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A CASCADE INVESTIGATION OF A CONVECTION- AND FILM-COOLED TURBINE VANE MADE FROM RADIALLY STACKED LAMINATES

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SUMMARY

Laminated convection- and film-cooled turbine vanes were experimentally investigated in a four-vane cascade to determine the heat-transfer characteristics of this type of cooling concept. Data from this investigation are compared with published data of an impingement-, convection-, and film-cooled vane and a full-coverage film-cooled vane (made from laminated sheet-type material) and unpublished data of a transpiration-cooled vane. Because of the differences in vane geometry, test conditions, and test facilities among the vanes being compared, the cooling effectiveness comparisons are qualitative rather than quantitative. However, the data presented compare favorably with an impingement, convection-, and film-cooled vane and were apparently similar to a transpiration-cooled vane and a full-coverage film-cooled vane. The vanes tested were designed for a gas temperature and pressure of 1640 K (2500^{o} F) and 31 newtons per square centimeter (45 psia), respectively, a coolant temperature of 920 K (1200^{o} F), and a coolant- to gas-flow ratio of 0.0487. Scaled gas temperature conditions and design gas conditions, bracketing the design point, were investigated and are reported herein.

INTRODUCTION

An experimental heat-transfer investigation was conducted on an air-cooled turbine vane made from photoetched laminates. This concept is unique in that laminates, with photoetched cooling passages, were stacked radially and diffusion bonded to form an integral vane airfoil and end platforms.

Transpiration-cooled turbine vanes have been shown to be the most efficient means of tolerating the high turbine inlet temperatures expected in future gas turbine engines (ref. 1). However, application of transpiration cooling, using conventional porous materials, is limited because of the fabrication problems, susceptibility to foreign object damage, and susceptibility to plugging of the air flow passages by oxidation and contaminated coolant. Many of these materials (wire wound, woven wire cloth, or sintered powered metal) are generally of low strength and require some type of supporting strut which increases the airfoil weight and adds to the fabrication complexity. Furthermore, the oxidation rate of these materials increases rapidly with increasing temperature resulting in the plugging of the small coolant passages in the porous material. The oxidation characteristics of a laminated sheet-type and a wireform-type porous material are examined in reference 2 where it is shown that the laminated sheet-type material is less susceptible to internal clogging than a wireform-type porous material. Reference 2 also discusses the heat-transfer and flow characteristics of a full-coverage film-cooled vane fabricated from a laminated sheet-type material. One type of wire alloy (GE 1541) that has proved to be highly oxidation resistant has been developed but has not been used extensively in actual transpiration cooled vane or blade application. The oxidation characteristics of this material are discussed in reference 1.

The laminated convection- and film-cooled turbine vane concept investigated herein was conceived by the Aerojet Liquid Rocket Company and the fabrication approach was based on their experience with laminated fuel injectors developed for a liquid-fuel rocket application. The vanes were designed for a gas temperature and pressure of 1640 K (2500° F) and 31 newtons per square centimeter (N/cm^2) (45 psia), respectively, a coolant temperature of 920 K (1200° F), and a coolant- to gas-flow ratio of 0.0487. The maximum design vane temperature was established as 1255 K (1800° F).

The experimental tests were made at nominal gas temperatures and pressures of 1255 and 1640 K (1800° and 2500° F) and 22.7 and 31 N/cm^2 (33 and 45 psia), respectively, coolant temperatures of 550 and 700 K (530 and 800° F), and coolant- to gas-flow ratios from 0.0404 to 0.635. The data reported herein are also compared with data from the laminated sheet-type vane of reference 2 - an impingement-, convection-, and film-cooled vane and a transpiration (sintered wire wound cloth) cooled vane.

SYMBOLS

A	surface area
h	heat-transfer coefficient
I_0, I_1	modified Bessel functions
k	thermal conductivity
L	length of suction or pressure surface

m $\sqrt{2h/kq}$

q fin width at base

r fin length

s distance between fins

T temperature

w mass flow rate

x distance from stagnation point

ΔTR temperature difference ratio, $(T_{g,M} - T_{c,s})/(T_{g,M} - T_c)$

η film-cooling effectiveness, $(T_{g,M} - T_f)/(T_{g,M} - T_{c,s})$

ϕ cooling effectiveness, $(T_{g,M} - T_w)/(T_{g,M} - T_{c,i})$

Subscripts:

c coolant

ci coolant inlet

cor corrected

e effective

f film

g gas

H hub

M midspan

s exit of film-cooling holes

T tip

w vane wall

APPARATUS

Cascade Description

A four-vane, five-channel cascade was used to obtain the data presented herein. A detailed description of this facility is given in reference 3. The test section was a 23° annular sector of a vane row and contained four vanes and five flow channels. A schematic cross-sectional view of the cascade test section is shown in figure 1. The central two vanes (locations 2 and 3) are generally used as the test vanes and contain the primary instrumentation. However, for these tests, the outer two vanes (locations 1 and 4) also contained some thermocouple instrumentation and are referred to as slave vanes. The slave vanes normally complete the flow channels for the two test vanes and also serve as radiation shields between the test vanes and the water-cooled cascade walls.

The cooling air was supplied to the four vanes from a single manifold and one venturi type flowmetering station. Prior to entering the manifold, the air was filtered by a 5-micrometer sintered-element filter.

Vane Description

The laminated vane discussed herein used the full-coverage film-cooling concept for the forward two-thirds of the vane surface and convection cooling in the aft portion of the vane. The overall dimensions of the vane were similar to a first-stage J-75 sized turbine vane with a chord of 6.28 centimeters (2.47 in.) and a span of 9.78 centimeters (3.85 in.). The airfoil portion of the vane was formed by stacking 0.25-millimeter- (0.010-in.-) thick laminates (platelets) which contained photoetched coolant passages. The inner and outer diameter platforms were formed by stacking 2.3-millimeter- (0.090-in.-) thick laminates. All laminates were made from thoria dispersed nickel-chromium material. An integral vane assembly was formed by diffusion bonding the laminates in a single operation. A schematic cutaway of a completed assembly is shown in figure 2(a).

