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AERODYNAMIC PERFORMANCE OF A CORE-ENGINE TURBINE STATOR VANE TESTED IN A TWO-DIMENSIONAL CASCADE OF 10 VANES AND IN A SINGLE-VANE TUNNEL

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16. Abstract

A turbine stator vane was tested in a two-dimensional cascade of 10 vanes and in a single-vane tunnel. The single-vane tunnel was a cold air version of a tunnel which will be used for high temperature heat transfer testing of cooled turbine vanes. The purpose of the investigation was to determine if the flow conditions in the single-vane tunnel were sufficiently similar to those of a 10-vane cascade to permit meaningful heat transfer testing. The vane was tested over a range of ideal exit critical velocity ratios. The principal measurements were vane surface static pressures and cross-channel surveys of exit static pressure, total pressure, and flow angle. The report includes a brief description of the test vane and tunnels. The results of the exit surveys, the vane surface pressure distributions, and overall performance in terms of flow and loss for the two test configurations are compared.

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AERODYNAMIC PERFORMANCE OF A CORE-ENGINE TRUBINE STATOR VANE TESTED IN A TWO-DIMENSIONAL CASCADE OF 10 VANES AND IN A SINGLE-VANE TUNNEL

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SUMMARY

A turbine stator vane, with a profile suitable for use in a core engine was tested in a two-dimensional cascade of 10 vanes and in a single-vane tunnel. The single-vane tunnel used in this investigation was a modification of the tunnel used for the 10-vane cascade. It was designed to duplicate the flow conditions of a tunnel which will be used for high temperature heat transfer testing of cooled turbine vanes. The purpose of the investigation was to determine if the flow conditions in the single-vane tunnel were sufficiently similar to those of a 10-vane cascade to permit meaningful heat transfer testing.

The vane was tested in both cascade configurations over a range of ideal exit critical velocity ratios from about 0.6 to 0.93. A film-cooled vane was also tested in the single-vane configuration to study the effect of coolant ejection on the vane surface pressure distribution. Finally, the effect on the performance of the single-vane tunnel, of additional small air flows which will be used in conjunction with infrared instrumentation was also investigated. The principal measurements were vane surface static pressures and cross-channel surveys of exit total pressure, static pressure, and flow angle.

The results of the investigation indicated that the similarity of the flow conditions in the single-vane tunnel compared to those in the 10-vane cascade was acceptable for heat transfer testing of cooled core turbine vanes. However, one of the side walls in the single-vane tunnel did interfere with the flow at the vane exit. And it was recommended that the design of this side wall be modified in order to improve the performance of the single-vane tunnel.

INTRODUCTION

The NASA Lewis Research Center has been involved in investigations of the performance of turbine blading for advanced jet engines. As a part of this program a single-vane two-dimensional tunnel is being built. This tunnel is to be used for high temperature heat transfer investigations of cooled turbine stator vanes for a core engine. The purpose of the investigation reported herein was to determine if the flow field in such a single-vane tunnel was sufficiently similar to the flow field of a two-dimensional 10-vane cascade to permit meaningful heat transfer testing.

An uncooled vane with the same profile as the cooled core turbine vane was tested in a two-dimensional cascade of 10 vanes. Supplementary side walls were then installed in the cascade tunnel to duplicate the geometry of the single-vane tunnel and the tests were repeated. A cooled version of the core turbine vane was also tested in this single-vane configuration to determine the effect of cooling air on the flow field. Infrared instrumentation will be used in the single-vane tunnel for heat transfer tests. The optical windows for this instrumentation are to be kept clean by a wash of cool air. The effect of this additional air flow on tunnel performance was simulated in the subject investigation by bleeding air through similar ports in the supplementary side walls.

Both single- and 10-vane cascade configurations were tested over a range of ideal exit critical velocity ratios $(V/V_{cr})_{id,3}$ from approximately 0.60 to 0.93. The principal measurements were vane surface static pressures and cross-channel surveys of exit total pressure, static pressure and flow angle. The results of the investigation include comparison of the exit surveys, vane surface static pressure distributions, and overall performance in terms of weight flow and kinetic energy loss data of single- and 10-vane cascade configurations.

