NASA TECHNICAL MEMORANDUM

NASA TM X-3249

NASA TM X-3249

TURBINE VANE COOLANT FLOW VARIATIONS AND CALCULATED EFFECTS ON METAL TEMPERATURES

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NATIONAL AERONAUTICS AND SPACE ADMINISTRATION • WASHINGTON, D. C. • JUNE 1975

1. Report No.	2. Government Accession No.	3. Recipient's Catalog No.
NASA TM X-3249		
4. Title and Subtitle TURBINE VANE COOLANT FLOW VARIATIONS AND CALCULATED EFFECTS ON METAL TEMPERATURES		5. Report Date June 1975
		6. Performing Organization Code
7. Author(s) Frederick C. Yeh, Peter L. Meitner, and Louis M. Russell		8. Performing Organization Report No. E-8254
		10. Work Unit No. 505-04
9. Performing Organization Name and Address Lewis Research Center National Aeronautics and Space Administration Cleveland, Ohio 44135		11. Contract or Grant No.
		13. Type of Report and Period Covered Technical Memorandum
12. Sponsoring Agency Name and Address National Aeronautics and Space Administration Washington, D.C. 20546		14. Sponsoring Agency Code
15. Supplementary Notes		

16. Abstract

Seventy-two air-cooled turbine vanes were tested to determine coolant flow variations among the vanes. Calculations were made to estimate the effect of measured coolant flow variations on local vane metal temperatures. The calculations were based on the following assumed operating conditions: turbine inlet temperature, 1700 K (2600° F); turbine inlet pressure, 31 N/cm^2 (45 psia); coolant inlet temperature, 811 K (1000° F); and total coolant to gas flow ratio, 0.065. Variations of total coolant flow were not large (about 10 percent from the arithmetic mean) for all 72 vanes, but variations in local coolant flows were large. The local coolant flow variations ranged from 8 to 75 percent. Calculated metal temperature variations ranged from 8 to 60 K (15° to 108° F).

17. Key Words (Suggested by Author(s))	18. Distribution Statement
Cooled turbine vanes Flow distribution Heat transfer	Unclassified - unlimited STAR category 34 (rev.)

19. Security Classif. (of this report)	20. Security Classif. (of this page)	21. No. of Pages	22. Price*
Unclassified	Unclassified	15	$3.25

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

TURBINE VANE COOLANT FLOW VARIATIONS AND CALCULATED EFFECTS ON METAL TEMPERATURES

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SUMMARY

A set of 72 air-cooled turbine vanes for a research turbojet engine was tested to determine coolant flow variations among the vanes. Calculations were also made to estimate the effect of the measured coolant flow variations on the local vane metal temperatures. The calculations were based on the following assumed operating conditions: turbine inlet temperature, 1700 K (2600° F); turbine inlet pressure, 31 newtons per square centimeter, (45 psia); inlet coolant temperature, 811 K (1000° F); and total coolant to gas flow ratio, 0.065.

Although the variations of total flow were not large (about 10 percent from the arithmetic mean) for all 72 vanes, variations in local coolant flows were large. The local variations from the local arithmetic mean ranged from 8 to 75 percent, with the largest variations occurring at the film-cooling slots. Calculated local metal temperature variations associated with the variations in coolant flow ranged from 8 to 60 K (15° to 108° F).

Large variations in local coolant flow such as those measured in this investigation are undesirable in a set of turbine vanes for heat-transfer or life research programs or for commercial use; these variations emphasize the importance of fabrication, inspection, and quality control in the procurement of cooled turbine hardware.

INTRODUCTION

An experimental investigation was conducted to determine coolant flow variations among a set of 72 air-cooled turbine vanes for a research turbojet engine, and calculations were made to estimate the effect of these measured coolant flow variations on local vane metal temperatures for typical engine operating conditions.