Two platelets were required to form a complete film-cooling slot passage. Each passage was composed of three segments, - a pore, a plenum, and a slot. An enlarged view of the pore-plenum-slot combination is shown schematically in figure 2(b). The lower platelet in the figure contained a nozzle-shaped metering pore and a plenum and is referred to as the pore-plenum platelet. The upper platelet contained the slot, which interconnected with the plenum, and is referred to as the slot platelet. A cross-sectional schematic of each type of platelet is shown in figure 3(a) and (b), respectively. Each slot in the vane was the same size and was photoetched 1.52 millimeters (0.060 in.) wide by 0.076 millimeter (0.003 in.) deep. The chordwise spacing between slots was 0.76 millimeter (0.030 in.). Because two platelets were required to form the pore-plenum-slot combination, each combination was required to cool a surface area of about 1.16 square millimeters (0.0018 in.^{2}). The minimum coolant flow needed to cool this area of the vane required a pore size equivalent to the minimum area obtained from a 0.051-millimeter (0.002-in.) depth of etch. There were 47 pore-plenum-slot combinations in each pair of platelets, and approximately 8800 combinations were in the complete vane.

The hollow core of the vane served as a plenum for each of the film-cooling passages and the convection-cooling passages of the trailing edge. To reduce the heat pickup by the coolant in this plenum a perforated metal insert was incorporated into the design of the airfoil.

There was an insufficient wall thickness in the trailing edge region to incorporate the pore-plenum-slot combination. Therefore, as mentioned previously, the trailing edge was convection cooled by photoetched chordwise passages in each platelet as shown in figures 3(a) and (b). Each passage cross section was 0.20 by 0.76 millimeter (0.008 by 0.030 in.) and 15.24 millimeters (0.60 in.) long.

For the design gas conditions (temperature and pressure of 1640 K (2500° F) and 31 N/cm^2 (45 psia), respectively) and a coolant inlet temperature of 920 K (1200° F), the calculated required coolant- to gas-flow ratio was 0.0487 to maintain an allowable maximum wall temperature of 1255 K (1800 F). Approximately 45 percent of this coolant flow was required for the convection-cooled trailing edge region.

The vanes were fabricated with an untwisted profile, similar to the midspan profile of a J-75 size turbine vane, to simplify the fabrication procedure and reduce the overall cost. In order to accommodate the untwisted profile within the J-75 platform geometry, the angle of attack was changed by 4°. In addition, the external profile was made oversized by 0.76 millimeter (0.030 in.) such that a final EDM operation was required to finish the vane profile to the proper dimensions and remove the excess material and small cracks resulting from the photoetching and bonding operations, respectively. The excess material referred to previously is shown schematically in figure 3(c). This view is a spanwise cross-sectional enlargement of the gas-side surface. The photoetching process, in effect, produced chordwise fins on both the interior (plenum) and exterior (gas-side) surfaces. The data reported herein were generated prior to the removal of this excess material on the gas-side surface. This approach was taken to guard against the possibility that the metering pores would become clogged during the final EDM operation.

INSTRUMENTATION

The combustion gas inlet conditions were measured by spanwise traversing probes. Gas stream hub and tip static pressures were also measured at the vane row trailing edge. These locations are shown in figure 1. This instrumentation was used to establish the gas operating conditions. This and other operational instrumentation are discussed in reference 3.

Twenty sheathed chromel-alumel thermocouples were located on the midspan of vanes 2 and 3. Vanes 1 and 4 contained a total of eight midspan thermocouples. See figure 4 for a schematic layout of each thermocouple position. Table I indicates each thermocouple position by vane number, surface distance from stagnation point, and sheath size. The majority of thermocouples had a 0.25-millimeter (10-mil) sheath diameter, but nine of them had a 0.51-millimeter (20 mil) sheath diameter. The fabrication and installation techniques used are described in reference 4.

TEST PROCEDURE

The design conditions for the vanes reported herein were a gas temperature and pressure of 1640 K (2500° F) and 310 N/cm^2 (45 psia), respectively, a coolant temperature of 920 K (1200° F), a coolant- to gas-flow ratio of 0.0487, a maximum design vane temperature of 1255 K (1800° F), and a midspan vane row exit Mach number of 0.85. Because of the short life expectancy of the small sheathed thermocouples and the difficulty of obtaining a coolant temperature of 920 K (1200° F), data were taken initially at gas and coolant conditions scaled from these design conditions. The scaling procedure is discussed in reference 5. The scaled gas conditions were 1255 K (1800° F) and 22.7 N/cm^2 (33 psia). The scaled coolant temperature was 710 K (820° F).

A second series of test points was run at the same scaled gas conditions but at a lower coolant temperature of 550 K (530° F). At the design gas conditions this later scaled coolant temperature was representative of a 700 K (800° F) actual coolant temperature. This latter test point was also run. See table II for a list of the test points taken.

RESULTS AND DISCUSSION

The experimental data generated by this investigation are summarized in table III. The coolant flow rate w_c listed in the table was the total flow to the four vanes. Flow calibration tests at ambient temperature indicated that each test vane (vanes 2 and 3) used about 28.3 percent of the total coolant flow. The gas flow rate w_g is listed on a per vane prorated basis. The coolant- to gas-flow ratio for each test vane can be calculated from the previous information and

$$\frac{w_c}{w_g} = \frac{w_{c,\,total} \times 0.283}{w_g} \tag{1}$$

Because of the coolant flow variation between the test vanes (2 and 3) and the slave vanes (1 and 4), only data for vanes 2 and 3 are presented and discussed in detail.

Temperature Distribution

A typical midspan temperature distribution is shown in figure 5 where the data from vanes 2 and 3 have been combined to form a composite. These data were taken from data set 1 of table III and represent the scaled design condition with the exception that w_c/w_g was 0.003 less than the design w_c/w_g. As shown by figure 5, the temperature distribution over the vane surface was nearly uniform with a maximum to minimum temperature difference of about 156 K (280 F deg).

Vane Cooling Effectiveness

A local cooling effectiveness is defined by

$$\varphi = \frac{T_{g,M} - T_w}{T_{g,M} - T_{ci}} \tag{2}$$

The gas and coolant temperatures $T_{g,M}$ and T_{ci} used in equation (2) were total inlet values. The wall temperature T_w was a local value. Local cooling effectiveness values (defined by eq. (2)), for selected locations on the test vanes are shown in figures 6(a) to (i) as functions of the total coolant- to gas-flow ratio w_c/w_g.

