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**COLD-AIR INVESTIGATION OF A TURBINE  
WITH TRANSPiration-COOLED STATOR BLADES**

**IV - Stage Performance With Wire-Mesh  
Shell Stator Blading**

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16. Abstract  Turbine performance characteristics were obtained with a single-stage axial-flow turbine which had stator blades employing transpiration coolant ejection through a wire-mesh shell. The turbine was tested over a range of speed and pressure ratio and with a nominal coolant fraction of 0.031. Additional tests were made at the design speed, in which the coolant flow was varied from 0 to 0.07 of the primary flow. The results were compared to similar results obtained from turbines using stator trailing-edge coolant ejection and a discrete hole transpiration-cooled stator. A base turbine with no cooling provisions was used as a standard for comparisons.			
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# COLD-AIR INVESTIGATION OF A TURBINE WITH TRANSPERSION-COOLED STATOR BLADES

## IV - STAGE PERFORMANCE WITH WIRE-MESH SHELL STATOR BLADING

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### SUMMARY

A cold-air experimental investigation was conducted on a 30.0-inch (0.762-m) single-stage research turbine to determine the effects on the turbine aerodynamic performance of coolant flow ejection through wire-mesh shell stator blades. These effects were determined over a range of coolant fraction (ratio of coolant flow to primary flow) from 0 to 0.07.

The performance of this turbine was compared to turbines having discrete hole transpiration-cooled stator blades, to slotted-trailing-edge coolant ejection stator blades, and also to a base reference turbine which had no provision for stator coolant flow. The stator blades for all four turbines were of the same profile geometry. All four stator configurations were tested with the same rotor.

When tested over a range of speed and pressure ratio and for a nominally constant coolant fraction of 0.031, the turbine with the wire-mesh transpiration-cooled stator blades yielded a primary-air efficiency of 0.900 at the equivalent design speed and an equivalent specific turbine work output of 17.00 Btu per pound (39 572 J/kg). The corresponding pressure ratio was 1.779. At this speed and work output, the primary efficiency curve was essentially flat with increasing coolant fractions from 0 to 0.07, the maximum tested. The thermodynamic efficiency, however, decreased significantly with increased coolant, mainly because the turbine was charged with the available energy of the coolant, and also because of a degradation in the performance of both the stator and rotor components. For example, at the same work output and speed as mentioned previously, and at a coolant fraction of 0.07, the thermodynamic efficiency of the turbine with wire-mesh transpiration-cooled stator blades was 0.10 less than at a coolant fraction of zero. In contrast, the thermodynamic efficiency with trailing-edge coolant ejection remained essentially unchanged. It was concluded that for the turbines tested with the two types of transpiration-cooled stator blades, the coolant flow did very little, if any, useful work. This conclusion contrasts markedly with observations made previously when the turbine with slotted trailing-edge-cooled stator blades was tested.

## INTRODUCTION

The performance requirements of gas turbine engines currently being considered for some types of advanced aircraft necessitate high turbine inlet temperatures. These high temperatures require turbine blade cooling. The effect of blade cooling on the aerodynamic performance of the turbine must be clearly understood in order to be factored into the turbine design and engine performance assessment. Little experimental information is available as to the effect on turbine performance caused by coolant mixing with the main gas stream. Accordingly, NASA Lewis Research Center has been conducting a series of tests on two methods of cooling. These are trailing-edge coolant ejection stator blades and transpiration-cooled stator blades. The transpiration-cooled stator blades are further divided into two types: discrete hole transpiration-cooled stator blades and wire-mesh transpiration-cooled stator blades.

To incorporate these cooling schemes, a 30-inch (0.762-m) cold-air research turbine was designed with physical features suitable for blade cooling. The blading for this type of turbine is characterized by thick profiles and blunt leading and trailing edges. The design procedure, together with coordinates for the stator and rotor profiles, is presented in reference 1. The stator was tested as a separate component; a detailed loss determination is presented in reference 2, with an added discussion in reference 3. The performance of the uncooled base turbine is reported in reference 4. The performance thus obtained is therefore used as a basis of comparison for the turbines with cooled stator configurations.

The stator blades of the base turbine were then modified to incorporate a coolant-air ejection slot along the trailing edge of each stator blade. The coolant air was ejected into the main stream along the entire length of the blade and in the same general direction as the primary air. Tests were conducted to determine the effect of this type of coolant ejection on turbine performance. The results of these tests, which are reported in references 5 to 7, show that neither the modification of the trailing edge nor the coolant ejection had much effect on either stator or turbine thermodynamic efficiency.

The second cooled configuration was fabricated using transpiration-type cooled stator blades. These blades were of the self-supporting shell type. The cooling air was discharged through discrete holes around the blade surface in a direction normal to the primary flow. Reference 8 shows that when this stator was tested separately, the performance decreased significantly (1) when compared to the same stator with the discrete holes filled and (2) when compared to the stator with blades containing trailing-edge slots.

This loss is also reflected in the stage tests reported in reference 9.

The blades of the third cooled stator configuration were also of the transpiration type, but made of a wire-mesh shell supported by a central core. The performance of this stator was reported in reference 10. Its performance was, in general, comparable to that found for the stator with discrete hole transpiration blades.

The investigation reported herein was conducted to experimentally determine the effect on turbine stage performance of the wire-mesh transpiration-cooled stator blades. Performance parameters in terms of mass flow, torque, and efficiency were obtained over a range of speed and pressure ratio for a coolant fraction (ratio of coolant flow to primary flow) of 0.031. The effect of varying the coolant fraction over a limited range of pressure ratio at design speed was also determined. The coolant fraction investigated ranged from 0 to 0.07. These results are compared to those obtained for the turbines equipped with the base design stator blades, the slotted-trailing-edge stator blades, and the discrete hole transpiration-cooled stator blades.

All turbine tests were conducted with the primary air maintained at a constant inlet stagnation pressure of 30 inches of mercury absolute ( $10.16 \text{ N/cm}^2$ ). Both the primary and coolant air were supplied by the laboratory combustion air system at a nominal temperature of  $543^\circ \text{R}$  (303 K). Therefore, no heat-exchange effects between primary and coolant air occur, as would be the case in a hot turbine.

## SYMBOLS

A	annular flow area, $\text{ft}^2$ ; $\text{m}^2$
g	force-mass conversion constant, $32.174 \text{ ft/sec}^2$
h	specific enthalpy, $\text{Btu/lb}$ , $\text{J/kg}$
J	mechanical equivalent of heat, $778.16 \text{ ft-lb/Btu}$
N	rotational speed, rpm
p	absolute pressure, $\text{lb/ft}^2$ ; $\text{N/m}^2$
R	gas constant, $53.34 \text{ ft-lb/(lb)}(^\circ\text{R})$ ; $287 \text{ J/(kg)(K)}$
T	temperature, ${}^\circ\text{R}$ ; K
$U_m$	blade velocity at mean height, $\text{ft/sec}$ ; $\text{m/sec}$

V	absolute gas velocity, ft/sec; m/sec
w	mass-flow rate, lb/sec; kg/sec
$\alpha$	absolute flow angle (measured from axial), positive in direction of rotor rotation, deg
$\gamma$	ratio of specific heats
$\delta$	ratio of turbine-inlet total pressure to U.S. standard sea-level pressure of 2116.22 lb/ft <sup>2</sup> abs (1459 N/cm <sup>2</sup> )
$\eta$	efficiency based on total-pressure ratio
$\sqrt{\theta_{cr}}$	ratio of critical velocity at turbine inlet to critical velocity ( $V_{cr} = 1019.1$ ft/sec (310.62 m/sec)) at U.S. standard sea-level air temperature of 518.7° R (288.17 K)
$\tau$	torque, ft-lb; N-m

Subscripts:

c	coolant flow
cr	conditions at Mach 1
h	hub radius
id	ideal
p	primary flow
r	rotor
t	tip radius
th	thermodynamic
0	measuring station at turbine inlet (see fig. 6)
1	measuring station at stator outlet
2	measuring station at rotor outlet

Superscript:

\* total state

## APPARATUS AND INSTRUMENTATION

The turbine used in this investigation was a 30-inch (0.762-m) tip diameter, single-stage, axial-flow turbine of the same aerodynamic design as the base turbine reported in reference 1. The design requirements of the base turbine are summarized as follows:

Equivalent specific work output, $\Delta h/\theta_{cr}$ , Btu/lb (J/kg) . . . . .	17.00 (39 572)
Equivalent mean blade speed, $U_m/\sqrt{\theta_{cr}}$ , ft/sec (m/sec) . . . . .	500 (152.4)
Equivalent mass flow, $w\sqrt{\theta_{cr}}/\delta$ , lb/sec (kg/sec) . . . . .	39.9 (18.098)

The design procedure used to evolve the blade shapes and a sketch showing the blade passage and profiles and the blading coordinates can also be found in reference 1.