Present day air-cooled turbine vanes and blades can have relatively complex internal passages. In many instances, the internal routing of the cooling air is such that the coolant is ejected from several different exit regions on the airfoil or in the platform assemblies. Because of the relatively small sizes of coolant passages within an airfoil, small variations in passage dimensions can result in relatively large variations in passage flow areas. Consequently, large variations in local coolant flow rates and in coolant flow distributions from design values may occur. These variations in coolant flow and distribution can affect the resulting airfoil temperature levels and temperature distributions under engine operating conditions. If the temperatures are markedly different from those anticipated in the original vane (or blade) design, the life of the component may be significantly different from that originally intended. For this reason, production blades and vanes are generally flow tested to determine their uniformity in total coolant flow, but checks of local flow uniformity are seldom made.

The purposes of this investigation were (1) to determine the magnitude of the variations in total and local coolant flow rates that may be expected in cooled turbine vanes which incorporate a combination of convection, impingement, and film cooling; (2) to determine if vanes that have comparable total coolant flow rates have significant variations in local coolant flow rates (such variations would indicate that total flow measurements alone may not be adequate); and (3) to calculate the local vane temperature variations associated with the local coolant flow variations to obtain an indication of the acceptability of the vanes for heat-transfer research or life testing.

Vane flow tests were conducted by using a bench-type flow apparatus supplied with room-temperature air. Both total flow measurements and local flow measurements at four coolant exit locations were made for each of the 72 vanes. The supply pressure at the cooling air inlet of each vane was maintained at 21.2 newtons per square centimeter (30.7 psia); the cooling air discharged to ambient room pressure. Vane metal temperature variations associated with measured coolant flow variations were calculated. Empirical temperature correlations based on existing engine data were used as the basis for calculating vane metal temperature variations. The calculations were performed for a maximum turbine inlet temperature of 1700 K (2600° F), a coolant inlet temperature of 811 K (1000° F), and an overall coolant to gas flow ratio of 0.065.

VANE DESCRIPTION

A cross-sectional view of the vane is shown in figure 1. The vane had a span of 9.8 centimeters (3.85 in.) and a chord of 6.3 centimeters (2.47 in.). Cooling air entered from a supply tube at the tip of the vane and flowed into a plenum chamber, where the flow divided into two parts. Part of the air flowed into a leading edge impingement tube which had 16 slots, 0.47 centimeter (0.185 in.) long and 0.020 to

0. 038 centimeter (0. 008 to 0. 015 in.) wide. The air flowed through these slots and impinged on the internal surface of the vane leading edge, where the surface area was increased by chordwise fins. After impinging on the leading edge surface, the air flowed between the fins into a leading edge exit passage and exhausted at the vane hub. The remainder of the airflow in the plenum chamber entered a midchord supply chamber. There were 481 holes on the suction side of this chamber and 334 holes on the pressure side. The holes were 0. 038 centimeter (0. 015 in.) in diameter. Flowing through these holes, the air impinged on the inside surfaces of the vane shell and flowed chordwise to the rear of the airfoil. Part of the air exited through film-cooling slots on the pressure and suction surfaces and the remainder through the split trailing edge. The pressure surface film-cooling slot was a continuous slot, 9. 27 by 0. 064 centimeter (3. 65 by 0. 025 in.), and was fed by 16 metering slots, 0. 25 by 0. 053 centimeter (0. 100 by 0. 021 in.), perpendicular to the pressure surface. The suction surface contained eight film-cooling slots, each 1. 11 by 0. 051 centimeter (0. 438 by 0. 020 in.). The suction surface film-cooling slots were fed by a total of 16 metering slots, each 0. 19 by 0. 051 centimeter (0. 075 by 0. 020 in.), perpendicular to the suction surface. The tapered trailing edge contained a staggered array of pin fins consisting of four rows of oblong pins and one row of round pins. The oblong pins were 0. 38 by 0. 25 centimeter (0. 15 by 0. 10 in.) and varied in height from 0. 18 to 0. 094 centimeter (0. 070 to 0. 037 in.). The round pins had a diameter of 0. 20 centimeter (0. 080 in.) and a height of 0. 064 centimeter (0. 025 in.).