SYMBOLS

a	distance along axial chord from leading edge, cm; in.
c_a	vane axial chord, cm; in.
$\bar{e}$	kinetic energy loss coefficient for uncooled vanes, $1 - (V_3/V_{id,3})^2$
$\bar{e}_p$	primary kinetic energy loss coefficient for cooled vanes, $1 - \left[W_3V_3^3/W_p(V_{id,3})_p^2\right]$
$\bar{e}_T$	thermodynamic kinetic energy loss coefficient for cooled vanes, $1 - \left\{W_3V_3^2\Big/\left[W_p(V_{id,3})_p^2 + W_c(V_{id,3})_c^2\right]\right\}$
p	absolute pressure, N/cm^2; lb/in.^{2}

s	vane spacing, cm; in.
t	tangential distance from vane trailing edge, cm; in.
V	velocity, m/sec; ft/sec
W	flow rate per unit of vane span, kg/(sec)(cm); lb/(sec)(in.)
δ	ratio of inlet total pressure to U.S. standard sea-level atmospheric pressure, $p_1'/10.132\ N/cm^2$; $p_1'/14.696\ lb/in.^2$
$\sqrt{\theta_{cr}}$	ratio of inlet critical velocity to critical velocity of U.S. standard sea-level air, $V_{cr}/310.6$ m/sec; $V_{cr,1}/1019.2$ ft/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 at vane exit where flow conditions are assumed uniform

Superscript:

'	total static conditions

APPARATUS AND PROCEDURE

Test Vane

The test vane profile along with the vane coordinates and velocity diagram are shown in figure 1. This figure also shows the location of the vane surface static taps and the exit survey plane. The length of the vanes between cascade end walls was 10.16 centimeters (4 in.). The cooled vane had the same external profile as shown in figure 1. This was a film-cooled vane with small holes over the entire vane surface. The cooling holes were inclined in the chordwise direction so that the coolant flow was ejected substantially in the direction of the primary flow.

Cascade Tunnels

Figure 2 shows a view of the partially assembled 10-vane cascade tunnel. This same tunnel, but with the supplementary side walls used to simulate the single-vane tunnel, is shown installed in the test cell in figure 3. The geometry of this single-vane configuration is shown in figure 4.

In operation, room air was drawn through the cascade tunnel, blading, and exhaust control valve into the laboratory exhaust system. The vanes were tested over a range of inlet total to exit static pressure ratios corresponding to ideal exit critical velocity ratios of about 0.60 to 0.93.

Cooling air for the film-cooled vane was supplied to both ends of the vane from manifolds on the tunnel end walls. An ASME flat plate orifice was used to measure the cooling air flow. The location of the ports used to simulate the infrared bleed flow is shown in figure 4. These ports were installed after the basic performance testing was completed so that they would not interfere with the flow conditions in the tunnel. The flow through these ports was measured with small, choked-flow orifices.

Instrumentation

The two vanes that formed the center channel in the 10-vane cascade were instrumented at midspan with static pressure taps. The location of these taps is shown in figure 1. For most of the testing in the single-vane configuration, these instrumented vanes formed part of the tunnel side walls as shown in figure 4. These instrumented vanes were interchanged with the center vane to obtain the center vane surface pressure distribution. The cascade tunnels also had wall static pressure taps in the inlet and exit sections. These taps were used to determine uniformity of flow and set 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 exit total pressure, static pressure, and flow angle were surveyed simultaneously with the rake shown in figure 5. The total pressure was measured with a simple square-ended proble. 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 measures a differential pressure which is proportional to flow angle. Strain-gage transducers were used to measure these pressures.

The rake was fixed at the design exit flow angle which placed the sensing elements at the survey plane shown in figures 1 and 4. The rake was traversed over a distance of almost two vane spaces. The traverse speed was about 2.54 centimeters (1 in.) 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 the photographs of the manometer board. These data were used to calculate the vane surface pressure coefficients. 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 4.102 centimeters (1.615 in.) 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 uniform. This location is 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.

RESULTS AND DISCUSSION

In this section, the performance of the uncooled vane in the 10-vane cascade and in the single-vane tunnel are compared. Also, the effect of the infrared port bleed air and vane cooling air on the performance of the single-vane tunnel are discussed.

Performance Comparison

The results of the surveys of exit total and static pressure for the 10- and single-vane configuration at approximately design exit ideal critical velocity ratio were shown in figures 6(a) and (b), respectively. The total pressure wake for the single-vane case is larger than the wake for the 10-vane case. This is an indication that the loss, which is discussed later, will also be larger. For determining loss and weight flow, the survey data were integrated between values of the fraction of vane spacing t/s from 1.0 to 2.0 so that the total pressure wake would be in the center of the interval. Within this interval, the two static pressure traces are reasonably similar except for somewhat higher static pressures on the pressure surface side of the center vane in the single-vane tunnel. Near the supplementary side wall facing the suction side of the center vane in the single-vane tunnel, (at $t/s = 2.2$) the static pressure was lower than it was for the 10-vane tunnel. This is an indication that this side wall causes a local increase in velocity downstream of the vane trailing edge.

