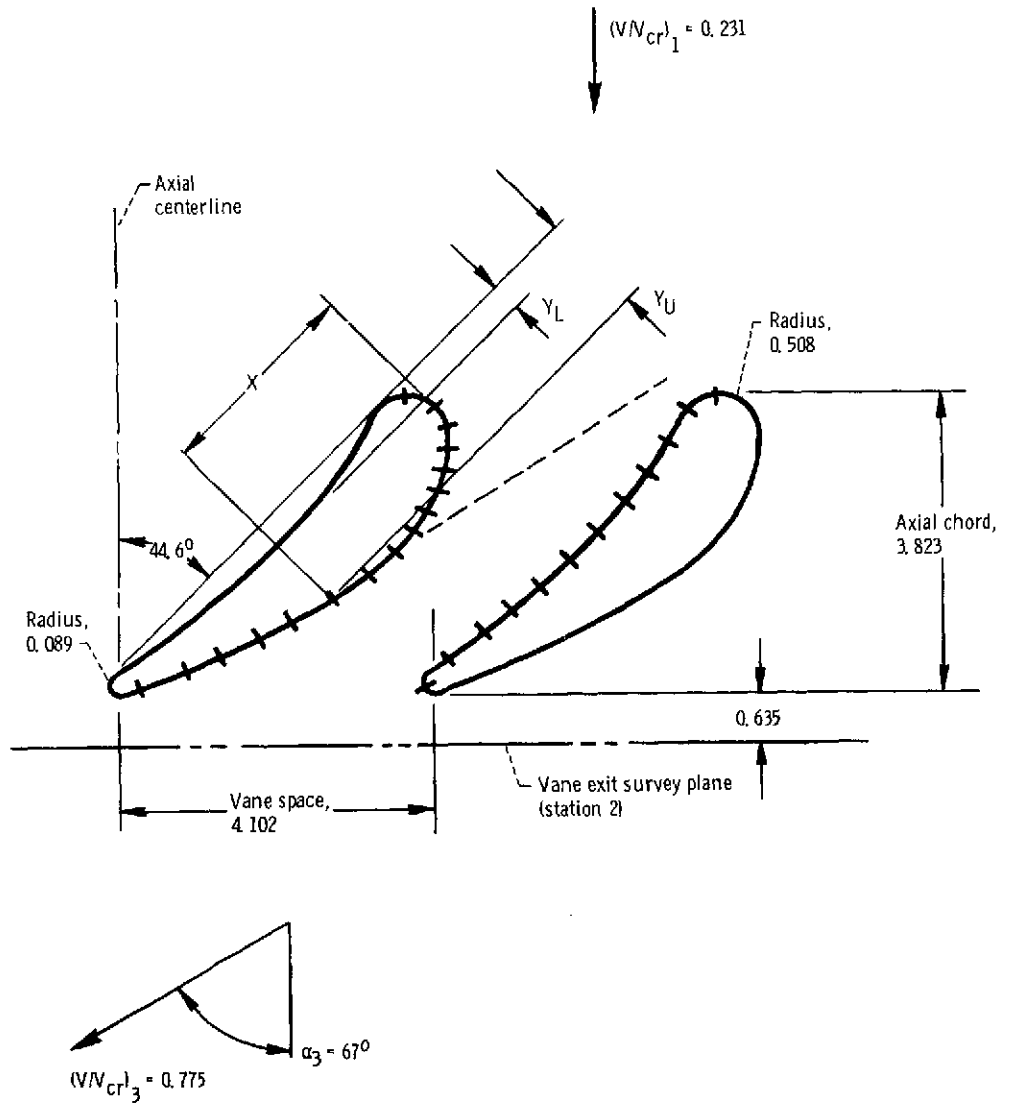


Figure 2 - Stator vane geometry. (Location of static taps indicated by tick marks. All dimensions in cm.)