This description of the vane geometry and figure 1 show that the coolant flow can be defined according to the four areas from which it exited:

(1) Leading edge flow, which exited from the vane hub at the leading edge region

(2) Trailing edge flow, which exited from the split trailing edge

(3) Pressure surface flow, which exited from the pressure surface film-cooling slot

(4) Suction surface flow, which exited from the suction surface film-cooling slots

VANE PROCUREMENT

The vanes tested in this investigation were procured according to an experimental rather than a production specification. Under the experimental specification greater freedom is allowed in the visual inspection to accept casting imperfections in various parts of the airfoil; greater freedom is allowed in the X-ray inspection to accept internal voids or shrinkage; and greater tolerances are allowed in the dimensional inspection as to the location of holes, wall thicknesses, and finished dimensions. Thus, vanes procured under this specification may differ in durability from vanes procured under a production specification; they do not differ in performance. In terms of coolant flow and heat-transfer performance, the vanes tested in this investigation were typical of production type hardware. The vanes were checked for coolant flow by the manufacturer, but not for local coolant flow distribution.

TEST APPARATUS AND INSTRUMENTATION

A bench-type flow apparatus, shown schematically in figure 2, was used to perform the flow tests. Laboratory service air at 86.2 newtons per square centimeter (125 psig) was filtered and passed through a pressure regulator before entering the test vane. Clamp sealing fixtures were used to block specific exit ports for different phases of the test. The clamp fixtures had rubber-faced jaws that were specifically fabricated to conform to the exit port contours. The jaws were positioned over the respective exit ports and clamped into place to provide leak-free sealing.

Air pressure measurements were made at the rotameter inlet and at the vane inlet. Air temperature was measured at the rotameter inlet. The rotameter used was a commercial model that was calibrated for air at densities corresponding to those encountered in the tests.

TEST PROCEDURE

Vane coolant flow rates through the four exit regions of the vane were measured. In order to obtain the exit flows in the most expeditious manner, four tests were performed for each of the 72 vanes in the following sequence: Cooling air at room temperature was admitted into each vane (1) with all exit ports open (pressure and suction surface film-cooling slots, leading edge exit, and trailing edge exit open); (2) with the pressure surface film-cooling slot blocked and the other exit ports open; (3) with the pressure and suction surface film-cooling slots blocked and the other exit ports open; and (4) with the pressure and suction surface film-cooling slots blocked, the leading edge exit blocked, and the trailing edge exit open. The air from the open ports was always discharged into the atmosphere. The pressure at the vane inlet was always maintained at 21.2 newtons per square centimeter (30.7 psia).

CALCULATION METHODS

Metal Temperatures

To determine the expected vane metal temperature variations due to coolant flow variations (from their arithmetic mean values) at the assumed engine operating conditions, two assumptions had to be made. First, the coolant flow distributions of each vane were assumed to be the same under engine operating conditions as they were in the bench tests. Second, the heat-transfer characteristics of all 72 vanes were taken to be those of a similar vane reported in reference 1. The average coolant flow distribution of the 72 vanes tested compared very favorably with the coolant flow distribution of the vane reported in reference 1. A comparison of these flows is shown in table I. The assumption of similar coolant flow distributions for both the bench tests and the engine operating conditions is reasonable, since the coolant flow through the vane is choked for both the bench tests and the assumed engine operating conditions. The ratios of coolant inlet pressure to ambient gas pressure for both the bench tests and the engine operating conditions were greater than 1. 1. The curves of figure 3 have zero slope for this condition and show that the coolant flow distribution was independent of the back pressure at the exit ports.