For the subject investigation, the stator was replaced with a stator having blades of the same profile, but with a wire-mesh shell material supported by a central core, and suitable for transpiration cooling (fig. 1). The supporting core structure is shown in

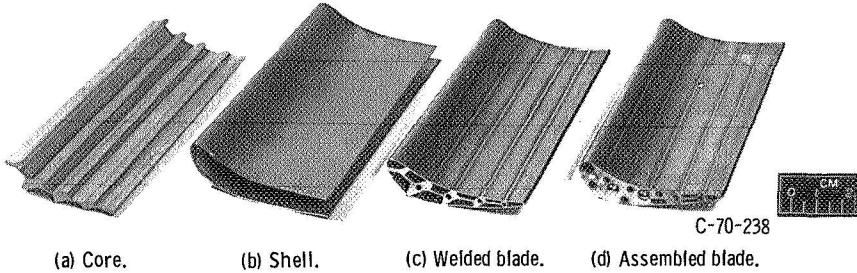


Figure 1. - Wire-mesh shell blade parts and assembly.

figure 1(a). The shell (fig. 1(b)) is stainless-steel wire mesh and is fabricated to yield a given constant porosity. In figure 1(c) the shell is shown welded to the core structure. A finished blade with the end cap and its metering orifices to the different compartments is shown in figure 1(d).

During fabrication, deviations from the desired blade profile as specified in reference 1 occur. One deviation is the valleys formed by the compactations at the electron beam weldments. Another deviation is the variation from desired surface curvature caused by unavoidable stretching of the porous shell material between the core ribs during the welding operation.

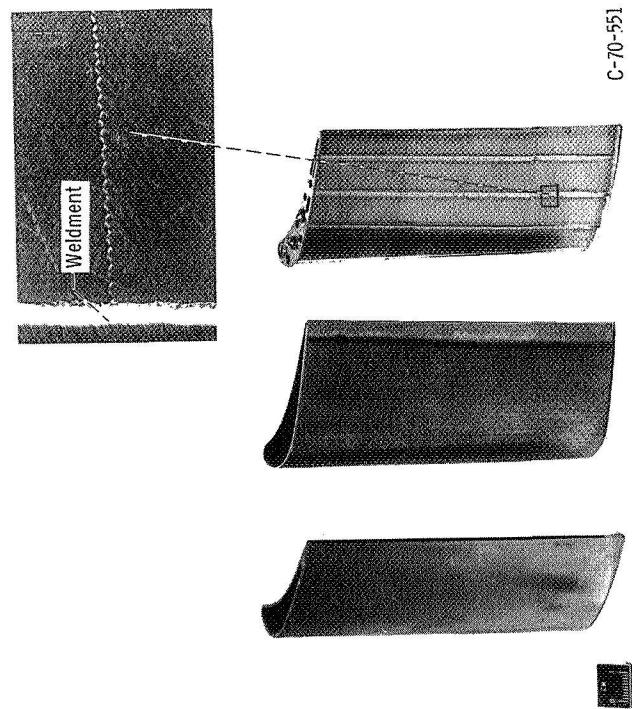


Figure 2. - Comparison of stator blades with three cooling provisions.

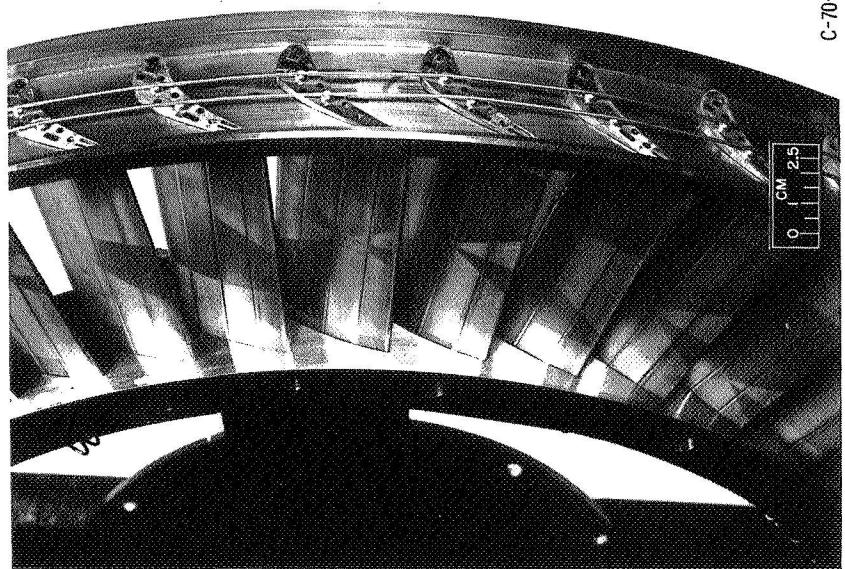


Figure 3. - Stator assembly of wire-mesh transpiration-cooled blades.

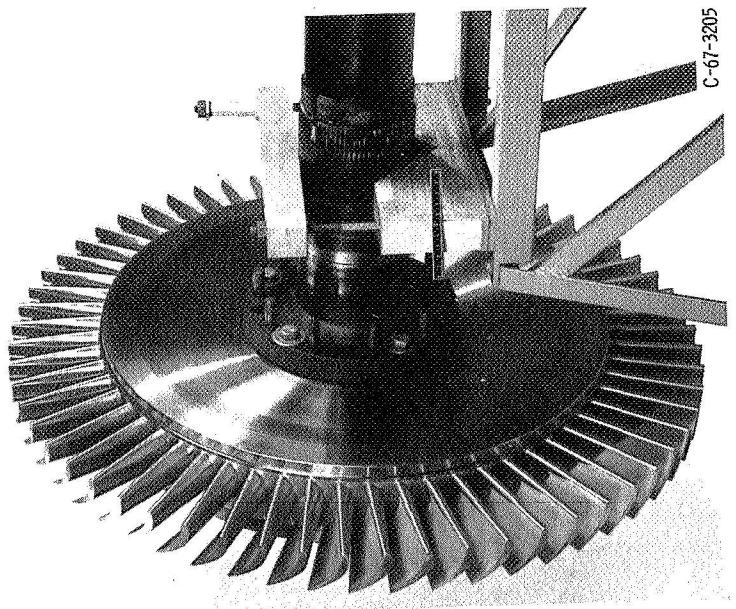


Figure 4. - Turbine rotor assembly.

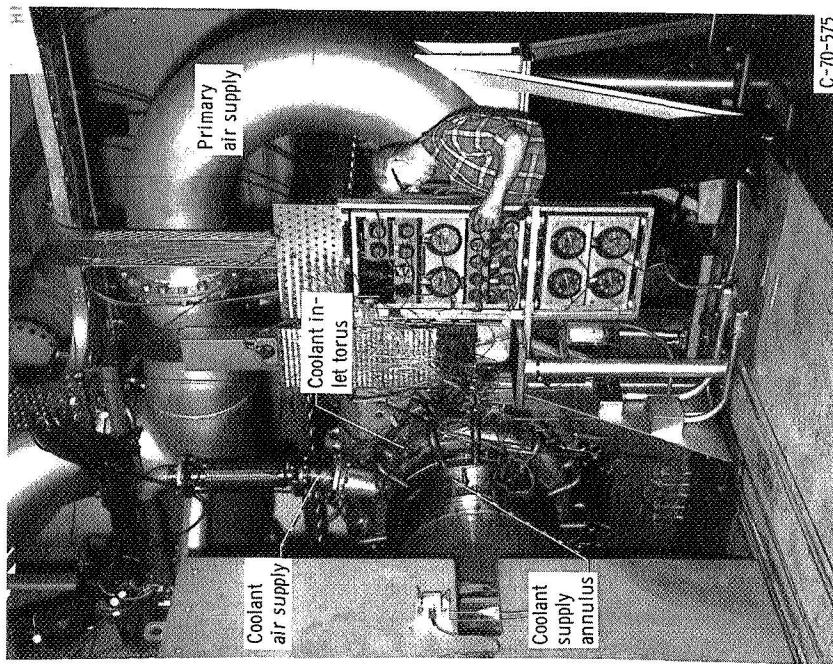


Figure 5. - Test facility.