It was mentioned in the Vane Description section that the vanes tested had, in effect, chordwise fins on the gas-side surface which would increase the heat transfer to the vanes. In order to compare these data with other data from transpiration- or full-coverage, film-cooled vanes it is necessary to correct the data presented herein for this fin effectiveness. Equation (2) can also be written as (ref. 7)

$$\varphi = \frac{1 + \frac{h_{g,fin}}{h_c}\frac{A_g}{A_c}\eta\Delta TR}{1 + \frac{h_{g,fin}}{h_c}\frac{A_g}{A_c}} \tag{3}$$

If a fin effectiveness is defined as

$$\frac{h_{g,fin}}{h_g} = \left(\frac{1}{s+q}\right)\left[\left(\frac{2}{m}\right)\frac{I_1(2mr)}{I_0(2mr)} + s\right] \tag{4}$$

equation (3) can be corrected for the fin effectiveness provided the product of the film effectiveness and the temperature difference ratio $\eta\Delta TR$ is known. The dimensions, as used in equation (4), are shown in figure 3(c). A corrected cooling effectiveness can be calculated using the experimental cooling effectiveness and the fin effectiveness ($h_{g,fin}/h_g = 2.1$, where m was equal to 25.8):

$$\varphi_{cor} = \frac{1 + \frac{h_g}{h_{g, fin}}\left(\frac{1 - \varphi}{\varphi - \eta\Delta TR}\right)\eta\Delta TR}{1 + \frac{h_g}{h_{g, fin}}\left(\frac{1 - \varphi}{\varphi - \eta\Delta TR}\right)} \tag{5}$$

Since φ_{cor} is a function of $\eta\Delta TR$, which is not known for the laminated vane, the cooling effectiveness correction was made using equation (6) where $\eta\Delta TR$ has been assumed to equal zero:

$$\varphi_{cor} = \frac{1}{1 + \frac{h_g}{h_{g, fin}}\left(\frac{1 - \varphi}{\varphi}\right)} \tag{6}$$

The comparisons discussed in the following paragraphs are based on data corrected using equation (6) and, therefore, provide qualitative rather than quantitative comparisons.

In the following discussion of the data, comparisons are made with data from other cooling concepts such as an impingement-, convection-, and film-cooled vane, a full-coverage film-cooled vane, and a transpiration-cooled vane. Cross-sectional views of these vanes are shown in figure 6.

Comparison of laminated vane with reference 7 vane. - The cooling effectiveness values of the laminated vane and an impingement-, convection-, and film-cooled vane tested in the same cascade and reported in reference 7 are compared in figure 7. The reference 7 vane was impingement cooled in the leading edge and midchord regions and had a convection-cooled (split) trailing edge region with film cooling over the aft portion of both the suction and pressure surfaces. The cooling effectiveness of the laminated vane is shown in figure 7 to be better than the cooling effectiveness of the reference 7 vane. The results of the comparison in the midchord and trailing edge regions are particularly significant because of the oversized and roughened profile of the laminated vane. This type of surface could have resulted in significantly higher unblown heat-transfer coefficients, particularly on the suction surface, since the vane of reference 7 probably had a laminar and transitional boundary layer whereas it is probable that a turbulent boundary layer existed on the laminated vane. This phenomenon was not accounted for by the correction of equation (6).

Also of importance in the leading edge comparison is that, by prorating an equal amount of cooling air to each pore, plenum, and slot, approximately 15 percent of the total vane cooling air was used in the leading edge region. Proper consideration of the static gas pressure distribution suggests the actual coolant flow to the leading edge region was less than 15 percent. This contrasts with the reference 7 vane which used between 25 and 30 percent of the total coolant flow in the leading edge region.

Comparison of laminated vane with reference 8 vane. - Cooling effectiveness data are also shown in figure 7 for selected regions of a full-coverage film-cooled vane fabricated from a laminated sheet-type material. These data were taken from reference 8. It can be seen that these data are similar to the data reported herein. However, this comparison may be somewhat misleading in that there were several notable differences associated with the data compared. Because of the simplified nature of the parameter φ used in the comparison and the correction required for data reported herein, these differences may be significant. In particular, these two laminated-type vanes had different profiles and chord lengths and were tested in different cascades at different gas conditions.

Comparison of laminated vane with porous wire cloth vane. - The dashed lines in figure 7 represent unpublished data taken from a transpiration-cooled vane tested in the cascade facility described in this report. The vane airfoil was fabricated from sintered wire wound cloth attached to a central strut. As noted in the figure, this type of vane (transpiration cooled) had a somewhat higher cooling effectiveness than the vane of reference 8 and the laminated vane of this report. Again, it should be noted that the differences in geometry, test facilities, and gas conditions between reference 8 data, the unpublished wire cloth vane data, and the data presented herein preclude quantitative comparisons of cooling effectiveness.

Comparison of scaled and actual test conditions. - Another significant result shown in figure 7 is the correlation of scaled test conditions (data sets 4, 5, and 6) and the actual test conditions (data set 7). This indicates the scaling procedure used was reasonable and that the cooling effectiveness at the design point can be deduced from the data given. Therefore, using equation (6) to find the "hot spot" wall temperature, based on the design conditions and the corrected experimental data here (thermocouple 80, fig. 6(b)), results in a 1220 K (1735° F) wall temperature which is 35 K (65 deg F) less than the maximum wall temperature for which the laminated vane was designed.

CONCLUDING REMARKS

The heat-transfer characteristics of a laminated convection-, and film-cooled turbine vane were investigated in a four-vane cascade. The design-point gas and coolant temperatures and the design-point coolant- to gas-flow ratio were bracketed by scaled experimental test conditions. These experimental data were compared with other published data and unpublished NASA data. The data reported herein were generated prior to a final profiling operation to guard against the possibility that this final fabrication step would clog the metering pores in the vanes and render the vanes unusable for heat-transfer tests. A fin effectiveness factor, representing this excess material on the vane surface, was used to correct the cooling effectiveness data presented. Due to the uncertainty in the cooling effectiveness correction and due to the differences in vane geometry, test conditions, and test facilities among the vane compared, the cooling effective ness comparisons presented below are qualitative rather than quantitative. The results indicate the following:

(1) The pore, plenum, slot concept provided a nearly uniform temperature distribution in the chordwise plane with a maximum to minimum temperature difference of [illegible] K (280 F deg) at scaled design conditions.

(2) The calculated maximum wall temperature, based on scaled design conditions, was approximately 1200 K (1735° F) which was about 35 K (65 F deg) less than the allowable wall temperature for actual design conditions.

(3) The corrected cooling effectiveness compared favorably with an impingement-, convection-, and film-cooled vane.