The expected metal temperature variations (from their arithmetic mean values) at engine operating conditions were calculated, under the previously described assumptions, in the following manner:

Reference 1 correlates the measured vane metal temperatures in terms of a temperature difference ratio φ_x, defined as

$$\varphi_x = \frac{T_g - T_w}{T_g - T_c} \tag{1}$$

where

T_g maximum local turbine inlet temperature

T_w local vane wall surface temperature

T_c coolant inlet temperature

This temperature difference ratio is expressed as a function of the local coolant to gas flow ratio and takes the form

$$\varphi_x = \frac{1}{1 + A\left(\frac{W_c}{W_g}\right)^B} \tag{2}$$

where

A, B experimentally derived constants

W_c local coolant flow past particular point on vane

W_g main stream gas flow per vane channel

From equations (1) and (2), the temperature difference between the "average" vane and each particular vane at the chosen engine operating conditions can be expressed as

$$T_{w,N} - T_{w,AV} = (T_g - T_c)\left[\frac{1}{1 + A\left(\dfrac{W_c}{W_g}\right)_{AV,E}^{B}} - \frac{1}{1 + A\left(\dfrac{W_c}{W_g}\right)_{N,E}^{B}}\right] \tag{3}$$

where

$T_{w,N}$	wall temperature of vane N of 72 vanes tested
$T_{w,AV}$	wall temperature of average of 72 vanes tested
$(W_c/W_g)_{AV,E}$	coolant to main stream gas flow ratio for average vane at engine operating conditions
$(W_c/W_g)_{N,E}$	coolant to main stream gas flow ratio for N^{th} vane at engine operating conditions

Also,

$$\left(\frac{W_c}{W_g}\right)_{N,E} = \left(\frac{W_c}{W_g}\right)_{AV,E} \frac{W_{c,N,F}}{W_{c,AV,F}} \tag{4}$$

where

$W_{c,N,F}$	coolant flow rate of N^{th} vane in reported flow tests
$W_{c,AV,F}$	coolant flow rate of average vane in reported flow tests

The following engine operating conditions were assumed: maximum local turbine inlet temperature, 1700 K (2600° F); turbine inlet pressure, 31 newtons per square centimeter (45 psia); coolant inlet temperature, 811 K (1000° F); and coolant to main stream gas flow ratio, 0.065.

The temperature calculations were performed for the following selected locations on the vane surface (fig. 4), where values of φ_x were known from measurements reported in reference 1:

(1) Leading edge stagnation point
(2) Trailing edge suction surface
(3) Midchord suction surface
(4) Midchord pressure surface

Internal Flow Distribution

Figure 4 shows that thermocouples 3 and 4 were located just upstream of the suction and pressure surface film-cooling slots, respectively. The local coolant flows past these locations could not be obtained directly from the flow tests, because the coolant flow past these locations was the sum of the coolant flows through the film-cooling slots and some fraction of the flow which was discharging through the trailing edge. For the purposes of this report, it was assumed that the portions of the coolant flow which discharged through the trailing edge from the suction and pressure side passages were directly proportional to the numbers of impingement holes in the suction and pressure sides of the midchord supply chamber, respectively (481 holes on the suction side and 334 on the pressure side, each 0.038 cm (0.015 in.) in diameter). For the described flow tests, the local coolant flow rates past thermocouples 3 and 4 were thus obtained by adding the respective film-cooling slot flow to the proper fraction of the trailing edge flow (481/815 for the suction side and 334/815 for the pressure side).

DISCUSSION OF RESULTS

The results of the bench flow tests of 72 turbine vanes are summarized in table II. The table lists the total flow rates through the vanes, as well as the local flow rates through the leading edge, the pressure surface film-cooling slot, the suction surface film-cooling slots, and the trailing edge. It is obvious by inspection of table II that large variations in local coolant flow rates occurred among the vanes tested, while variations in total flow were relatively small. The coolant flows through the suction and pressure surface film-cooling slots showed the greatest variations. The variations in coolant flow to the leading edge and the trailing edge and in total coolant flow were less, but nevertheless significant. The range of flow rates, the arithmetic mean values for the 72 vanes, and the deviations from the arithmetic mean are listed in table III.

The large coolant flow variations through the pressure and suction surface film-cooling slots were partly caused by inconsistencies in the slot widths, which, while within the allowable dimensional tolerances, caused significant slot area variations. Another possible reason for the large coolant flow variations was shifts in the vane inserts (midchord supply chamber walls) from their design position. Such a shift was observed after the vane with the largest flow variations was sectioned for visual inspection. Repeated X-ray attempts to detect the insert shift prior to sectioning had been unsuccessful.