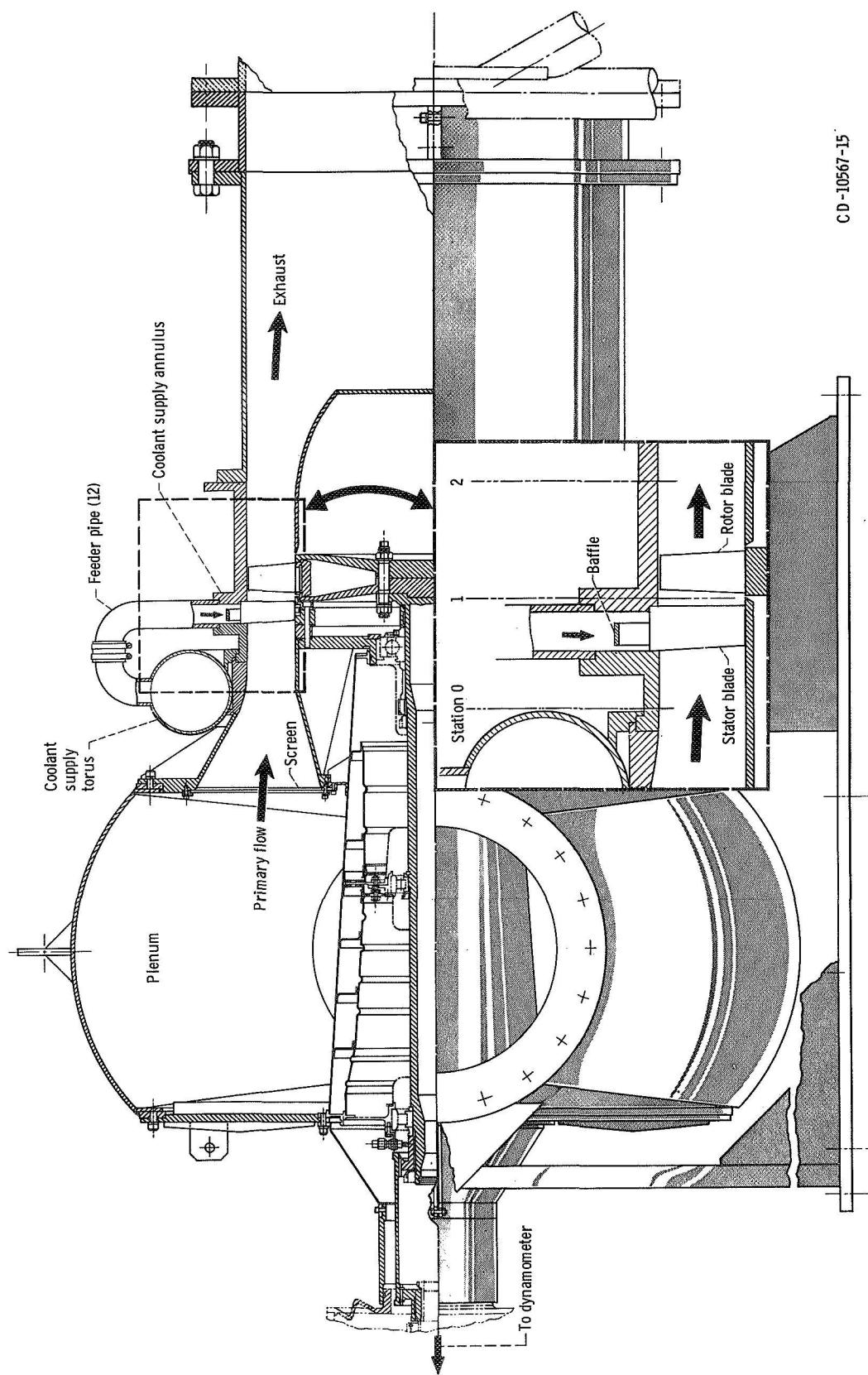


Figure 6. - Turbine test section with measuring stations.

Figure 2 shows the three cooled-stator-blade configurations. Figure 2(a) is the slotted-trailing-edge blade, which is a modification of the hollow-base (uncooled) stator blade and is discussed in references 5 to 7. In figure 2(b) is the discrete hole transpiration-cooled stator blade which is discussed in references 8 and 9. Figure 2(c) shows a typical wire-mesh transpiration-cooled stator blade with an enlarged view of the wire mesh and a typical weldment. A closeup view of the subject wire-mesh transpiration-cooled stator assembly is shown in figure 3. This stator was installed in the test facility along with the base rotor (fig. 4). The test facility was the same as that described in detail in reference 4, and modified to incorporate the stator cooling-air system as reported in reference 5. The test facility is shown in figure 5.

The instrumentation was the same as that described in reference 9, in which comparable tests were conducted on the turbine having discrete hole transpiration-cooled stator blades. Briefly, total and static pressures and temperatures were measured at the turbine inlet. Static pressures were measured at the stator exit. At the turbine exit, total and static pressures were measured, along with the outlet flow angles. The measuring stations are shown in figure 6. In addition, turbine rotative speed was measured with an electronic counter in conjunction with a magnetic pickup and a sprocket secured to the rotor shaft. Turbine torque was measured, using a strain-gage-type load cell. The load cell and the readout system were calibrated before and after each day's testing.

All instrumentation was connected to a 100-channel data acquisition system which measured and recorded the electrical signals from the appropriate transducers. At each data point, five readings of each transducer were recorded and subsequently numerically averaged.

## PROCEDURE

The performance tests for the turbine with wire-mesh transpiration-cooled stator blades were conducted in two phases. In the first phase, the performance data were taken over a range of overall pressure ratio and speed. The speed was varied from 60 to 110 percent of equivalent design speed (4407 rpm) in increments of 10 percent. At a given speed, the pressure ratio was varied from approximately 1.4 to pressure ratios into the choked flow regions. It was desired to approximate the coolant fraction which resulted from similar turbine tests with the discrete hole transpiration-cooled stator

blades (ref. 9). In that reference investigation, a coolant-air supply annulus pressure of 31.0 inches of Hg absolute ( $10.5 \text{ N/cm}^2$ ) was maintained. Due to the porosity differences between the two types of transpiration-cooled stator blades, it was necessary to maintain the coolant-air supply annulus pressure of the wire-mesh stator blades at  $36.0 \pm 0.5$  inches of Hg absolute ( $12.2 \pm 0.17 \text{ N/cm}^2$ ). This resulted in a nominally constant coolant fraction of 0.031.

In the second phase, performance data were taken at design speed and over a range of coolant fraction and pressure ratio bracketing a primary specific work output of 17.00 Btu per pound ( $39,572 \text{ J/kg}$ ),  $\Delta h = (\tau N \pi / 30J) / w_p$ . The coolant mass-flow rate was varied by regulating the coolant-air supply annulus pressure. The range of coolant fraction investigated was from 0 to 0.07. This coolant fraction variation resulted in blade cavity pressures both below and above the turbine-inlet pressure.

In both phases, the turbine-inlet primary-air total-state conditions were 30.0 inches of Hg absolute ( $10.16 \text{ N/cm}^2$ ) and approximately  $545^\circ \text{ R}$  ( $303 \text{ K}$ ). Overall pressure ratios were set by adjusting the turbine-exit pressure.

Turbine performance was based on total-pressure ratio. The inlet total pressure was calculated (as in refs. 4 and 7) from the static pressure, primary mass flow, annulus area, and total temperature by using the following equation:

$$\frac{p'_0}{p_0} = \left[ \frac{1}{2} + \sqrt{\frac{1}{4} + \frac{\gamma - 1}{2g\gamma} \left( \frac{w_p}{p_0 A_0} \right)^2 RT'_0} \right]^{\gamma/(\gamma-1)} \quad (1)$$

The outlet total pressure was calculated (as in refs. 4 and 7) by using static pressure, combined primary and coolant mass flows, annulus area, area-averaged turbine-exit flow angle, and total temperature, as follows:

$$\frac{p'_2}{p_2} = \left[ \frac{1}{2} + \sqrt{\frac{1}{4} + \frac{\gamma - 1}{2g\gamma} \left( \frac{w_p + w_c}{p_2 A_2} \right)^2 \frac{RT'_2}{\cos^2 \alpha_2}} \right]^{\gamma/(\gamma-1)} \quad (2)$$

The total temperature used in equation (2) was derived by using the inlet total temperature, torque output, total mass flow, and speed data.

Two efficiencies are used in this report: the primary efficiency  $\eta_p$  and the thermodynamic efficiency  $\eta_{th}$ . In equation form,

$$\eta_p = \frac{w_p \Delta h_p + w_c \Delta h_c}{w_p \Delta h_{id,p}} = \frac{\frac{\tau N \pi}{30J}}{w_p \Delta h_{id,p}} \quad (3)$$

$$\eta_{th} = \frac{w_p \Delta h_p + w_c \Delta h_c}{w_p \Delta h_{id,p} + w_c \Delta h_{id,c}} = \frac{\frac{\tau N \pi}{30J}}{w_p \Delta h_{id,p} + w_c \Delta h_{id,c}} \quad (4)$$

The primary efficiency relates the total power of both fluids to the ideal power of only the primary flow, whereas the thermodynamic efficiency accounts for the ideal energies of both fluids.

## RESULTS AND DISCUSSION

The results of the cold-air experimental investigation on the subject single-stage turbine equipped with wire-mesh transpiration-cooled stator blades are discussed in three parts. First, the overall performance of the turbine, operated with a nominal stator coolant fraction of 0.031 over a range of speed and pressure ratio, is discussed. Second, the effect of varying coolant fraction on turbine performance is discussed. These tests were made only at the design speed, and at a turbine primary specific-work output of 17.00 Btu per pound (39 572 J/kg) corresponding to equivalent design work of the base turbine. A range of coolant fractions from 0 to 0.07 was investigated. Third, the results are compared with the results obtained previously (1) for the turbine with discrete hole transpiration-cooled stator blades (ref. 9), (2) for the turbine with slotted-trailing-edge stator blades (ref. 7), and (3) for the turbine with base (uncooled) stator blades (ref. 4). Differences in performance, as affected by stator configuration, are tabulated.

## Turbine Performance with a Nominal Stator Coolant Fraction of 0.031

The effects of varying pressure ratio and speed on turbine aerodynamic performance are shown for a range of pressure ratios from 1.4 to a ratio where the mass flow was choked and for speeds from 0.60 to 1.10 of design.