The vane surface pressure distributions are compared in figure 7. These pressures are presented as a vane surface pressure coefficient, which varies in a manner similar to velocity, against the fraction of axial chord. The vane surface pressure distribution for the 10-vane cascade (fig. 7(a)), and for the single-vane tunnel (fig. 7(b)); with the pressure taps on the side walls (see fig. 4) are quite similar. However, the pressure distribution on the center vane of the single-vane tunnel, shown in figure 7(c) is different. The pressure coefficient (velocity) on the suction surface near the trailing edge is significantly higher than it was in either the 10-vane cascade (fig. 7(a)) or on the side wall of the single-vane tunnel (fig. 7(b)). This is probably a reflection of the higher velocities on the side wall downstream of the trailing edge and opposite the suction surface of the center vane which was discussed in conjunction with the exit survey results. The pressure distribution on the suction surface of the center vane of the single-vane tunnel at about design exit velocity ratio is, in fact, quite similar to the pressure distribution found in the 10-vane cascade at exit velocity ratios much higher than design. A vane surface pressure distribution for the 10-vane cascade at an ideal exit critical velocity ratio (0.91) much higher than design (0.775) is shown in figure 8.

A comparison of the overall performance of the 10-vane cascade with the single-vane tunnel is made in terms of the equivalent weight flow and kinetic energy loss in figures 9(a) and (b), respectively. The equivalent flow for the 10-vane cascade increases with ideal exit critical velocity ratio in a manner typical of subsonic cascades. The equivalent flow for the single-vane tunnel follows this trend up to an exit velocity ratio of about 0.7. At this point the single-vane tunnel apparently begins to choke. At design ideal exit critical velocity ratio, 0.775, the equivalent flow per unit of vane length for the single-vane tunnel is 4 percent less than for the 10-vane cascade. Apparently, the side wall opposite the suction surface of the center vane and downstream of the trailing edge interferes with and restricts the flow at the vane exit. The reduced area in this region results in the higher velocities near the trailing edge on the suction surface of the center vane, reduced flow, and premature choking.

The kinetic energy loss coefficients for the two tunnel configurations are compared in figure 9(b). The loss for the single-vane tunnel was significantly higher than it was for the 10-vane cascade except at the lowest exit velocity ratio investigated. In the range of exit velocity ratios of greatest interest, from 0.7 to 0.8, the loss in the single-vane tunnel was about one point higher than the loss in the 10-vane cascade. The loss curve for the single-vane tunnel is similar in shape to the loss curve for the

10-vane cascade. However, it is displaced to the left - in the direction of lower exit velocity ratios. This is an indication that the higher loss was also the result of the higher velocities which occurred on the suction surface of the center vane in the single-vane tunnel.

Film-Cooled Vane

A single film-cooled vane was tested in the center vane position of the single-vane tunnel. The purpose of this test was to determine the effect of the vane cooling air on tunnel performance. The results of a survey of exit total and static pressure and the pressure distribution on the vanes forming the single-vane tunnel side walls are shown in figures 10(a) and (b), respectively. The data shown in this figure are for approximately design ideal exit critical velocity ratio and coolant supply total temperature and pressure equal to tunnel inlet total temperature and pressure, respectively. At these conditions, the coolant flow was 3.7 percent of the primary flow and the total flow (coolant plus primary) was about the same as it was for the uncooled vane in the single-vane tunnel.

The total pressure wake is larger than it was for the uncooled vane, indicating a larger loss which was expected. The kinetic energy loss based on primary flow only, $\bar{e}_p$ was 0.036 and the thermodynamic loss $\bar{e}_T$ which is based on the ideal energy of both primary and coolant flows was 0.071. The loss for the uncooled vane in the single-vane tunnel was 0.034. The vane-to-vane variation in static pressure is slightly smaller than for the uncooled vane, but the shape of the trace shown in figure 10(a) is very similar to that of the uncooled vane (fig. 6(b)). The distribution of vane side wall static pressure, shown in figure 10(b) is also quite similar to that of the uncooled vane (fig. 7(b)). From this information it was concluded that the flow conditions around the film-cooled vane were essentially the same as for the uncooled vane.

Effect of Air for Infrared Instrumentation

Two ports were installed in the supplementary side wall opposite the suction surface of the center vane of the single-vane tunnel. The location of these ports is shown in figure 4. These ports were used to supply air in order to simulate the air flow for the infrared instrumentation in the heat transfer tunnel. The air for these ports came from a 82.7-N/cm^2 (120-psia) filtered supply. It was individually metered and measured with small 1.88-millimeter (0.072-in.) choked-flow orifices installed in 6.35 millimeters (1/4 in.) lines for each port. The testing was done at approximately design ideal exit critical velocity ratio and over a range of flow from about 1 to 2.5 percent of primary flow for each port. This flow percentage was based on the 3.81 centimeters (1.5 in.) vane length used in the heat transfer tunnel. The design flows are 1.5 percent for the upstream port and 0.6 percent for the downstream port.