(4) The corrected cooling effectiveness was apparently comparable to that of a full-coverage film-cooled vane made from laminated sheet-type material but slightly less than that for a transpiration-cooled vane made from wire cloth.

Lewis Research Center,
National Aeronautics and Space Administration,
Cleveland, Ohio, August 1, 1974
501-24.

REFERENCES

1. Esgar, Jack B.; Colladay, Raymond S.; and Kaufman, Albert: An Analysis of the Capabilities and Limitations of Turbine Air Cooling Methods. NASA TN D-5992, 1970.

2. Hickel, Robert O.; and Warren, Edward L.: Experimental Investigation of the Flow, Oxidation, Cooling, and Thermal-Fatigue Characteristics of a Laminated Porous Sheet Material. NASA TN D-6664, 1972.

3. Calvert, Howard F.; Cochran, Reeves P.; Dengler, Robert P.; Hickel, Robert O.; and Norris, James W.: Turbine Cooling Research Facility. NASA TM X-1927, 1970.

4. Crowl, Robert.; and Gladden, Herbert J.: Methods and Procedures for Evaluating, Forming, and Installing Small-Diameter Sheathed Thermocouple Wire and Sheathed Thermocouples. NASA TM X-2377, 1971.

5. Gladden, Herbert J.; and Livingood, John N. B.: Procedure for Scaling of Experimental Turbine Vane Airfoil Temperatures from Low to High Gas Temperature. NASA TN D-6510, 1971.

6. Gladden, Herbert J.; Gauntner, Daniel J.; and Livingood, John N. B.: Analysis of Heat-Transfer Tests of an Impingement-Convection-Film-Cooled Vane in a Cascade. NASA TM X-2376, 1971.

7. Nealy, D. A.; Anderson, R. D.; and Hufford, A. A.: Design and Experimental Evaluation of a Turbine Vane Fabricated from Laminated Porous Material. EDR-6296, Allison Division, General Motors (NASA CR-72649), 1969.

TABLE I. - LOCATION AND SIZE OF THERMOCOUPLES

Thermocouple	Dimensionless surface distance, x/L[a]	Sheath diameter, mm
Vane 1		
62	0.820	0.51
63	.397	.25
64	.134	
65	.723	
66	.548	
67	.280	
68	.033	
Vane 2		
75	0.883	0.51
76	.822	.51
77	.649	.25
78	.384	
79	.119	
80	.059	
81	.236	
82	.496	
92	.694	
83	.798	.51
84	.873	.51
74	.399	.25

Thermocouple	Dimensionless surface distance, x/L[a]	Sheath diameter, mm
Vane 3		
85	0.883	0.51
86	.737	.25
87	.473	
88	.206	
89	.051	
90	.169	
91	.399	
97	.798	.51
98	.873	.51
99	.822	.51
100	.384	.25
101	.119	.25
Vane 4		
69	0.038	0.25
70	.290	
71	.491	
72	.652	
73	.426	

[a]Suction surface length, 7.68 cm; pressure surface length, 6.62 cm.

TABLE II. - NOMINAL TEST CONDITIONS[a]

Data set	Total gas pressure		Total gas temperature		Coolant temperature		Coolant- to gas-flow ratio
	N cm^2	psia	K	^oF	K	^oF	
[b]1	22.7	33	1255	1800	710	820	0.0458
[b]2	↓	↓	↓	↓	710	820	.0535
[b]3	↓	↓	↓	↓	710	820	.0597
4	↓	↓	↓	↓	550	530	.0515
5	↓	↓	↓	↓	550	530	.0404
6	↓	↓	↓	↓	550	530	.0635
7	31.0	45	1640	2500	700	800	.0536

[a]See table III for actual test conditions.

[b]Represents scaled design conditions.

TABLE III. - EXPERIMENTAL DATA GENERATED FROM LAMINATED VANES

[Total inlet gas pressure, $P_{g,L}$, 31.0 N/cm^2.]

(a) Vane 1.

Data set	Total inlet gas temperature, K			Inlet coolant temperature, $T_{c,i}$, K	Gas flow per vane, W_g, kg/sec	Total coolant flow, $W_{c,total}$, kg/sec	Thermocouple						
	Hub, $T_{g,H}$	Midspan, $T_{g,M}$	Tip, $T_{g,T}$				62	63	64	65	66	67	68
							Tip			Midspan			
							Temperature, K						
4	1209.5	1289.2	1156.1	543.8	0.308	0.05616	785.9	868.7	901.7	838.8	909.0	949.7	974.1
5	1210.0	1294.4	1159.4	539.8	.308	.04394	811.0	898.6	749.6	872.1	947.8	974.8	1028.6
6	1194.6	1294.8	1294.8	560.8	.299	.06707	782.7	858.2	879.6	820.2	882.3	932.9	929.4
1	1194.3	1282.2	1145.2	708.1	.303	.04907	874.5	930.4	963.2	927.4	983.4	1005.4	1023.5
2	1196.1	1279.9	1136.3	715.1	.303	.05734	870.4	919.6	946.1	914.9	967.8	997.4	1001.4
3	1195.7	1281.6	1135.2	727.0	.303	.06381	866.2	913.4	933.8	906.8	956.1	989.6	985.4
7	1534.7	1665.5	1448.0	706.2	.372	.07049	1028.9	1138.3	1184.1	1098.7	1208.2	1274.3	1317.8

(b) Vane 2.

Data set	Total inlet gas temperature, K			Inlet coolant temperature, $T_{c,i}$, K	Gas flow per vane, W_g, kg/sec	Total coolant flow, $W_{c,total}$, kg/sec	Thermocouple											
	Hub, $T_{g,H}$	Midspan, $T_{g,M}$	Tip, $T_{g,T}$				75	76	77	78	79	80	81	82	92	83	84	74
							Midspan											Tip
							Temperature, K											
4	1209.5	1289.2	1156.1	543.8	0.308	0.05616	852.1	826.1	830.1	920.3	910.6	1000.2	989.1	974.4	905.4	871.7	880.0	917.6
5	1210.0	1294.4	1159.4	539.8	.308	.04394	888.1	864.1	877.7	975.1	996.1	1085.6	1067.6	1032.1	949.2	907.1	913.5	962.7
6	1194.6	1294.8	1159.7	560.8	.299	.06707	833.9	806.5	804.1	889.7	862.8	938.8	930.5	937.0	877.9	852.3	862.7	892.1
1	1194.3	1282.2	1145.2	708.1	.303	.04907	951.9	933.6	940.0	1005.2	995.5	1064.1	1059.6	1019.4	997.2	967.6	972.3	983.9
2	1196.1	1279.9	1136.3	715.1	.303	.05734	935.8	916.9	918.7	981.2	956.2	1019.5	1020.4	1003.7	975.6	950.1	956.9	966.5
3	1195.7	1281.6	1135.2	727.0	.30[illegible]	.06831	927.2	908.2	907.1	968.5	939.1	994.4	996.1	1046.9	963.7	941.2	948.9	956.6
7	1534.7	1665.5	1448.0	706.2	.372	.07049	1096.2	1063.1	1071.8	1198.1	1181.3	1296.4	1313.9	1292.1	1180.1	1127.0	1136.4	1178.0

(c) Vane 3.