Overall turbine performance. - The overall performance of the turbine equipped with the wire-mesh transpiration-cooled stator blades is presented in figure 7. Equiva-

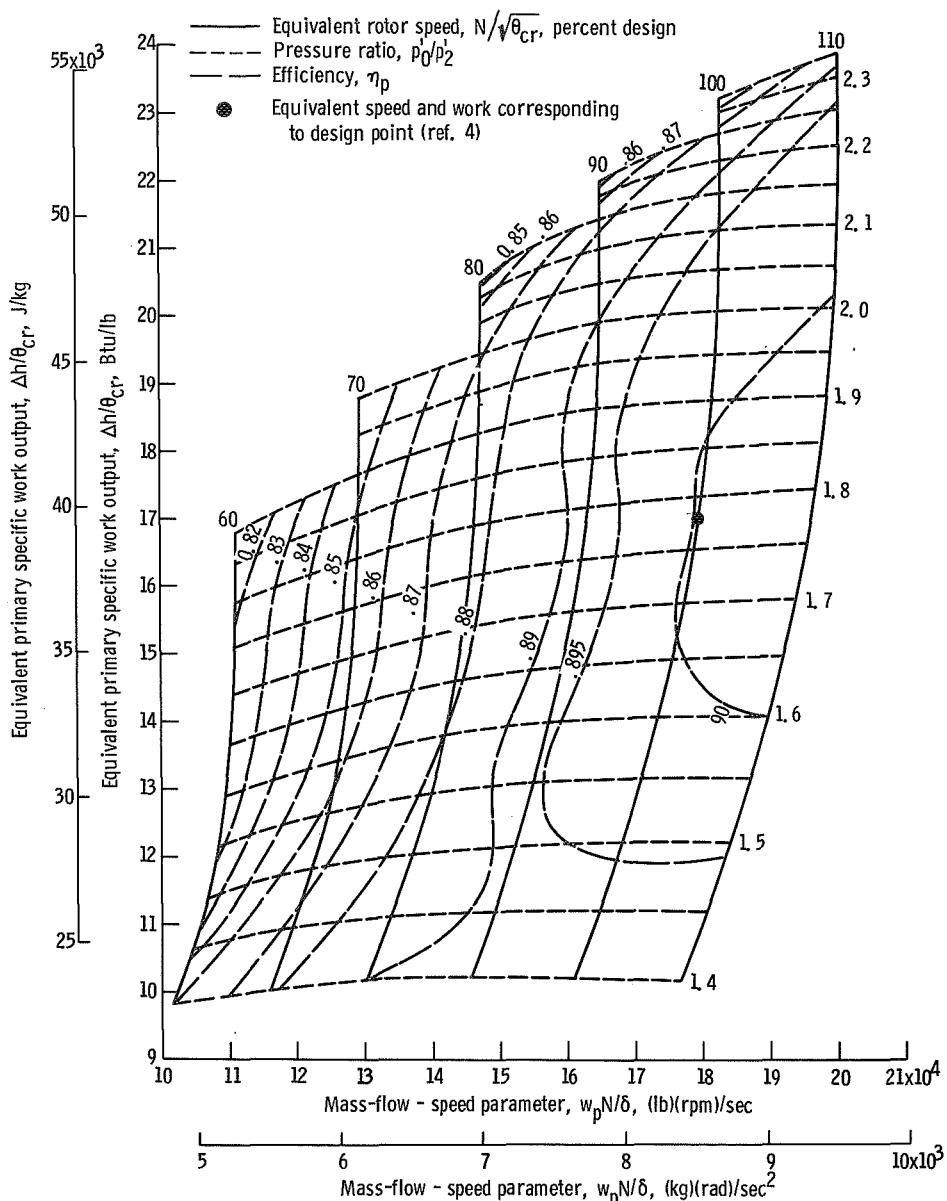


Figure 7. - Overall performance map. Turbine inlet pressure, 30.0 inches of Hg abs (10,159 N/cm<sup>2</sup>); inlet temperature, 545° R (303 K); stator coolant fraction, nominally 0.031.

TABLE I. - TURBINE PERFORMANCE PARAMETERS

[All data obtained at equivalent design rotor speed (4407 rpm) and at an equivalent specific work output of 17.00 Btu/lb (39 572 J/kg).]

Stator configuration	Equivalent primary-air mass flow		Coolant fraction, $w_c/w_p$	Total-pressure ratio, $p'_0/p'_2$	Outlet flow angle, $\alpha_2$ , deg	Coolant supply pressure		Primary air efficiency	Thermo-dynamic efficiency
	lb/sec	kg/sec				in. Hg abs	N/cm <sup>2</sup>		
Wire-mesh transpiration	40.70	18.46	0.031	1.779	-17.5	36.0±0.5	12.19±0.17	0.900	0.867
Discrete hole transpiration (ref. 9)	40.14	18.21	.0326	1.771	-15.3	31.0	10.50	.907	.876
Trailing-edge ejection (ref. 7)	39.72	18.02	.0468	1.713	-15.5	30.0	10.15	.958	.917
Design (un-cooled) (ref. 4)	40.64	18.43	-----	1.751	-15.2	-----	-----	.923	.923

lent primary-air specific work output  $\Delta h/\theta_{cr}$  is shown as a function of the mass-flow - speed parameter  $w_p N/\delta$  for the various rotor speeds  $N/\sqrt{\theta_{cr}}$ . Lines of constant total-pressure ratio  $p'_0/p'_2$  and contours of primary efficiency  $\eta_p$  are also included in the figure. The primary efficiency is based on the total-pressure ratio across the turbine. The coolant fraction over the range of speed and pressure ratio covered by the map was nominally 0.031.

Referring to figure 7, at design speed and a specific work output of 17.00 Btu per pound (39 572 J/kg), the efficiency was 0.900 and the pressure ratio was 1.779. These and other pertinent turbine performance parameters for the subject turbine and three reference turbines are listed in table I. In general, the efficiency level for this turbine was high over the range of variables tested. Except for pressure ratios above 2.18, the entire design speed curve shows efficiencies above 0.89.

Torque and mass flow. - The variation of equivalent torque  $\tau/\delta$  and of equivalent mass flow  $w_p \sqrt{\theta_{cr}}/\delta$  with total-pressure ratio for the equivalent speeds investigated is presented in figures 8 and 9, respectively. Data from the faired curves of these two figures were read at selected increments of pressure ratio and these values were used to calculate the performance map shown in figure 7.

The torque curves (fig. 8) show that, for the range of conditions tested, torque continually increased with increasing pressure ratio for all speeds investigated. For any constant pressure ratio, torque increased with decreasing speed. Limiting blade loading was not encountered over the range of speeds and pressure ratios investigated.

The curves of primary mass flow (fig. 9) show increasing flow with pressure ratio for all speeds investigated, until choked (constant) values were obtained. These choked mass-flow values decreased with increasing rotor speeds, indicating the stator was un-choked, and the rotor was limiting the flow. At design speed, the choking mass flow was 41.46 pounds per second (18.81 kg/sec), obtained at pressure ratios above 2.2. At the pressure ratio of 1.779, where a specific work output of 17.00 Btu per pound (39 572 J/kg) was obtained, the primary mass flow was 40.70 pounds per second (18.46 kg/sec).

Turbine-outlet flow angle. - The average turbine-outlet flow angle is shown in figure 10 as a function of pressure ratio and speed. These angle data are the numerical average of readings taken at radial positions corresponding to the area center radii of five equal annular areas. The average outlet angle indicated at design speed and the pressure ratio of 1.779 was  $-17.5^\circ$ . This compares to the mean-section design outlet flow angle of  $-17.8^\circ$  (ref. 4).

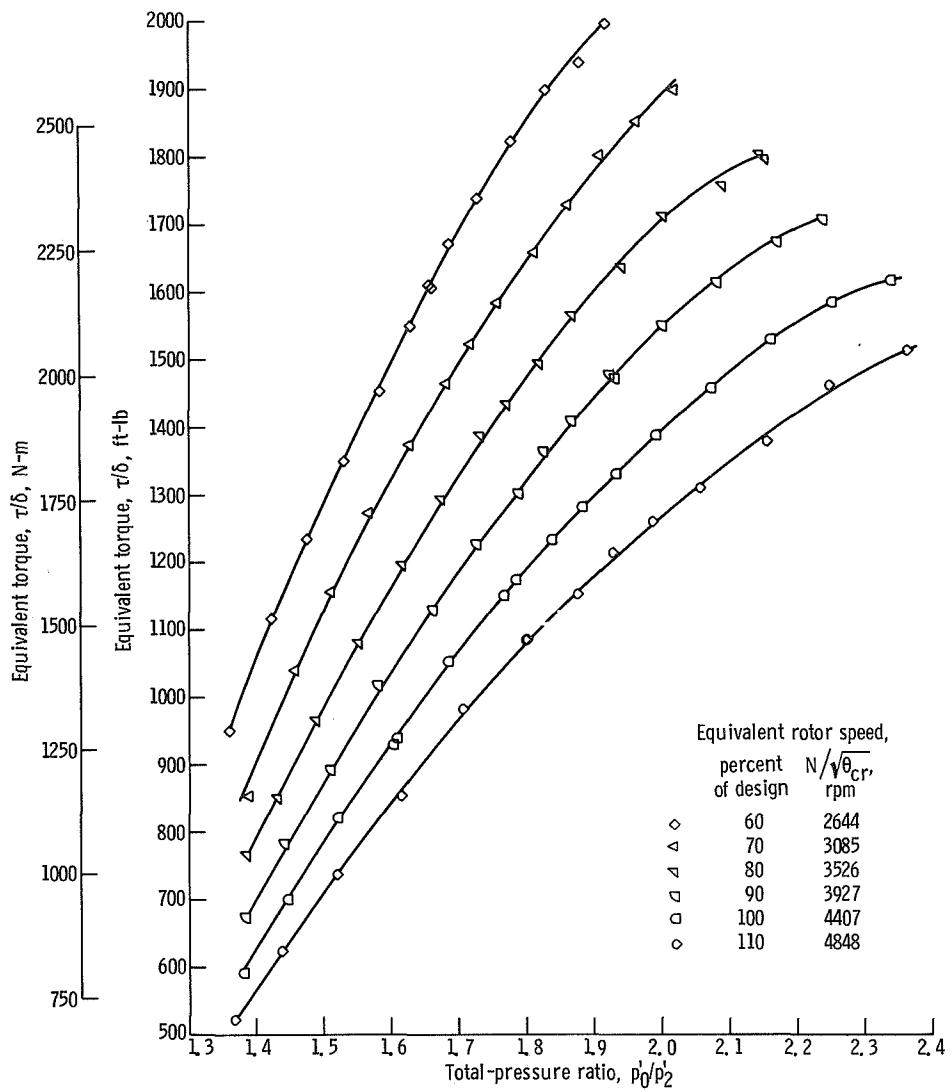


Figure 8. - Variation of equivalent torque with total-pressure ratio and equivalent rotor speed. Stator coolant fraction, nominally 0.031.