It was found that air flowing from the upstream port had virtually no effect on tunnel performance. Air flowing from the downstream port, however, further increased the velocity along the side wall downstream of the vane trailing edge and on the rear portion of the suction surface of the center vane. However, at the lowest flow investigated, which was higher than the projected design flow, this increase in velocity was quite small.

CONCLUDING REMARKS

The results of the investigation indicated that the similarity of the flow conditions in the simulated single-vane tunnel compared to those of the 10-vane cascade was acceptable for heat transfer testing of cooled core turbine vanes. The results also indicated that the performance of the single-vane tunnel could be improved by modifying the design of one of the side walls. The side wall opposite the suction surface and downstream of the vane trailing edge interfered with and restricted the flow at the vane exit. This resulted in reduced flow, increased loss and higher velocities on the suction surface of the center vane compared to the performance of the 10-vane cascade. Air flowing from the downstream infrared port aggravated these conditions. Although, at the infrared port flow rate projected for use in the heat transfer tunnel, this effect would be small. Accordingly, it was recommended that the angle of this side wall, downstream of the vane trailing edge be opened slightly and that a shallow relief be cut into the wall to accomodate the infrared port air flow.

Lewis Research Center,
National Aeronautics and Space Administration,
Cleveland, Ohio, January 22, 1973,
501-24.

Vane coordinates, in.

X	Y_L	Y_U
0	0.200	0.200
0.050	-----	.335
.100	-----	.395
.150	-----	.441
.200	-----	.477
.250	-----	.506
.300	.024	.528
.350	.046	.547
.400	.064	.561
.450	.079	.570
.500	.093	.576
.550	.105	.579
.600	.115	.581
.700	.129	.575
.800	.141	.562
.900	.148	.542
1.000	.151	.520
1.100	.150	.494
1.200	.145	.469
1.300	.139	.437
1.400	.129	.404
1.500	.117	.371
1.600	.103	.334
1.700	.087	.294
1.800	.070	.250
1.900	.051	.205
2.000	.033	.157
2.100	.010	.105
2.186	.035	.035

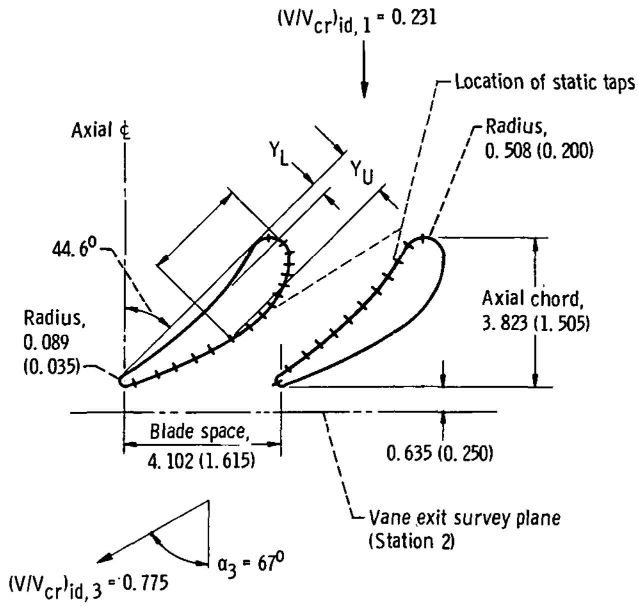

Figure 1. - Stator vane geometry. (All dimensions in cm (in.) except as noted.)