Data set	Total inlet gas temperature, K			Inlet coolant temperature, $T_{c,i}$, K	Gas flow per vane, W_g, kg sec	Total coolant flow, $W_{c,total}$, kg sec	Thermocouple											
	Hub, $T_{g,H}$	Midspan, $T_{g,M}$	Tip, $T_{g,T}$				85	86	87	88	89	90	91	97	98	99	100	101
							Midspan											Tip
							Temperature, K											
4	1209.5	1289.2	1156.1	543.8	0.308	0.05616	837.5	794.3	875.4	880.8	976.6	926.6	961.6	855.6	861.1	738.9	844.4	886.1
5	1210.0	1294.4	1159.4	539.8	.308	.04394	871.3	832.2	922.3	925.2	1052.9	1008.6	1022.8	891.7	900.0	769.4	883.3	938.9
6	1194.6	1294.8	1159.7	560.8	.299	.06707	817.2	772.1	843.2	858.7	919.2	866.1	919.5	833.3	844.4	730.6	833.3	851.7
1	1194.3	1282.2	1145.2	708.1	.303	.04907	937.4	908.2	968.2	962.3	1040.9	1005.0	1033.1	950.0	955.6	844.4	919.4	951.7
2	1196.1	1279.9	1136.3	715.1	.303	.05734	921.2	890.4	946.1	943.2	1000.2	965.1	1003.6	936.1	944.4	838.9	905.6	925.0
3	1195.7	1281.6	1135.2	727.0	.303	.06831	913.1	882.0	933.4	936.1	978.2	942.6	987.7	905.6	919.4	833.3	897.2	913.9
7	1534.7	1665.5	1448.0	706.2	.372	.07049	1072.3	1017.9	1129.3	1130.3	1232.7	1179.2	1231.5	1100.0	1105.6	944.4	1061.1	1107.8

(d) Vane 4.

Data set	Total inlet gas temperature, K			Inlet coolant temperature, $T_{c,i}$, K	Gas flow per vane, W_g, kg sec	Total coolant flow, $W_{c,total}$, kg sec	Thermocouple				
	Hub, $T_{g,H}$	Midspan, $T_{g,M}$	Tip, $T_{g,T}$				69	70	71	72	73
							Midspan				Tip
							Temperature, K				
4	1209.5	1289.2	1156.1	543.8	0.308	0.05616	1016.2	1025.4	------	926.2	879.3
5	1210.0	1294.4	1159.4	539.8	.308	.04394	1061.0	1059.3	------	957.6	938.4
6	1194.6	1294.8	1159.7	560.8	.299	.06707	975.8	1005.7	------	907.1	881.7
1	1194.3	1282.2	1145.2	708.1	.303	.04907	1051.1	1064.7	------	991.6	945.2
2	1196.1	1279.9	1136.3	715.1	.303	.05734	1023.6	1046.4	------	976.7	931.9
3	1195.7	1281.6	1135.2	727.0	.303	.06831	1006.3	1033.8	------	968.5	924.1
7	1534.7	1665.5	1448.0	706.2	.372	.07049	1293.4	1297.3	------	1189.8	1157.8

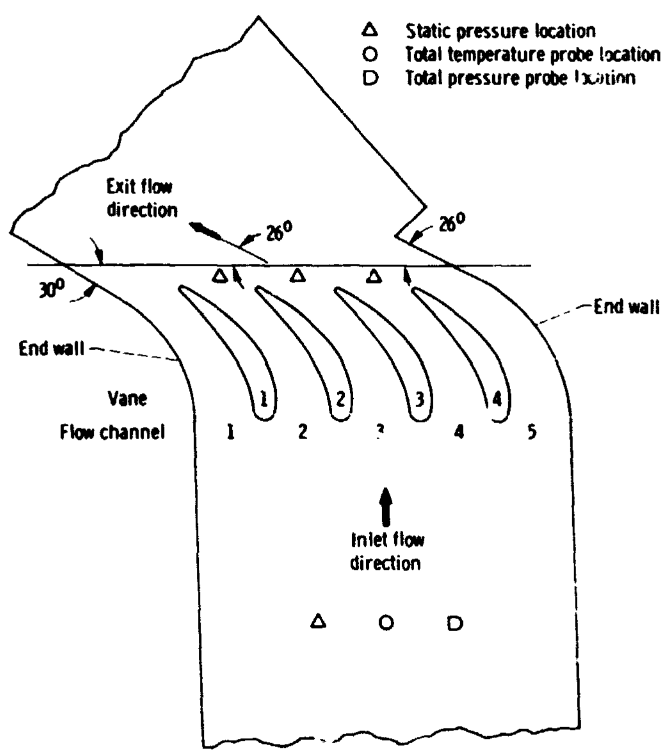
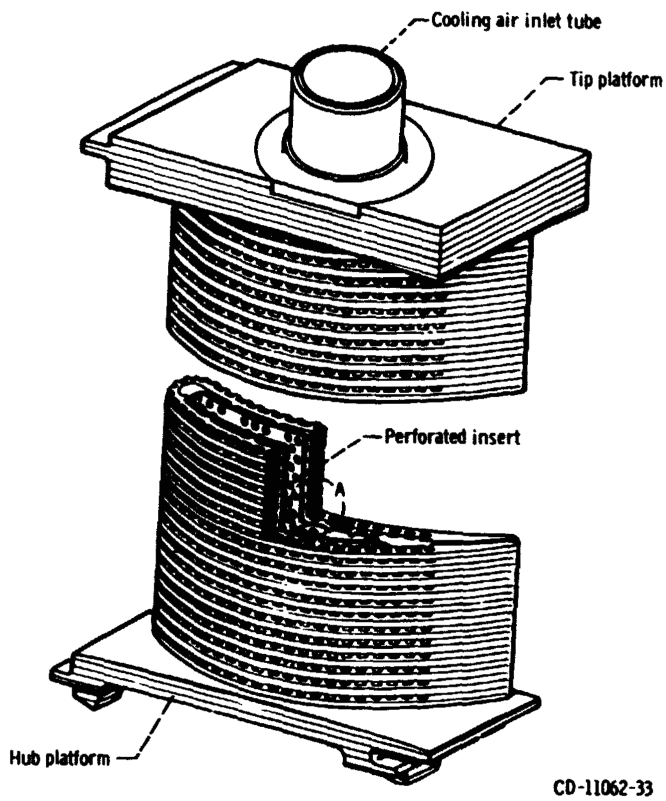
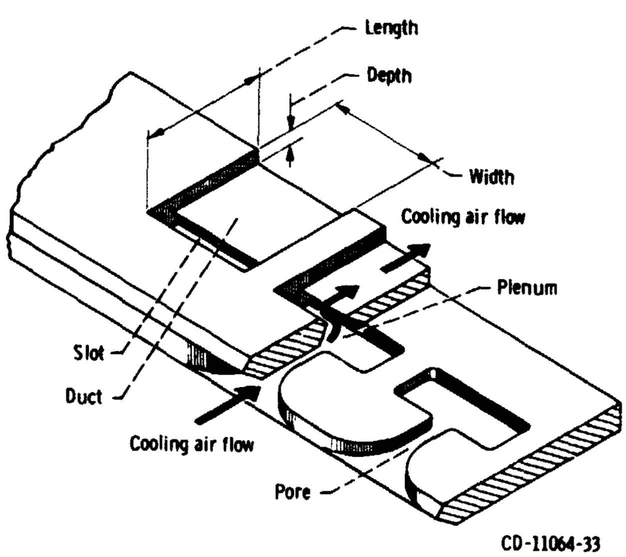