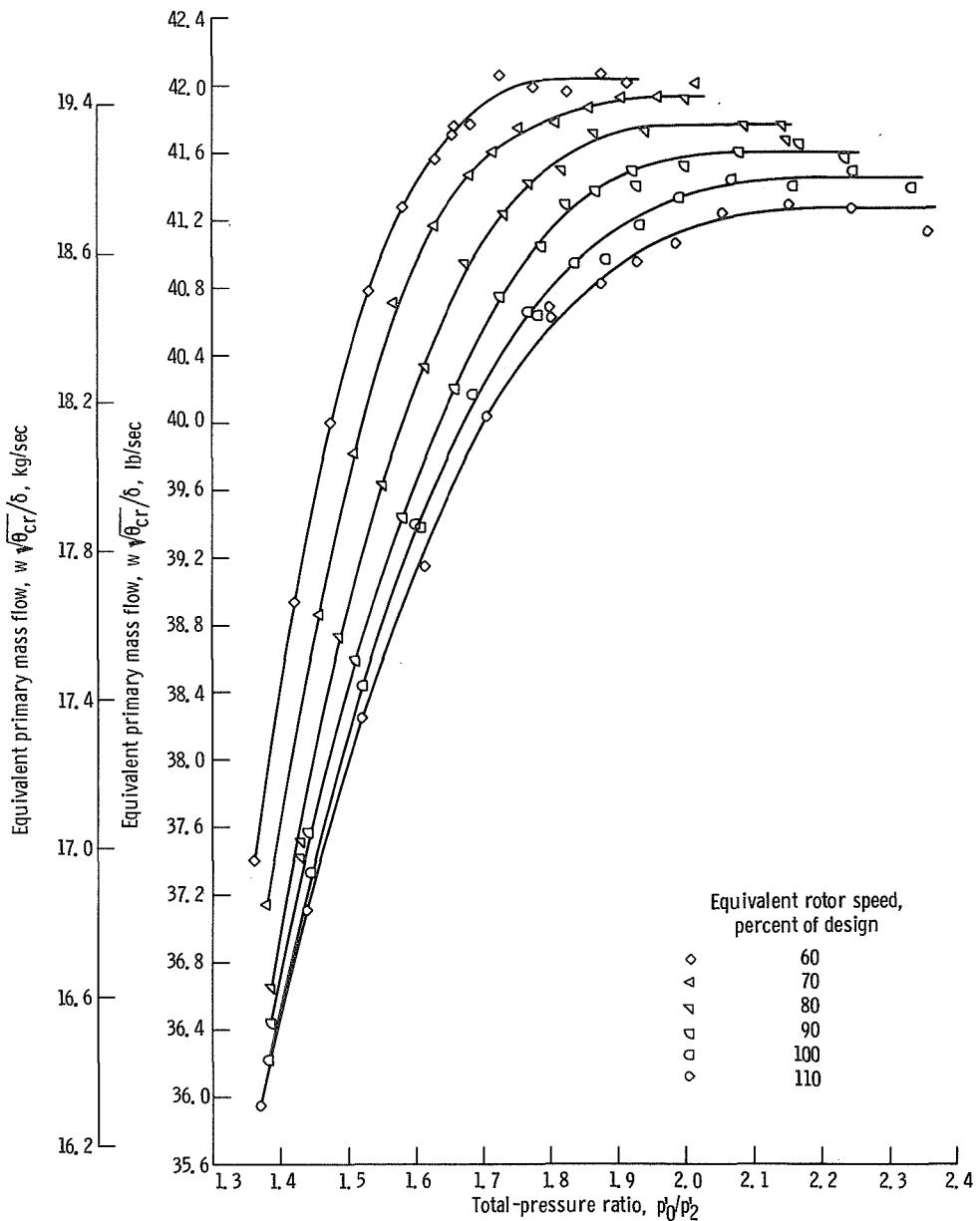


Figure 9. - Variation of equivalent mass flow with total-pressure ratio and equivalent rotor speed.  
Stator coolant fraction, nominally 0.031.

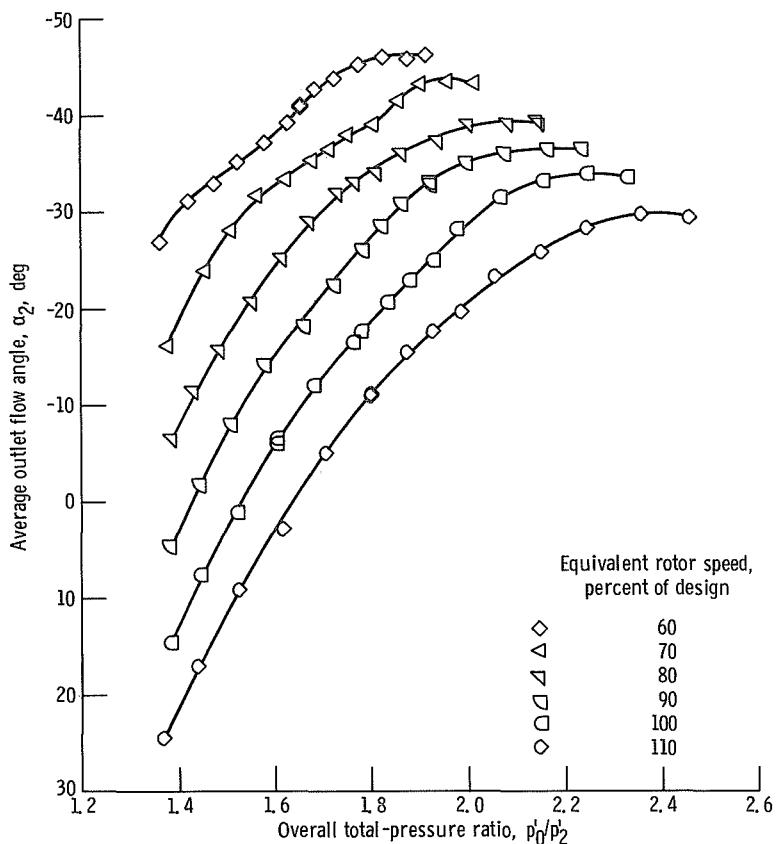


Figure 10. - Variation of outlet flow angle with total-pressure ratio and equivalent rotor speed. Stator coolant fraction, nominally 0.031.

Stator-outlet static pressure. - The static pressure at the hub and tip at the stator exit (measuring station 1, fig. 6) is shown in figure 11. These data were obtained at design speed and over a range of overall total-pressure ratio. Also shown are results from crossplots of data obtained during the stator component tests of reference 9. These crossplots were made for a comparable coolant fraction of 0.031. Very good agreement between the two tests is noted. This indicates that the rotor had only a small effect on the stator-outlet static-pressure gradient.

Coolant fraction. - As stated in the section PROCEDURE, the performance data were obtained with a nominal coolant fraction of 0.031. This coolant flow is shown in figure 12 as a function of pressure ratio and for the range of speeds investigated. At pressure ratios greater than about 1.6 the coolant fraction was constant at a value of 0.031.