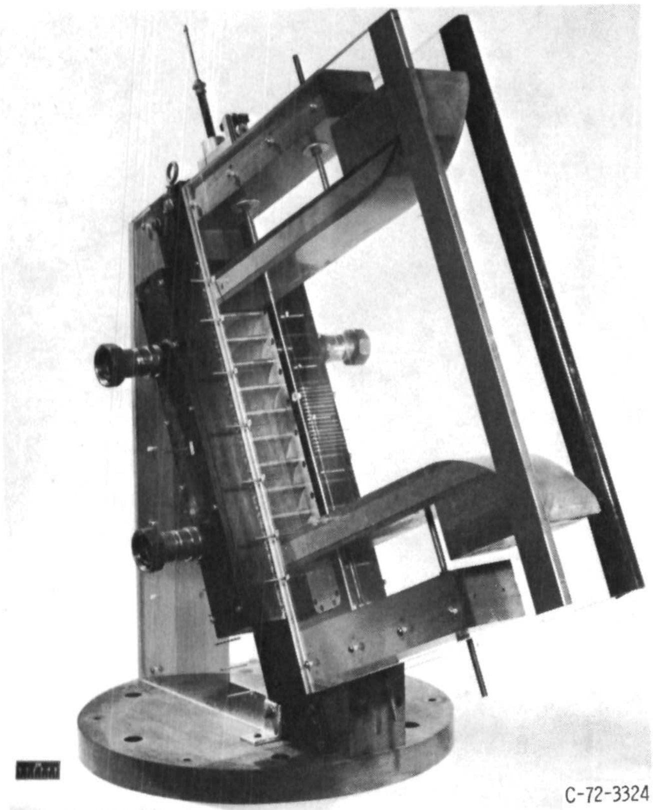

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Figure 2. - Ten-vane cascade tunnel.

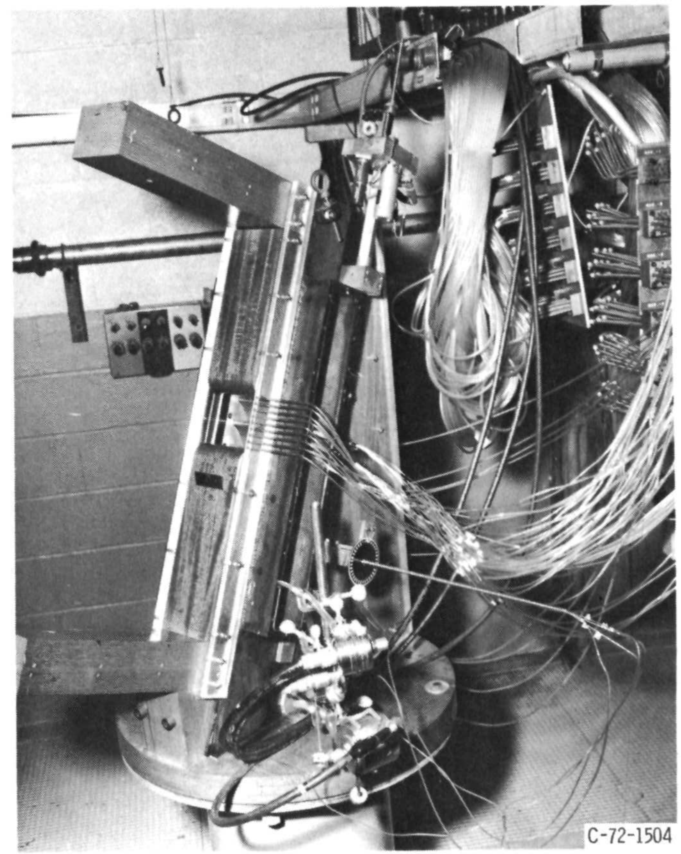

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Figure 3. - Supplementary side walls installed in 10-vane tunnel.

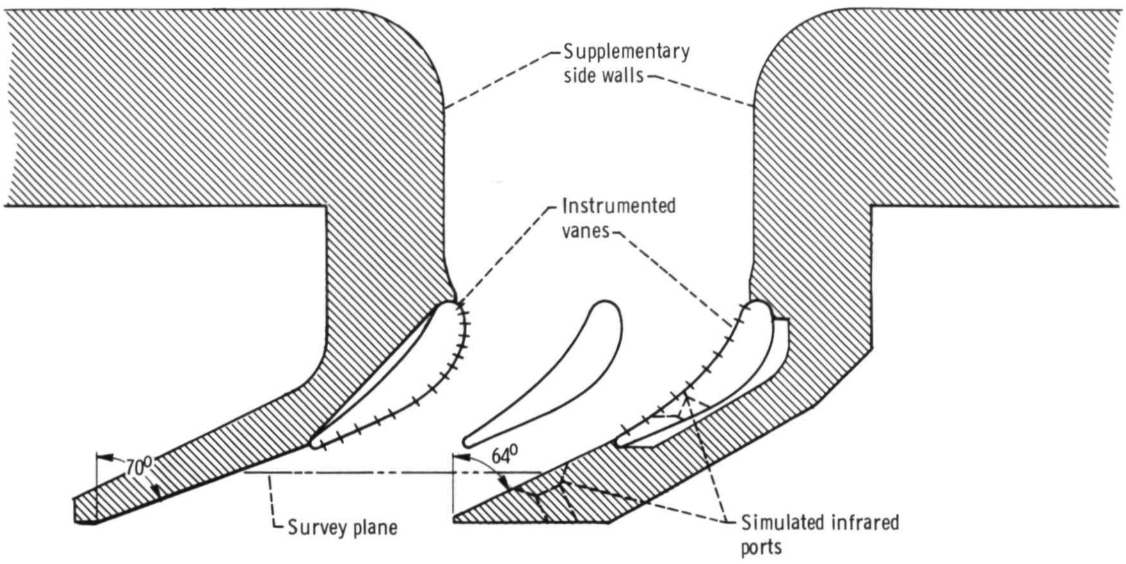


Figure 4. - Geometry of single-vane tunnel.