Figure 1. - Cross-sectional schematic of cascade test section.

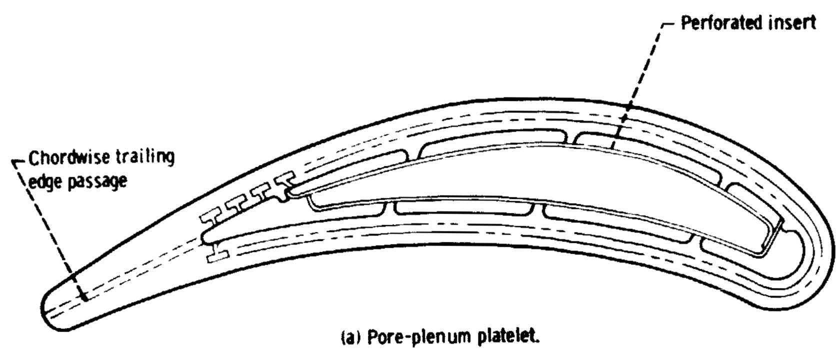


(a) Cutaway view.

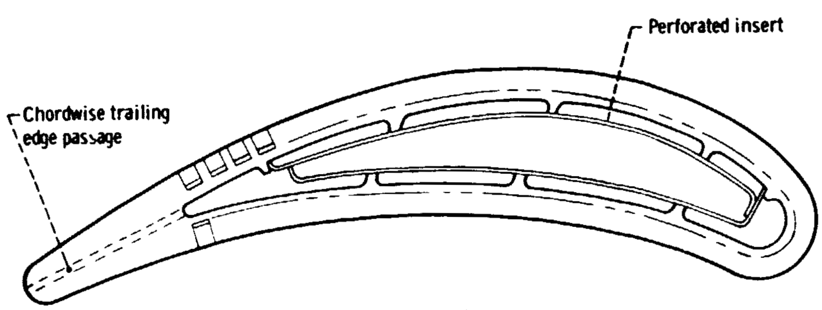


(b) Platelet detail.

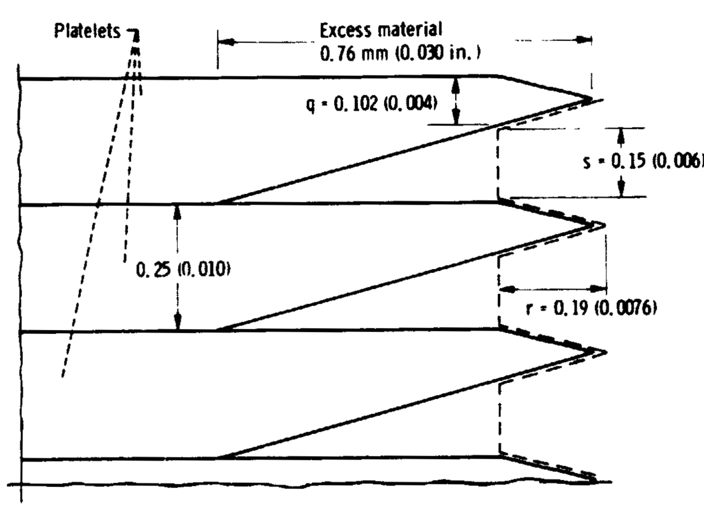
Figure 2. - Laminated; convection- and film-cooled turbine vane.



(a) Pore-plenum platelet.



(b) Slot platelet.



(c) Schematic of excess material on gas side surface of platelets. Fin representation indicated by dashed line. Scale, approximately 100:1. All dimensions are given in millimeters (in.).

Figure 3. - Cross-sectional schematic of laminated vane.