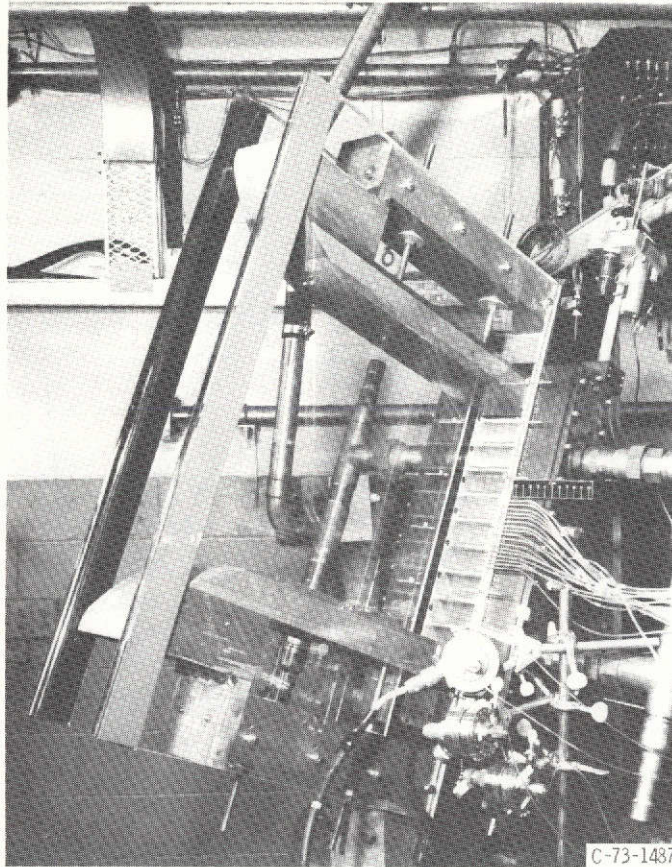


Figure 3. - Ten-vane cascade tunnel.

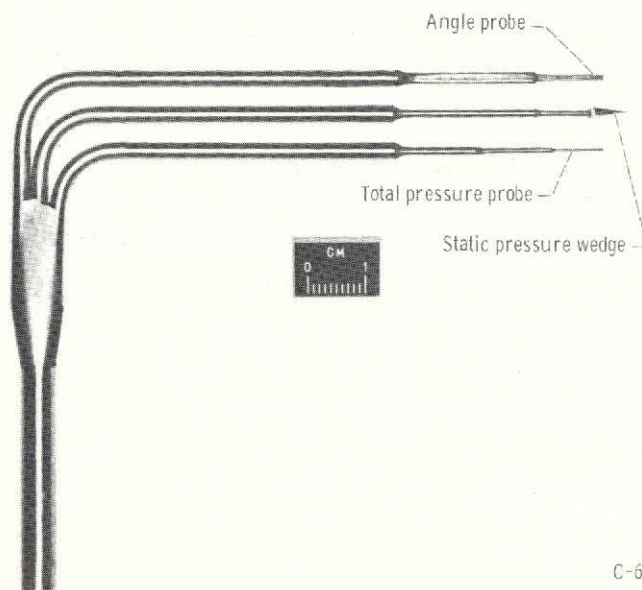


Figure 4. - Combination exit survey probe.