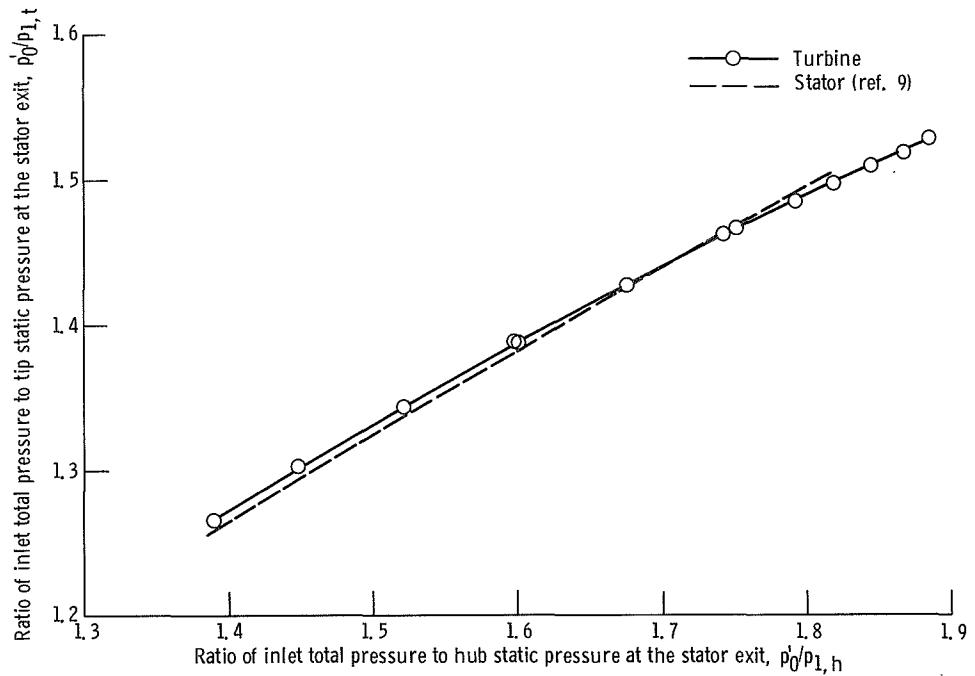


Figure 11. - Static-pressure variation of the stator exit for a range of turbine total-pressure ratio, with and without rotor in place. Rotor speed, 4407 rpm (equivalent design); stator coolant fraction, 0.031.

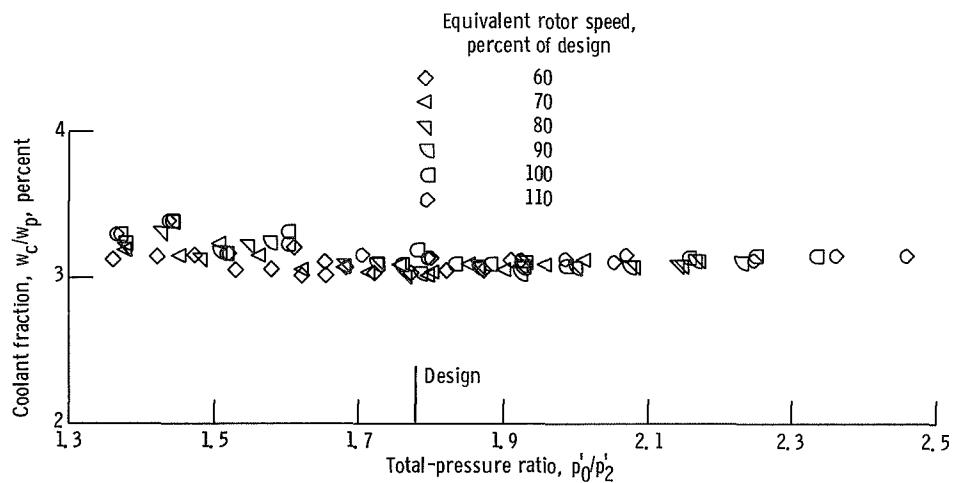


Figure 12. - Variation of stator coolant fraction with total-pressure ratio and equivalent rotor speed.

## Effect of Coolant Flow Variation on Turbine Performance

The effects of coolant-air flow on turbine aerodynamic performance are shown for a nominal primary specific-work output of 17.00 Btu per pound (39 572 J/kg) at the design speed and for a range of coolant fraction from 0 to 0.07.

Equivalent values of torque and mass flow as functions of pressure ratio were derived from the experimental data for selected coolant flows. From these values, the work output, primary efficiency, and thermodynamic efficiency were determined.

Torque and mass flow. - Variations of equivalent torque, primary flow, total flow, and primary work as functions of coolant fraction are shown in figure 13. The curves

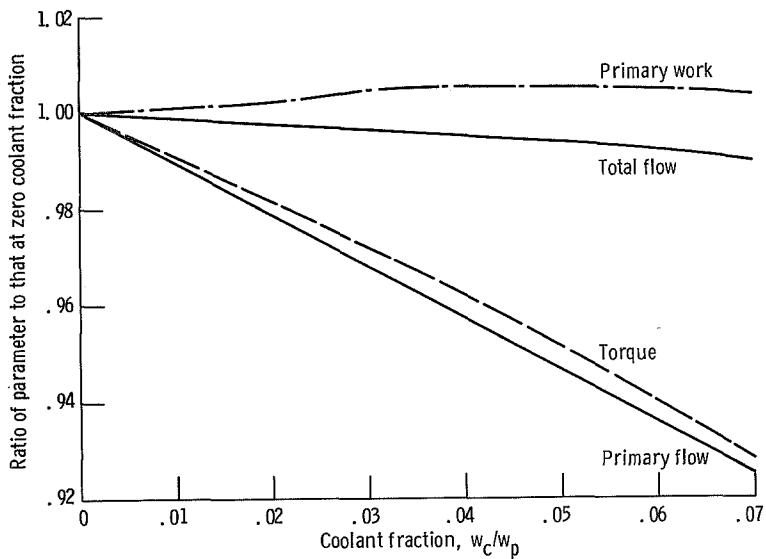


Figure 13. - Variation of pertinent performance parameters with coolant fraction at a constant pressure ratio of 1.793 (corresponding to 17.00 Btu per pound (39 572 J/kg) at zero coolant flow) and equivalent design rotor speed.

shown are for design equivalent speed and a constant total overall pressure ratio of 1.793, which corresponds to a primary specific work output of 17.00 Btu per pound (39 572 J/kg) at zero coolant fraction. All quantities on the figure were normalized to their respective values at zero coolant fraction. Figure 13 also shows that the sum of the primary flow and the coolant flow (total flow) remained relatively constant over the range of coolant fractions investigated. This is in general agreement with the results reported in reference 9. A maximum reduction of 1 percent in total flow occurred at a coolant fraction of 0.07. The reduction in primary flow and torque is seen to be in al-

most direct proportion to the coolant fractions. For example, at a coolant fraction of 0.07 the primary flow was 0.075 percent less than the primary flow at zero coolant fraction. The corresponding value of torque was down 0.072. Dividing torque by primary mass flow results in the primary work output  $\Delta h$ , which is shown in figure 13 as a function of coolant fraction. An increase in primary work of about 1/2 percent over that at zero coolant fraction occurred at a coolant fraction of 0.04. Between coolant fractions of 0.04 and 0.07 the primary work decreased slightly. At 0.07 coolant fraction the primary work was about 0.3 percent greater than the primary work at zero coolant fraction.

Turbine efficiencies. - Variation of primary efficiency and thermodynamic efficiency as a function of coolant fraction for the wire-mesh transpiration-cooled stator blading is shown in figure 14. The thermodynamic efficiency is a measure of the loss characteristics when considering the ideal energies of both the primary and coolant flows. Both efficiencies were calculated for a primary specific work output of 17.00 Btu per pound (39 572 J/kg). Both efficiencies are normalized to a zero-coolant-fraction efficiency of 0.895. The primary efficiency increased to a maximum of 1.007 at a coolant fraction of 0.045 and then decreased slightly to 1.005 at a coolant fraction of 0.07.

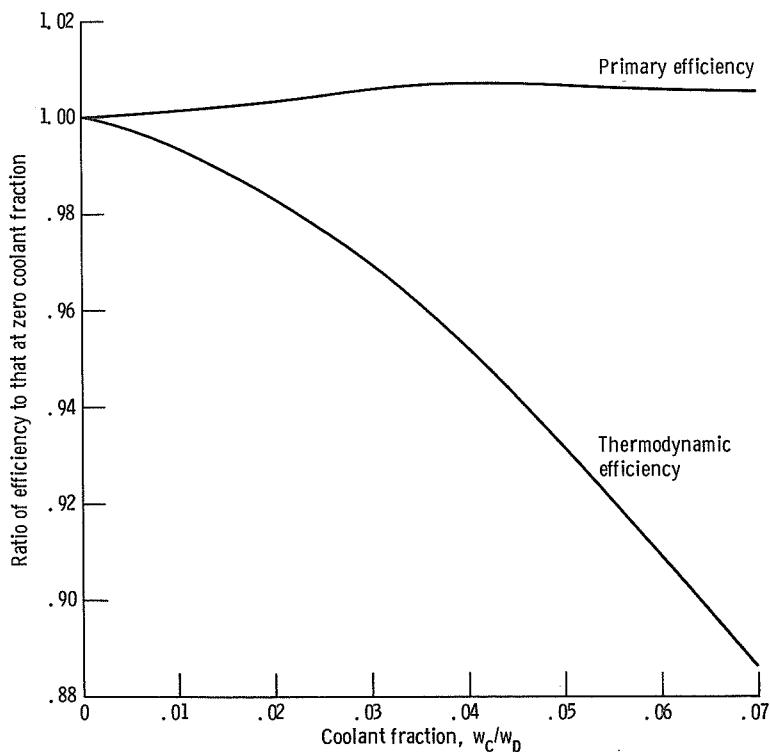


Figure 14. - Variation of efficiency with coolant fraction. Equivalent design rotor speed; specific-work output, 17.00 Btu per pound (39 572 J/kg) based on primary flow.

Figure 14 shows that the thermodynamic efficiency of the turbine decreased significantly with coolant fraction. For example, at a coolant fraction of 0.07 the turbine efficiency was down more than 11 percent.