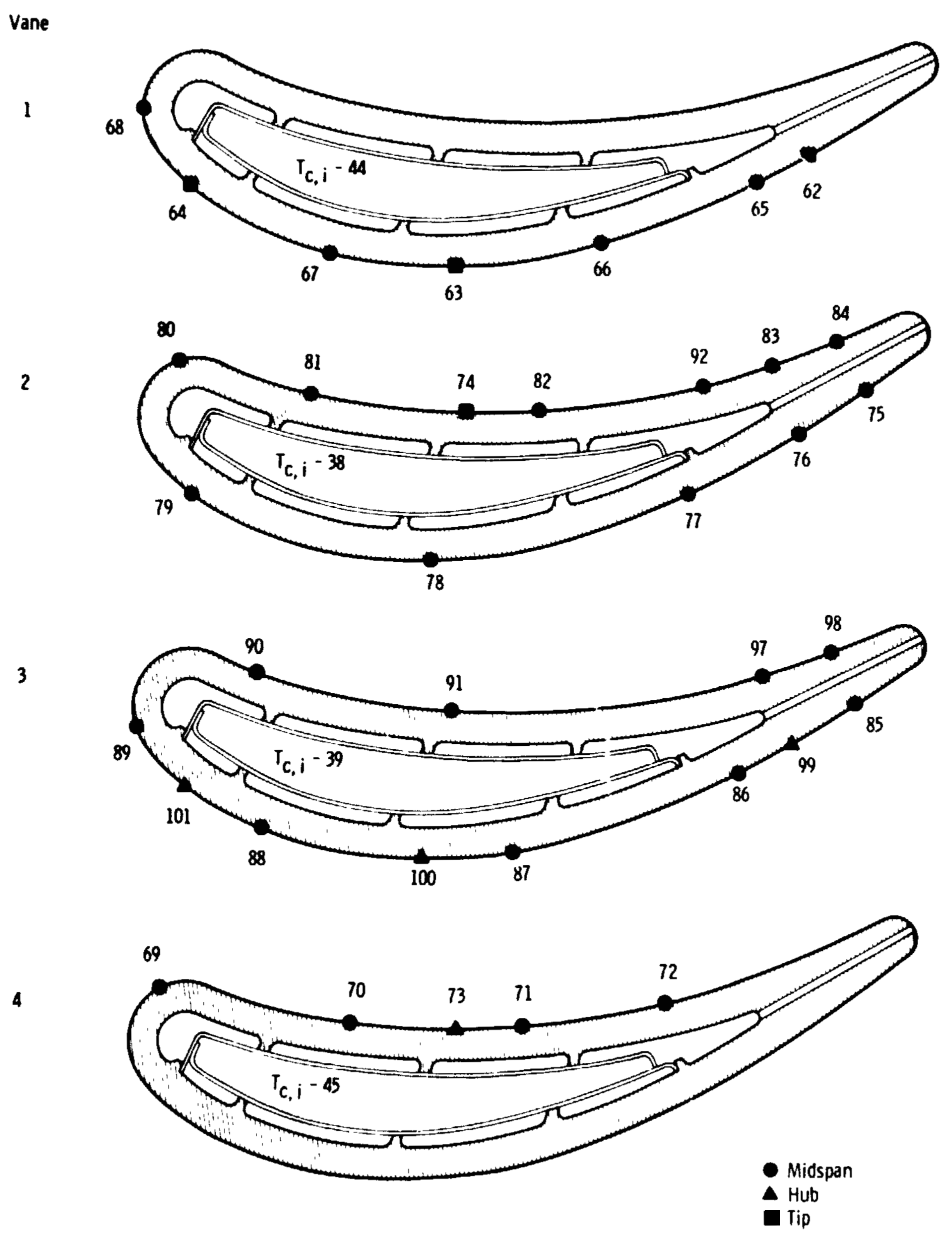


Figure 4. - Thermocouple locations on laminated convection and film-cooled turbine vanes.

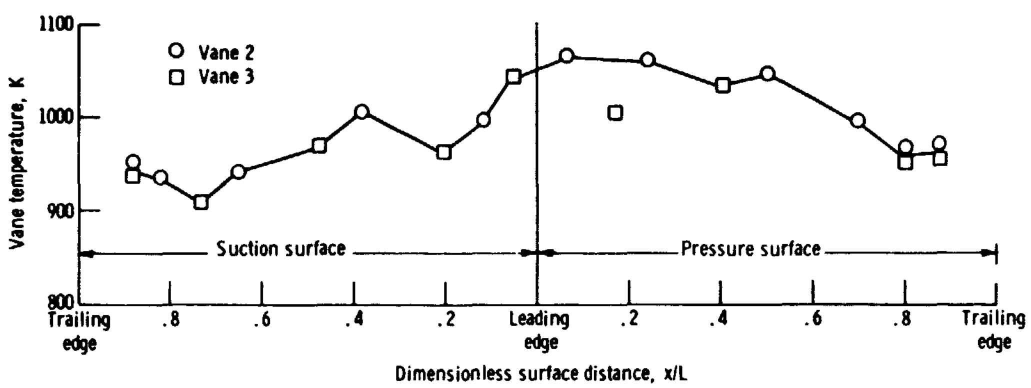
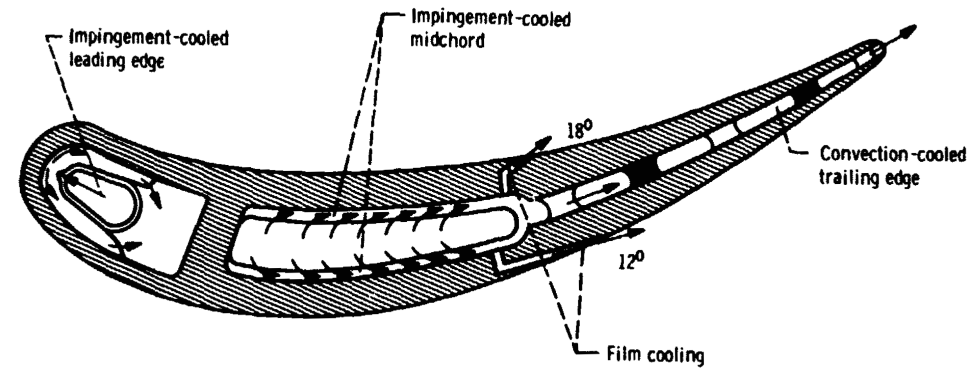
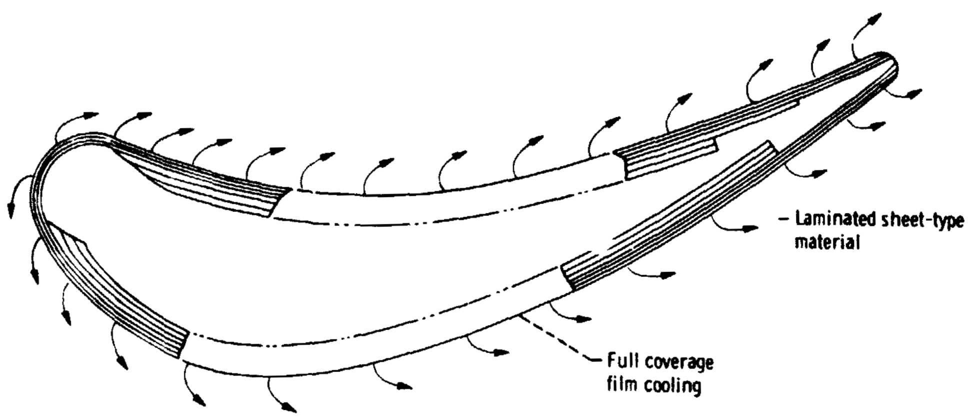