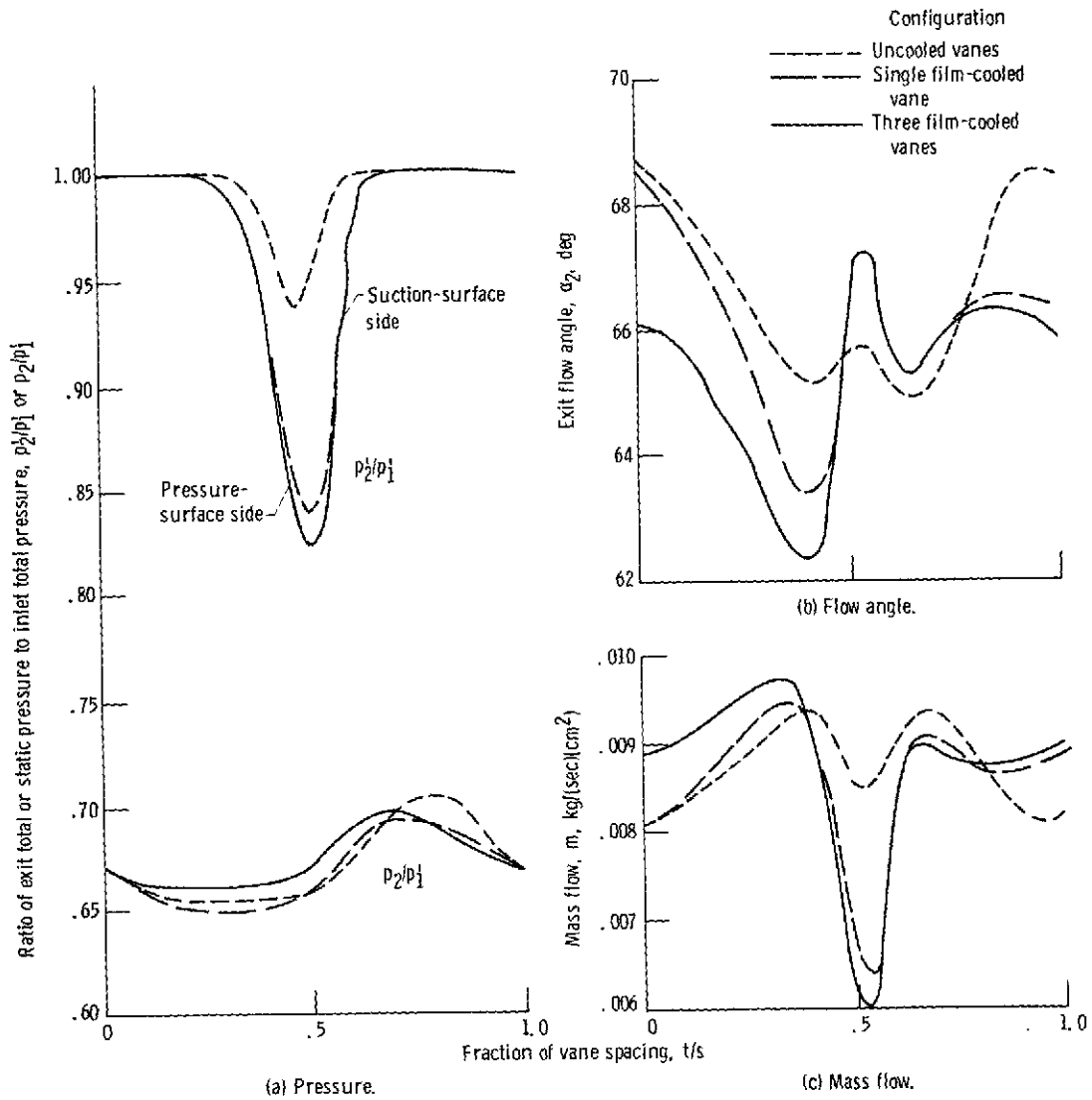


Figure 5. - Vane-to-vane variation in exit flow conditions at station 2 for three vane configurations at an ideal exit critical velocity ratio $(V/V_{cr})_{id,3}$ of 0.8.

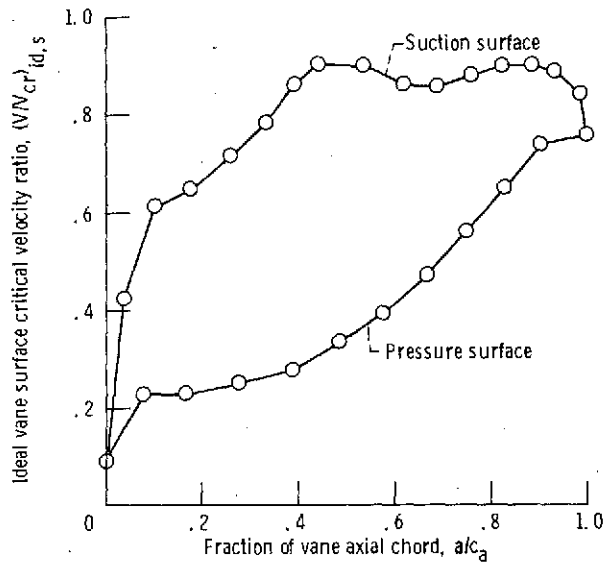


Figure 6. - Variation of ideal vane surface critical velocity ratio for solid (uncooled) vanes at ideal exit critical velocity ratio $(VN_{cr})_{id,3}$ of 0.8.