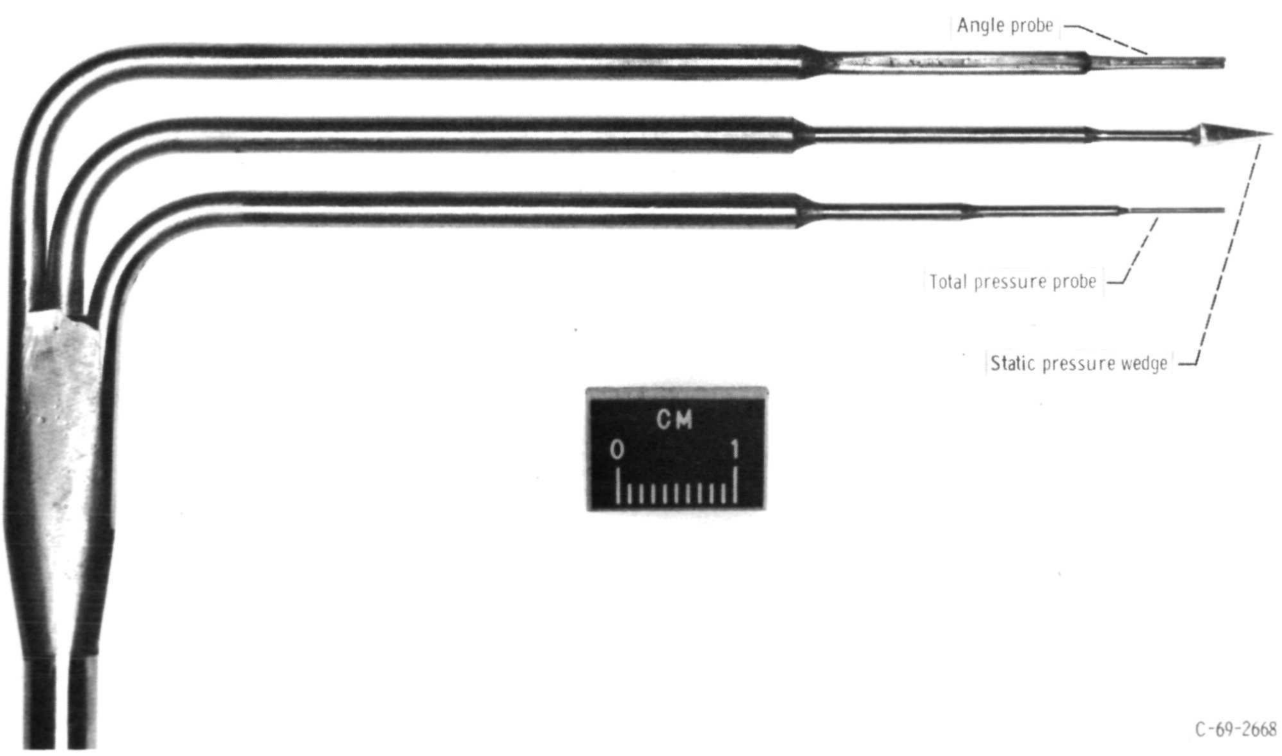
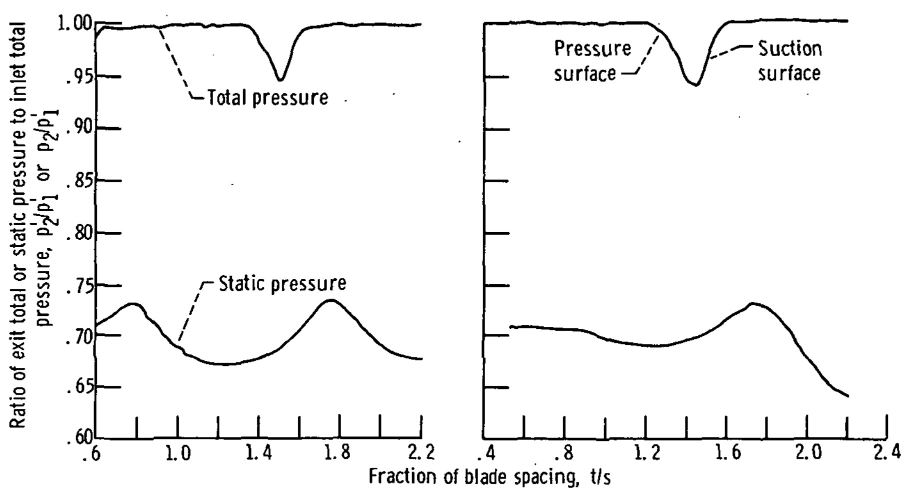