The maximum variations in the calculated vane metal temperatures, corresponding to the observed maximum flow variations, are presented in table IV. At the vane leading edge, the maximum temperature variations from the mean were +9 and -8 K (+16° and -15° F). This small temperature variation was consistent with the small observed flow variations. Temperature variations at other points of the vane were much higher. At the trailing edge, the variations from the mean were +60 and -36 K (+108° and -65° F). At the suction and pressure surface film-cooling slots, the variations were +24 and -18 K (+44° and -32° F) and +54 and -25 K (+97° and -45° F), respectively. The greatest temperature variations occurred at the trailing edge, although the coolant flow variations at this point were less than the variations at either the suction or the pressure surface film-cooling slot. Apparently, for this vane design, the trailing edge temperatures were additionally influenced by the film-cooling flows exiting upstream of the trailing edge.

Large variations in coolant flow and metal temperatures from vane to vane, such as those calculated in this report, may be acceptable in special applications where the vanes are subjected only to moderate gas temperatures or can be overcooled to maintain a low nominal wall temperature. However, for research vanes used to develop methods of predicting heat-transfer performance and vane life expectancy, and for vanes in commercial use, these large flow variations are not desirable. For research purposes, it is desirable to keep variations in local wall temperature between vanes (or blades) to within 5.6 K (10° F). For the set of vanes tested this criterion could not be met if a group of three or more vanes was needed for a test program.

Although the results presented in this report are limited to one batch of vanes of one particular configuration, they do provide an indication of the need for tighter tolerances, better quality control, and better inspection in the procurement of both production and research blades and vanes. Specification of acceptable local coolant flows is desirable in the procurement of vane or blade hardware because measurements of total flows alone are probably inadequate. Methods are needed to detect nondestructively changes in interior structure such as shifts in vane inserts, which in this investigation could be detected only by visual inspection after a blade was sectioned. The tighter tolerance and quality control requirements are particularly important with increased turbine inlet pressures and temperatures, at which the metal temperatures are more sensitive to flow variations than at the test engine conditions used in this investigation.

SUMMARY OF RESULTS

The following results were obtained from the measurement of the total and local coolant flows for 72 air-cooled turbine vanes and from the calculation of the metal temperature variations due to the flow variations:

1. Although the variations in total coolant flow through the vanes about an arithmetic mean were not large (about 10 percent), the local coolant flow variations were large (8 to 74 percent). The large local coolant flow variations were attributed partly to allowable dimensional tolerances and partly to a shift in the internal vane insert in some vanes.

2. The largest coolant flow variations occurred at the film-cooling slots.

3. The calculated metal temperature variations due to flow variations were large (8 to 60 K (15^{o} to 108^{o} F)) at the assumed turbine operating conditions.

4. There were no groups of vanes greater than two that would have local temperature variations between vanes no greater than 5.6 K (10^{o} F), a level which would be desirable for research vanes.

5. The large measured local variations in coolant flow and the associated variations in predicted vane metal temperatures emphasized the importance of fabrication and inspection quality control in the procurement of cooled turbine hardware.

6. Specification of acceptable local coolant flows would be desirable in the procurement of blades and vanes since specification of only total flow can be inadequate.

Lewis Research Center,
National Aeronautics and Space Administration,
and
U.S. Army Air Mobility R&D Laboratory,
Cleveland, Ohio, March 20, 1975,
505-04.

REFERENCE

1. Yeh, Frederick C.; Gladden, Herbert J.; and Gauntner, James W.: Comparison of Heat Transfer Characteristics of Three Cooling Configurations for Air-Cooled Turbine Vanes Tested in a Turbojet Engine. NASA TM X-2580, 1972.