## COMPARISON OF TURBINE PERFORMANCE WITH FOUR STATOR CONFIGURATIONS

In this section, performance parameters of the turbine with wire-mesh transpiration-cooled stator blades are compared with those of the other two cooled turbines and the uncooled base turbine. Curves of primary efficiency, thermodynamic efficiency, and primary flow are presented for the three cooled turbines over a range of coolant fraction from 0 to 0.07. Data were obtained with the turbines operating at the design equivalent speed and a primary specific work output of 17.00 Btu per pound (39 572 J/kg). Corresponding rotor component efficiencies were also determined, and are compared herein.

Turbine primary efficiency. - A comparison of the primary efficiencies for the three turbines with different cooled-stator configurations is shown in figure 15 for the range of coolant fractions tested. It is immediately apparent that the curve for the turbine having wire-mesh transpiration-cooled stator blades was relatively flat, increasing from 0.895 at zero coolant fraction to about 0.900 at coolant fractions of 0.03 and above. This trend is similar to that observed for the turbine having discrete hole transpiration-cooled stator blades (ref. 9), but the efficiency level is some 1/2 to 1 point lower. That the two efficiency curves rose slightly with coolant fraction is indicative that the cooling air did indeed produce some turbine work (see fig. 13). The similarity of the trends for the two turbines with transpiration-cooled stators was expected, inasmuch as both stators introduce cooling air into the primary flow by an effusion process. The fact that the turbine with wire-mesh stator blades resulted in lower efficiencies may be attributed to the blade contour irregularities as previously mentioned in the section APPARATUS AND INSTRUMENTATION.

In marked contrast, the turbine with slotted-trailing-edge stator blades produced a considerably higher primary efficiency level (fig. 15), with efficiency increasing continually with increasing coolant fraction. This was discussed in reference 7, where it was pointed out that the coolant air contributed to the turbine work output.

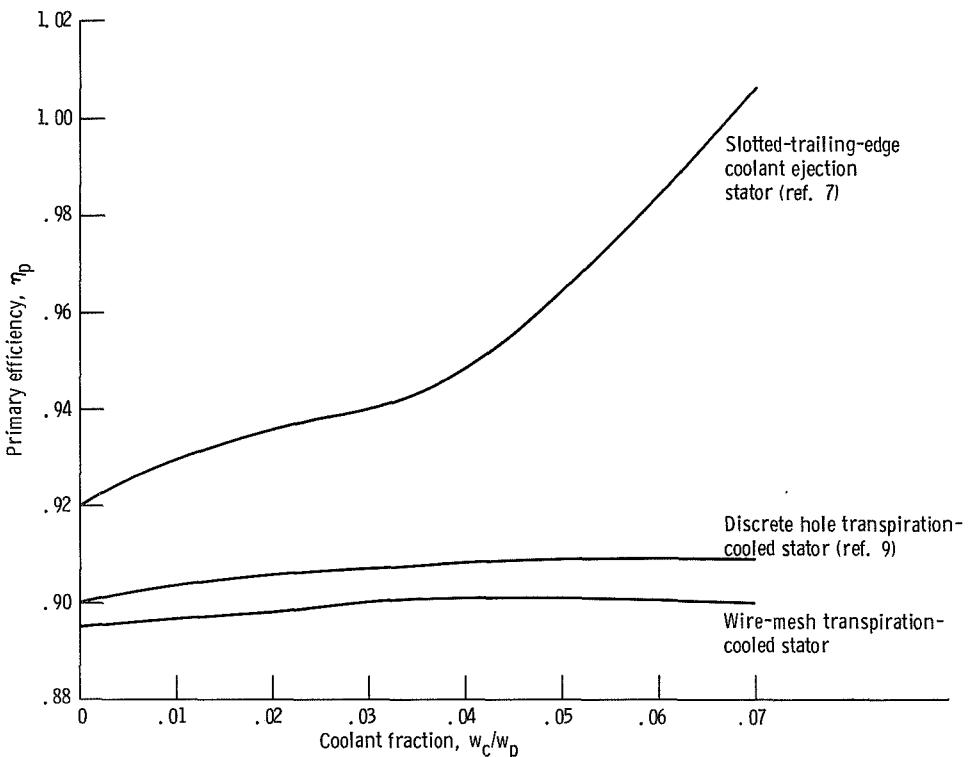


Figure 15. - Comparison of turbine primary efficiency with coolant fraction for three stator-cooling configurations. Equivalent design speed; specific-work output, 17.00 Btu per pound (39 572 J/kg) based on primary flow.

Turbine thermodynamic efficiency. - The thermodynamic efficiencies for the turbines with the three different cooled-stator configurations are shown in figure 16. The efficiencies for the two turbines with transpiration-cooled stator blades decreased in a parabolic fashion with increasing coolant fraction, with the subject wire-mesh stator-turbine configuration yielding the lower efficiency. With no coolant flow, this turbine yielded an efficiency of 0.895. At a coolant fraction of 0.07, the thermodynamic efficiency decreased to 0.793, some three points less than that observed for the turbine with the discrete hole stator blading. The increasing divergence of the two efficiency curves with added coolant fraction results from a required increase in coolant supply annulus pressure to yield a given coolant flow rate (see table I, coolant supply pressure). Increasing the coolant supply pressure results in a corresponding increase of coolant-flow ideal energy  $w_c \Delta h_{id,c}$  chargeable to the turbine (see eq. (4)). Obviously then, since very little of the available energy of the coolant was transformed into work output, the thermodynamic efficiency would be expected to decrease as indicated by figure 16.

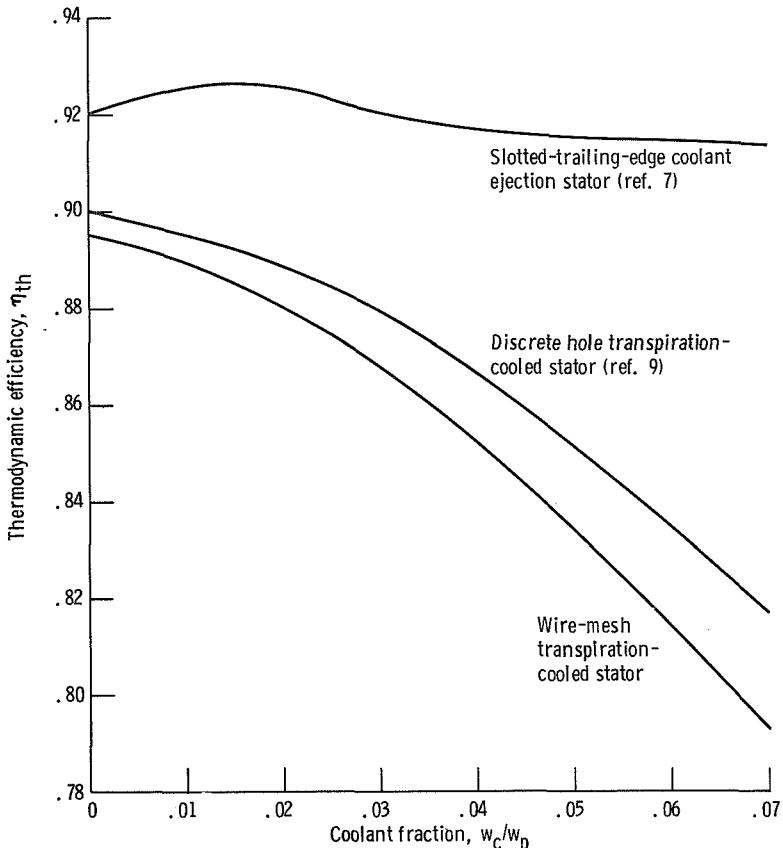


Figure 16. - Comparison of turbine thermodynamic efficiency with coolant fraction for three stator-cooling configurations. Equivalent design rotor speed; specific-work output, 17.00 Btu per pound (39 572 J/kg) based on primary flow.

The trend of thermodynamic efficiency with coolant fraction for the turbine with the trailing-edge coolant ejection stator blades (fig. 16) differs considerably from that found for the two turbines with transpiration-cooled stator blades. This turbine configuration exhibited a relatively flat efficiency trend ( $0.920 \pm 0.006$ ). Again, as stated for the primary efficiency curve (fig. 15), the cooling air developed useful turbine work when ejected from the stator-blade trailing edges, but very little useful work was added with the transpiration-type stator blades.

Both the primary and thermodynamic efficiency curves (figs. 15 and 16) have indicated the inability of the coolant air to develop significant useful turbine work with the transpiration-cooled stators as tested. This disadvantage, compared to that for the turbine with the slotted-trailing-edge stator configuration, is inherent since the coolant flow is injected normal to the primary-air flow. In contrast, cooling air from the slotted-trailing-edge stator blades is in the direction of the primary-air flow.