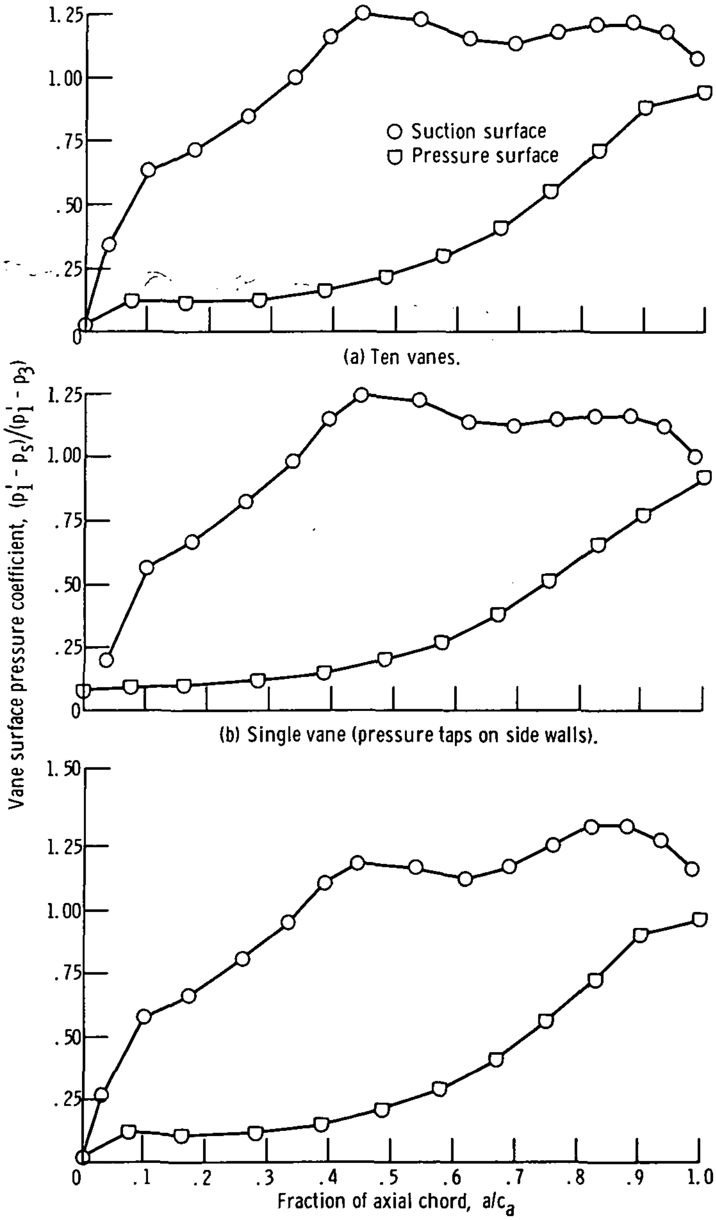