TABLE I. - COMPARISON OF MEASURED COOLANT

FLOW DISTRIBUTIONS

Flow exit port	Average of 72 vanes	Data from ref. 1
	Coolant flow, percent of vane total	
Leading edge	26.2	28.0
Trailing edge	48.7	47.5
Suction surface film-cooling slots	12.7	11.0
Pressure surface film-cooling slot	12.4	13.5

TABLE II. - COOLANT FLOW RATES FOR 72 AIR-COOLED TURBINE VANES

[Based on bench test data.]

Vane	Leading edge flow, kg/sec	Trailing edge flow, kg/sec	Suction surface flow, kg/sec	Pressure surface flow, kg/sec	Total flow, kg/sec	Vane	Leading edge flow, kg/sec	Trailing edge flow, kg/sec	Suction surface flow, kg/sec	Pressure surface flow, kg/sec	Total flow, kg/sec
1	0.00601	0.01050	0.00274	0.00416	0.02341	37	0.00617	0.01015	0.00285	0.00169	0.02087
2	.00628	.01033	.00375	.00324	.02360	38	.00578	.01202	.00207	.00284	.02271
3	.00556	.01367	.00118	.00071	.02111	39	.00592	.01097	.00357	.00330	.02376
4	.00564	.01190	.00262	.00282	.02297	40	.00606	.01021	.00191	.00346	.02165
5	.00641	.01161	.00084	.00202	.02089	41	.00588	.01243	.00248	.00139	.02218
6	.00572	.01044	.00242	.00191	.02049	42	.00565	.01191	.00359	.00373	.02488
7	.00574	.01063	.00321	.00360	.02319	43	.00604	.01005	.00259	.00290	.02159
8	.00551	.01133	.00296	.00184	.02164	44	.00613	.01031	.00222	.00256	.02123
9	.00576	.01065	.00327	.00350	.02318	45	.00592	.01151	.00270	.00313	.02325
10	.00602	.01039	.00289	.00340	.02271	46	.00601	.01031	.00342	.00282	.02257
11	.00593	.01103	.00243	.00278	.02217	47	.00595	.01190	.00285	.00159	.02230
12	.00572	.01197	.00315	.00292	.02376	48	.00598	.01053	.00267	.00367	.02284
13	.00593	.01059	.00242	.00311	.02205	49	.00612	.01063	.00338	.00329	.02342
14	.00633	.01032	.00304	.00251	.02219	50	.00545	.01065	.00355	.00194	.02158
15	.00612	.01049	.00262	.00400	.02324	51	.00627	.01043	.00345	.00278	.02293
16	.00622	.01013	.00276	.00349	.02260	52	.00587	.01108	.00231	.00326	.02252
17	.00590	.01077	.00235	.00242	.02144	53	.00573	.01271	.00302	.00203	.02348
18	.00577	.01152	.00317	.00294	.02340	54	.00594	.01176	.00195	.00272	.02237
19	.00611	.00981	.00332	.00382	.02305	55	.00623	.00776	.00309	.00334	.02043
20	.00589	.01160	.00297	.00296	.02341	56	.00583	.00780	.00388	.00322	.02073
21	.00592	.01021	.00317	.00376	.02306	57	.00607	.01054	.00255	.00326	.02242
22	.00590	.01029	.00381	.00237	.02237	58	.00593	.01192	.00282	.00261	.02327
23	.00623	.01002	.00394	.00349	.02368	59	.00621	.01037	.00242	.00179	.02080
24	.00547	.01245	.00298	.00250	.02340	60	.00606	.00932	.00236	.00272	.02047
25	.00575	.01056	.00261	.00339	.02231	61	.00617	.01054	.00267	.00077	.02015
26	.00558	.01165	.00329	.00274	.02327	62	.00619	.00773	.00312	.00366	.02070
27	.00547	.01220	.00259	.00363	.02389	63	.00583	.01243	.00327	.00155	.02307
28	.00587	.01155	.00324	.00316	.02382	64	.00616	.01057	.00201	.00182	.02056
29	.00544	.01211	.00376	.00238	.02369	65	.00579	.01213	.00306	.00350	.02447
30	.00574	.01194	.00272	.00384	.02424	66	.00596	.01139	.00347	.00296	.02378
31	.00598	.01194	.00202	.00373	.02368	67	.00594	.01178	.00333	.00238	.02343
32	.00569	.01105	.00276	.00294	.02244	68	.00573	.01160	.00352	.00328	.02413
33	.00590	.01239	.00347	.00240	.02417	69	.00622	.00989	.00391	.00329	.02331
34	.00576	.01169	.00220	.00225	.02190	70	.00581	.01208	.00233	.00237	.02259
35	.00585	.01202	.00325	.00358	.02470	71	.00632	.01043	.00271	.00141	.02086
36	.00584	.01202	.00324	.00215	.02325	72	.00614	.01070	.00287	.00268	.02239