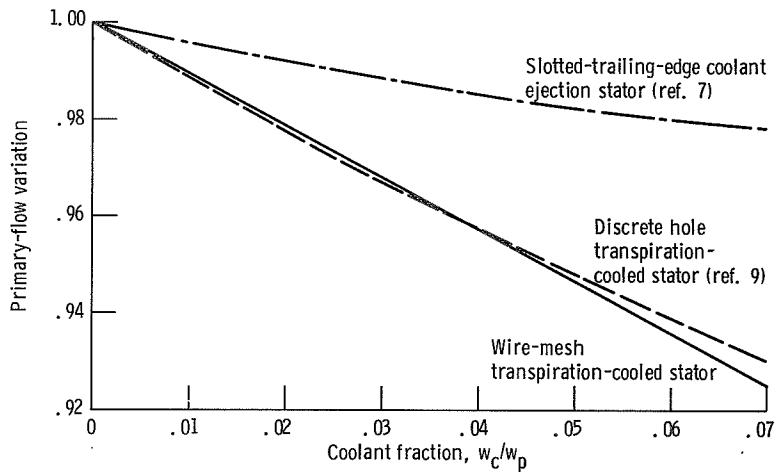


Figure 17. - Comparison of turbine primary flow variation with coolant fraction for three stator cooling configurations. Equivalent design rotor speed; specific-work output, 17.00 Btu per pound (39 572 J/kg) based on primary flow.

Primary flow. - The variation in primary flow for the cooled-stator configurations is shown in figure 17. The primary flow variation of the two turbines with transpiration-cooled stator blades is almost identical. The reduction in primary flow is essentially in direct proportion to the coolant fraction addition. In the two turbines with transpiration-cooled stator blades, most of the coolant is added upstream of the stator throat and causes blockage of the primary flow. In addition, the coolant is ejected normal to the primary flow path and reduces the momentum of the primary-air flow.

The reduction in primary flow for the turbine with slotted-trailing-edge stator blades is much less than for the turbines with transpiration-cooled stator blades. This reduced variation is a result of the coolant being ejected downstream of the stator throat and nearly parallel to the primary flow.

Rotor efficiency. - In reference 7 it was concluded that the rotor efficiency remained relatively constant over the range of coolant fractions investigated when the rotor was used with the stator with trailing-edge slots. In reference 9, however, it was shown that rotor efficiency was affected by the stator cooling air of the discrete hole transpiration-cooled stator. It was of interest, thus, to determine whether the rotor efficiency was also affected by the cooling air emitting from the wire-mesh transpiration-cooled blades.

The rotor efficiency used herein is the same as defined in reference 9 using the equation:

$$\eta_r = \frac{\Delta h_r}{\Delta h_{id,r}} = \frac{\frac{\tau N \pi}{30(w_p + w_c)J}}{\left( \frac{V_1^2}{2gJ} \right) + (h_1 - h_{id,2}^*)} \quad (5)$$

which expresses the specific work output of the total flow through the rotor to the ideal work. The ideal work, as used herein, is equal to the absolute kinetic energy of the flow from the stator plus the ideal enthalpy drop across the rotor corresponding to the static-to total-pressure drop across the rotor  $p_1/p_2^*$ . The kinetic energy from the stator  $V_1^2/2gJ$  was determined from measured pressures and the experimental stator results of reference 10. Station 1 is shown in figure 6 and it is assumed that the values obtained at this station are for the averaged, after-mixed, uniform flow regime, which is probably not the case, as will be explained later.

Rotor efficiency for the turbine with wire-mesh transpiration-cooled stator blades is shown as a function of coolant fraction in figure 18. The rotor efficiency is 0.918 at zero coolant fraction and decreases to about 0.906 at a coolant fraction of 0.06. A slight rise in efficiency occurred when the coolant fraction was increased from 0.06 to 0.07. That the rotor efficiency generally decreased with coolant fraction can be related to the previously mentioned 10-point drop in thermodynamic efficiency at the 7-percent coolant-air addition (see fig. 16), indicating the stator contributed to most of the turbine loss.

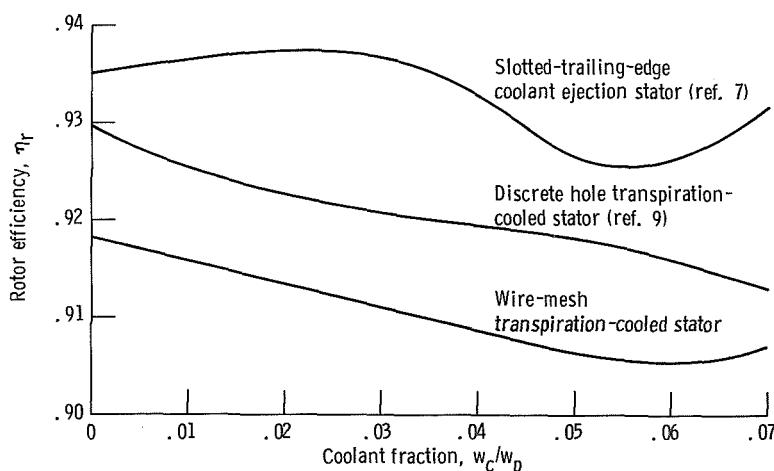


Figure 18. - Comparison of rotor efficiency with coolant fraction for two types of transpiration-cooled blades. Equivalent design rotor speed; specific-work output, 17.00 Btu per pound (39 572 J/kg) based on primary flow.

Also shown in figure 18, for comparison, are the rotor efficiencies for the turbines with the discrete hole transpiration-cooled stator blades and slotted-trailing-edge stator blades. A comparison of the two rotor efficiency curves for transpiration-cooled stator blades shows that, at zero coolant fraction, the efficiency of the turbine with wire-mesh stator blades is approximately  $1\frac{1}{4}$  points lower than the 0.93 rotor efficiency of the turbine with discrete hole stator blades. As coolant fraction is increased, this difference remains essentially constant until, at a coolant fraction of 0.07 the difference is about 1/2 point.

The rotor efficiency of the turbine with slotted-trailing-edge stator blades is 0.935 at zero coolant fraction. As the coolant fraction is increased, the rotor efficiency increases 1/4 point at 0.02 coolant fraction, then decreases almost 1 point at coolant fractions of 0.05 and 0.06. Increasing the coolant fraction to 0.07 results in an efficiency rise to a value of 0.932.

The decrease in rotor efficiency for both turbines with transpiration-cooled stator blades was probably a result of the thickening boundary layer caused by the interaction of the coolant and primary flow on the suction or convex surface of the stator blades (refs. 8 and 10). This interaction of flows produced wakes at the stator exit with larger non-uniformities in the radial and circumferential velocity profiles than were encountered with either stator trailing-edge coolant ejection (ref. 6) or the base uncooled stators (ref. 2). As the figures show, this effect is more pronounced in the turbine with wire-mesh transpiration-cooled stator blades than in the turbine with discrete hole transpiration-cooled stator blades. Another factor causing this difference could be the undeturning at the hub of the turbine with wire-mesh transpiration-cooled stator blades, as discussed in reference 10.

## SUMMARY OF RESULTS

A cold-air, 30-inch (0.762-m), single-stage experimental turbine having wire-mesh transpiration-cooled stator blades was tested to determine the effect of coolant flow ejection on the turbine aerodynamic performance. The testing was conducted in two phases. First, the turbine was operated over a range of speed and pressure ratio with a nominal coolant fraction of 0.031. Second, at equivalent design speed and a primary specific work

output of 17.00 Btu per pound (39 572 J/kg), the coolant fraction was varied from 0 to 0.07. The performance of this turbine was compared to those obtained from similar tests wherein the turbine was equipped (1) with transpiration-cooled stator blades having discrete holes and (2) with stator blades having trailing-edge coolant ejection slots. Performance characteristics for all three cooled-turbine configurations are compared to a base reference turbine which had no coolant flow provisions. Inlet total-state conditions for all four turbines were 30.0 inches of Hg absolute ( $10.16 \text{ N/cm}^2$ ) and about  $545^\circ \text{ R}$  (303 K). The following major results were obtained:

1. The subject turbine yielded a primary-air efficiency of 0.900 at the equivalent design speed and a primary specific work output of 17.00 Btu per pound (39 572 J/kg), operating conditions that correspond to the equivalent design conditions for the base turbine. The attendant pressure ratio was 1.779, and the coolant fraction was 0.031.
2. The performance trends at design speed (primary efficiency, thermodynamic efficiency, and primary flow as a function of coolant fraction) that were obtained herein with the subject stator were very similar to those obtained with the turbine having discrete hole transpiration-cooled stator blades. The trends obtained for both types of transpiration-cooled stator blade differed markedly from those obtained for the turbine with trailing-edge coolant ejection stator blades.
3. The primary efficiency of the turbine, which relates the work output to the ideal energy of only the primary flow, was essentially flat over the range of coolant fraction tested. The primary efficiency was 0.895 at zero coolant fraction, increasing to about 0.900 at a coolant fraction of 0.03 and above.
4. The thermodynamic efficiency of the turbine, which relates the work output to the ideal energy of both the primary flow and the coolant flow, decreased with increasing coolant fraction and exhibited a trend similar to that obtained with the turbine that had discrete hole transpiration-cooled stator blades. At zero coolant fraction the efficiency was 0.895, with a drop of 10 points when the coolant fraction was increased to 0.07. The low thermodynamic efficiency encountered in the turbine with wire-mesh transpiration-cooled stator blades was caused primarily by the inability of the stator coolant to develop useful work output. An inherent disadvantage of this stator coolant scheme is that the coolant flow is ejected normal to the primary-air flow direction.

5. The rotor efficiency was reduced by 1 point as the coolant fraction was varied from 0 to 0.07. This loss was attributed to the increased nonuniformities in circumferential velocity profile at the stator exit due to the addition of coolant.

Lewis Research Center,

National Aeronautics and Space Administration,

Cleveland, Ohio, January 8, 1971,

720-03.

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