Figure 5. - Combination exit survey probe.



(a) Ten vanes. (b) Single vane.

Figure 6. - Comparison of vane-to-vane variation of exit total and static pressures for 10-vane and single-vane cascades at approximately design $(V/V_{cr})_{id,3}$.



(c) Single vane (pressure taps on center vane).

Figure 7. - Comparison of vane surface static pressures for 10-vane and single-vane cascades at approximately design $(V/V_{cr})_{id,3}$.

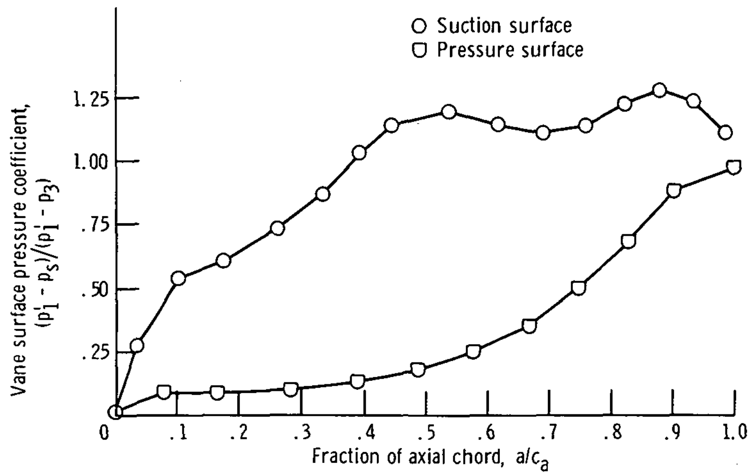
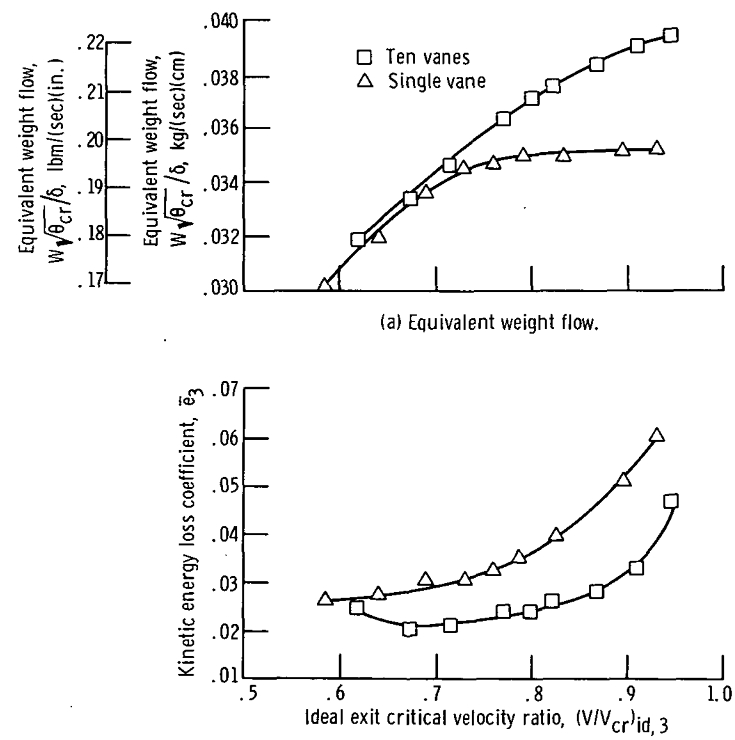


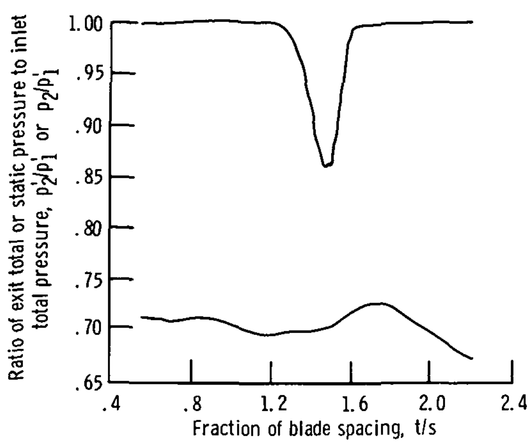
Figure 8. - Vane surface static pressures in 10-vane cascade at $(V/V_{cr})_{id,3} = 0.91$.



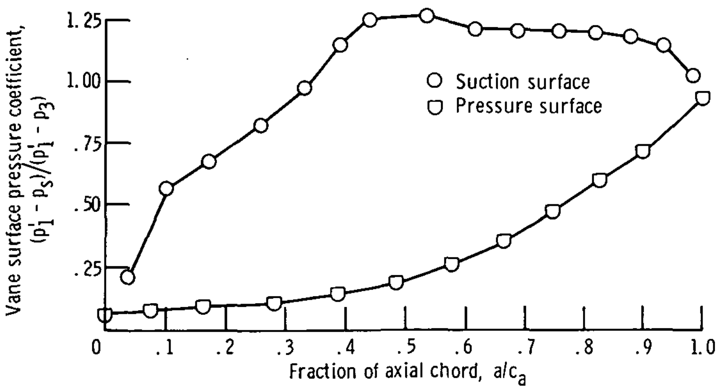
(a) Equivalent weight flow.

(b) Kinetic energy loss coefficient.

Figure 9. - Comparison of overall performance of uncooled vane in 10-vane and single-vane cascades.



(a) Vane-to-vane variation of exit total and static pressure.



(b) Vane surface static pressures (pressure taps on side walls).

Figure 10. - Variation of exit and vane surface pressures for film-cooled vane at approximately design $(V/V_{cr})_{id,3}$.

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