TABLE III. - VARIATIONS IN FLOW RATE IN 72 VANES

Flow exit port	Flow range		Arithmetic mean		Maximum flow variation from mean, percent	
	kg/sec	lbm/sec	kg/sec	lbm/sec	Positive	Negative
Leading edge	0.00544-0.00641	0.01200-0.01410	0.00592	0.01306	8.2	8.1
Trailing edge	0.0077 -0.0137	0.0170 -0.0302	0.0110	0.0242	24.6	30.0
Suction surface film-cooling slots	0.00084-0.00391	0.00185-0.00862	0.00288	0.00634	35.9	70.8
Pressure surface film-cooling slot	0.00071-0.00416	0.00157-0.00917	0.00281	0.00619	48.2	74.7
Total flow	0.0201 -0.0249	0.0433 -0.0549	0.0226	0.0498	10.1	11.1

TABLE IV. - MAXIMUM VARIATIONS IN CALCULATED

VANE METAL TEMPERATURES

Thermocouple location on vane surface	Maximum temperature variation from mean				Maximum total temperature variation	
	Positive		Negative			
	K	^{o}F	K	^{o}F	K	^{o}F
Leading edge	9	16	8	15	17	31
Trailing edge	60	108	36	65	96	173
Midchord suction surface	24	44	18	32	42	76
Midchord pressure surface	54	97	25	45	79	142

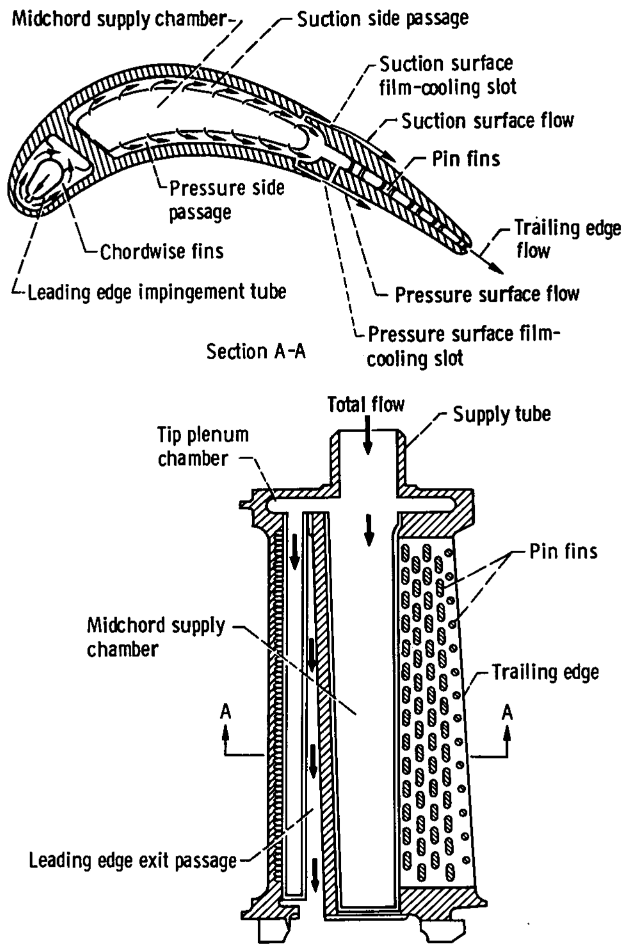


Figure 1. - Vane internal flow configuration.