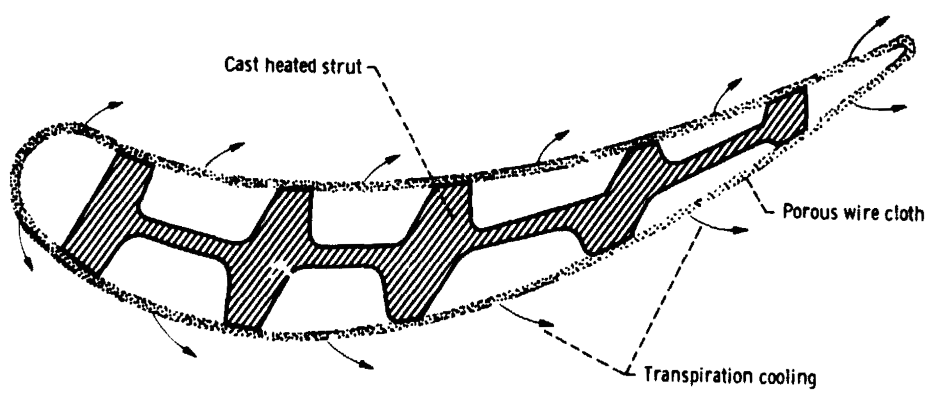
Figure 5. - Data set 1. Temperature distribution composed of temperatures from vanes 2 and 3. Gas inlet total temperature, 1255 K (1800° F); gas inlet total pressure, 22.7 newtons per square centimeter (33 psia); coolant inlet temperature, 701 K (802° F); coolant- to gas-flow ratio, 0.04575.



(a) Reference 7 vane.



(b) Reference 8 vane.



(c) Sintered wire cloth vane. CD-11325-33

Figure 6. - Reference vanes.

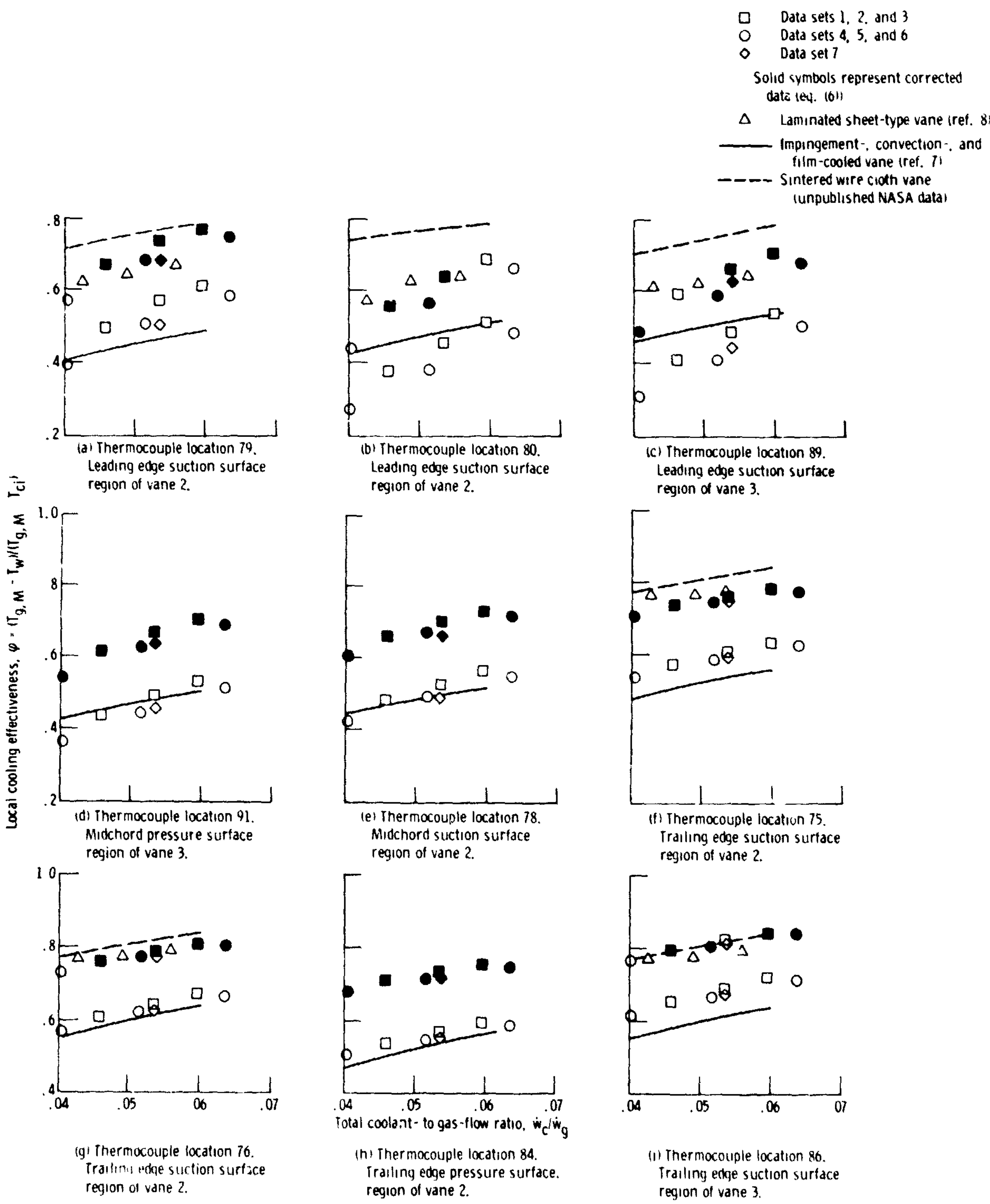


Figure 7. Local cooling effectiveness as function of nominal total coolant- to gas-flow ratio for selected locations of vanes 2 and 3.

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