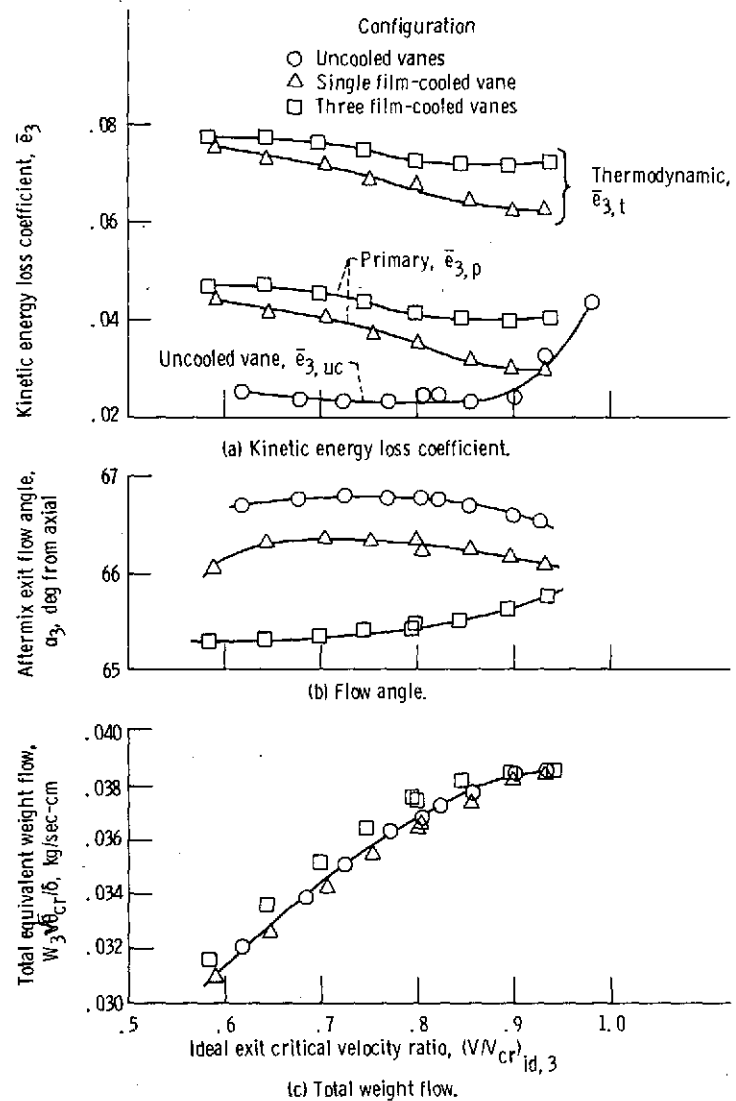


Figure 7. - Overall performance as function of ideal exit critical velocity ratio for three vane configurations with coolant supply pressure equal to cascade inlet total pressure.

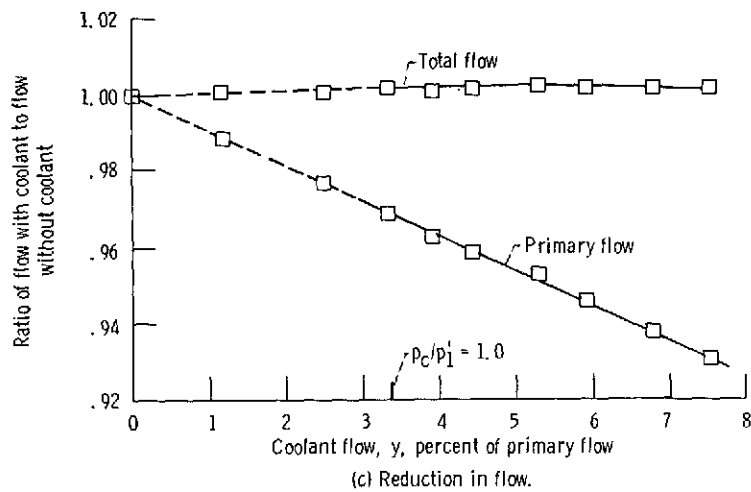
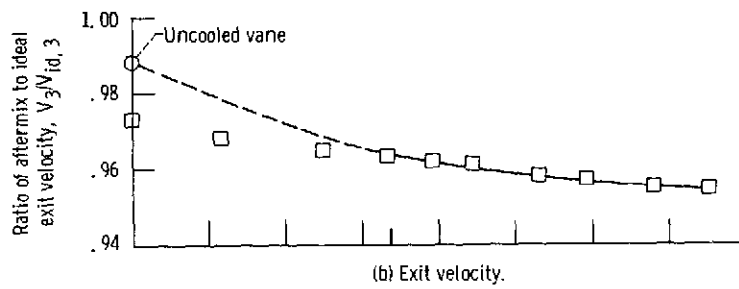
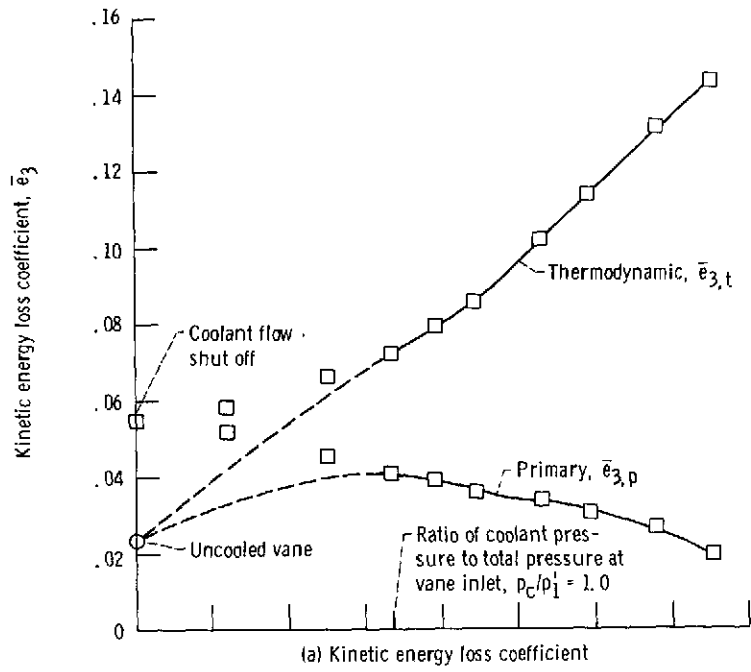


Figure 8. - Overall performance as a function of percent coolant flow for three-cooled-vane configuration at ideal exit critical velocity ratio $(V/V_{cr})_{id,3}$ of 0.8.