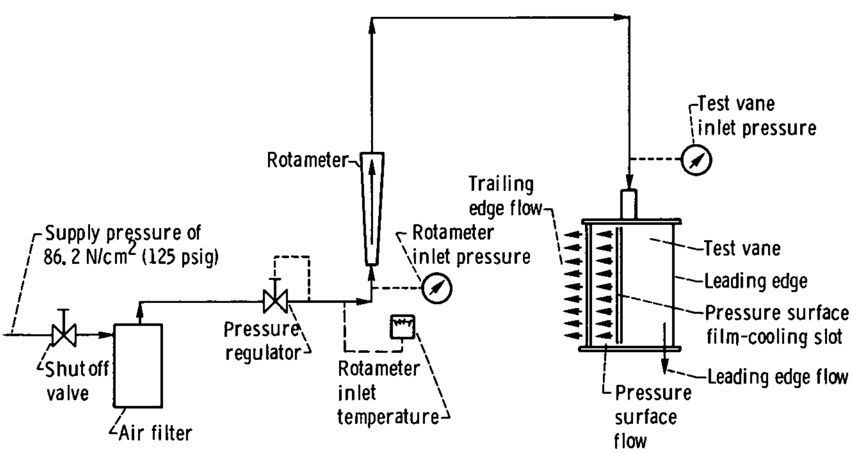


Figure 2. - Flow test apparatus.

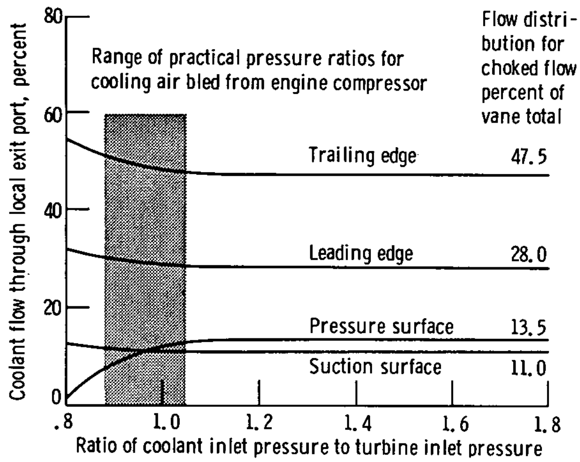
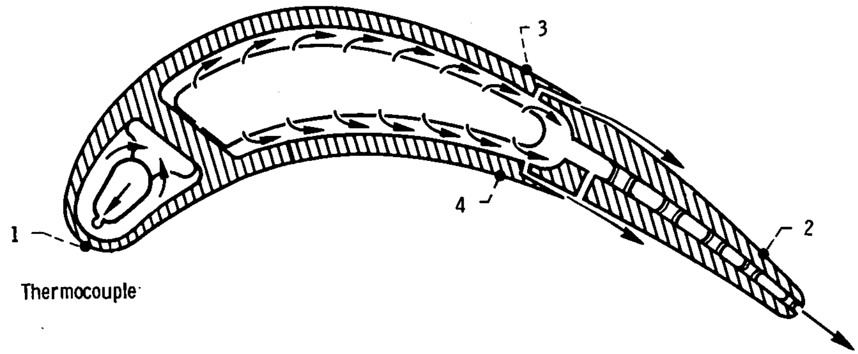


Figure 3. - Vane coolant flow distribution among exit ports as function of ratio of vane inlet coolant pressure to turbine inlet pressure (from ref. 1).



Thermo-couple	Thermocouple location	Constants in eq. (2)	
		A	B
1	Leading edge	0.1796	-0.48
2	Trailing edge	.0580	-.76
3	Suction surface	.0602	-.60
4	Pressure surface	.0544	-.67

Figure 4. - Vane midspan locations where local metal temperatures were calculated and correlation constants (constants from ref. 1).

NASA-Langley, 1975 E-8254

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