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Aerodynamic Effect of a Honeycomb Rotor Tip Shroud on a 50.8-Centimeter-Tip-Diameter Core Turbine

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National Aeronautics
and Space Administration

Scientific and Technical
Information Branch

SUMMARY

A 50.8-cm-tip-diameter turbine equipped with a rotor tip shroud of hexagonal cell (or honeycomb) cross section has been investigated in warm air (416 K). Test results were also obtained for the same turbine operated with a solid shroud for comparison purposes. The objectives of the investigation were (1) to determine the combined effect of the honeycomb and the honeycomb shroud coolant flow on turbine performance and (2) to determine the effect of the honeycomb with the coolant flow turned off.

The results showed that the combined effect of the honeycomb shroud and the coolant flow was a reduction of 2.8 points in efficiency at design speed, pressure ratio, and coolant flow rate. With the coolant system inactivated, the honeycomb shroud caused a decrease in efficiency of 2.3 points. These results and those obtained from a small reference turbine indicate that the dominant factor governing honeycomb tip shroud loss is the ratio of honeycomb depth to blade span. The loss results of the two turbine tests could be correlated on this basis.

The same honeycomb and coolant effects are expected to occur for the hot (2200 K) version of this turbine.

INTRODUCTION

There is a considerable effort at NASA Lewis Research Center involved with aerodynamic, heat transfer, and life aspects of high-temperature core turbines. A major part of this program is the design and evaluation of a 50.8-cm-diameter research turbine designed for an inlet temperature and pressure of 2200 K and 386 N/cm^2 (hereinafter called the hot research turbine). At the high temperatures involved, considerable cooling is required for all parts that come in contact with the hot gas. The static shroud over the rotor consists of a honeycomb shroud backed by a transpiration-cooled layer of Poroloy. The backside of the Poroloy is, in turn, impingement-cooled by flow metered through small holes in a supply plate. The use of honeycomb provides protection against rotor tip rubs. However, it can cause large penalties in aerodynamic performance. For example, test results obtained from a small (13.5-cm diam) aircraft starter turbine designed with a honeycomb shroud showed about an 8-point penalty in efficiency compared with those obtained with the honeycombs filled to effect a solid shroud (ref. 1). When the honeycomb was filled to 50 percent of its depth, about 80 percent of the deficit was recovered. Although the honeycomb depth of the 50.8-cm-diameter research turbine is shallower, relative to the blade height, than the 13.5-cm starter turbine, there was concern that the loss could be significant.

It was considered desirable to determine the expected penalty in performance of the 50.8-cm hot research turbine before testing. An existing 50.8-cm uncooled turbine that had already been tested (refs. 2 to 5) was selected for these tests. It had the same diameters and blading profiles as the hot research turbine. The same honeycomb shroud as well as a solid shroud was fabricated for the test turbine. Rotor tip running clearances were measured during testing to determine differences in performance due to clearance change.

The turbine was tested in the warm core turbine facility at Lewis (ref. 6). Tests were conducted at nominal inlet conditions of 461 K and 28.6 N/cm^2 and at the design equivalent rotative speed of 7832 rpm. A range of overall total pressure ratio from 1.5 to 2.4 was set across the turbine. In addition, at design pressure ratio (1.81), static shroud coolant flow fraction (ratio of coolant flow to primary inlet flow) was varied from 0 to 0.025. This bracketed the design value of 0.015 coolant flow, which, for the geometry and coolant metering holes involved, gave the same volumetric flow ratio of coolant-to-primary flows for the test turbine as for the hot research turbine.

Efficiency results are presented for the subject turbine tested with the solid shroud and with the honeycomb shroud, both with and without coolant. In addition, the results are compared with those of the smaller turbine of reference 1.

SYMBOLS

C	clearance, mm
c	chord length, cm
D	diameter, cm
h	vane or blade height, cm
$\Delta h'$	specific work, J/g
N	rotative speed, rpm
p	pressure, N/cm^2
r	radius, cm
s	spacing, cm
T	absolute temperature, K
U	blade velocity, m/sec
V	absolute gas velocity, m/sec
W	relative gas velocity, m/sec
w	mass flow rate, kg/sec
y	coolant fraction, ratio of coolant flow-to-primary flow
δ	ratio of inlet total pressure to U.S. standard sea-level pressure, p_0/p^*
ε	function of γ in relating parameters to those using air inlet conditions at U.S. standard sea-level conditions $(0.73959/\gamma)[(\gamma+1)/2]^{\gamma/(\gamma-1)}$
η	total efficiency, based on inlet-total to exit-total pressure ratio
θ_{cr}	squared ratio of critical velocity at turbine inlet to critical velocity at U.S. standard sea-level air $(V_{cr}/V_{cr}^*)^2$

Subscripts:

b blade

cr condition corresponding to Mach 1

h hub section

m mean section

ref reference

t tip section

v vane

x axial component

0 station in turbine inlet plenum

1 station at turbine inlet

2 station at vane exit, also blade inlet

3 station at turbine exit

Superscripts:

' absolute total state

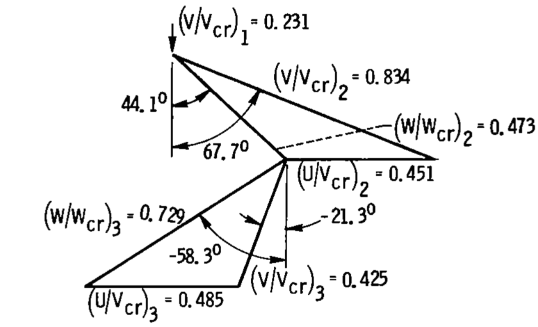
* U.S. standard sea-level conditions

TURBINE DESCRIPTION

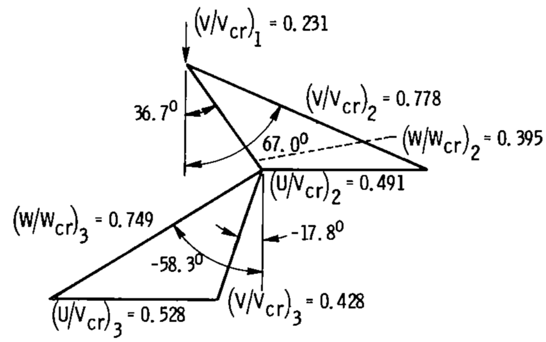
The 50.8-cm-diameter turbine of this investigation was equipped with solid vanes and blades described in references 2 and 5. Initial tests were conducted with a solid static shroud over the rotor which included a Nichrome alloy (80 percent Ni - 20 percent Cr) rub strip to prevent rotor tip damage. The solid shroud was then replaced with the honeycomb shroud and the turbine retested with and without shroud coolant.

Base Uncooled Turbine

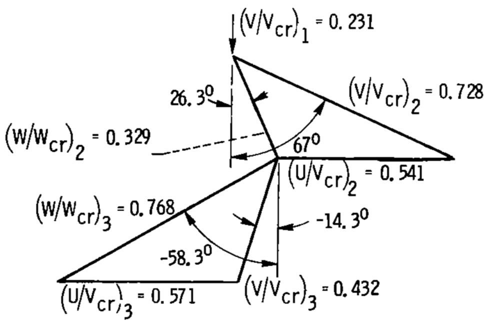
The design operating values for the single-stage core turbine are summarized in table I. Actual engine, equivalent, and test conditions are shown. The velocity diagrams evolved to meet the design aerodynamic requirements are shown in figure 1. All quantities represent the free-stream, uniform flow conditions before and after each blade row. The test turbine geometry values are summarized in table II, and the vane and blade coordinates are shown in table III. The vanes and blades are characterized by blunt leading and trailing edges to allow for cooling, low aspect ratio, and high thickness-to-chord ratio. The test turbine vanes and blades were fabricated with no twist at constant mean-section profile to facilitate fabrication of the complex internal inserts used for cooling tests (refs. 3 and 5).



(a) Hub section; radius ratio $r_h/r_t = 0.850$.

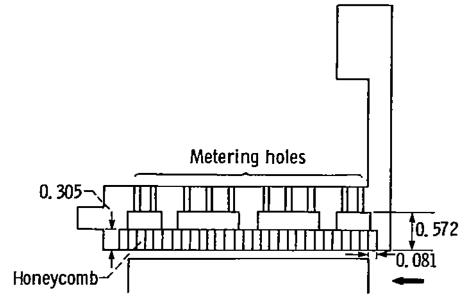


(b) Mean section; radius ratio $r_m/r_t = 0.925$.

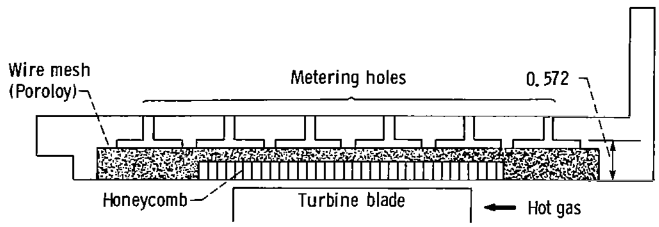


(c) Tip section; radius ratio $r_t/r_t = 1.000$.

Figure 1. - Turbine design velocity diagram.



(a)



(b)

(a) Subject test turbine.
(b) Hot research turbine.

Figure 2. - Turbine tip static honeycomb shrouds. (All dimensions in cm.)

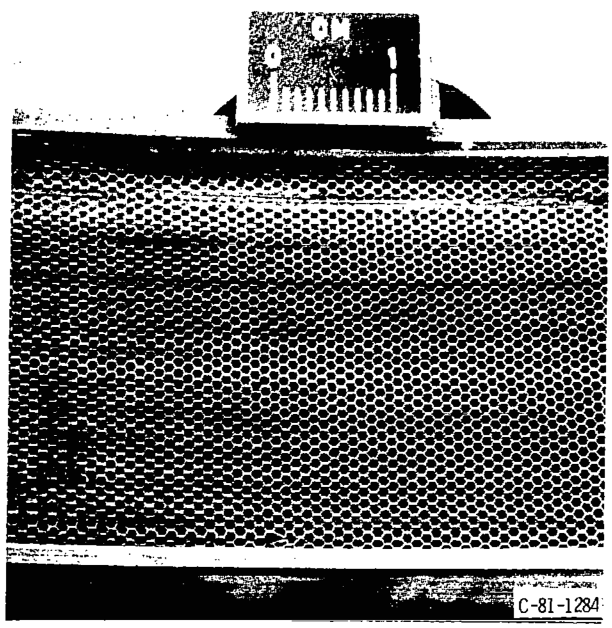


Figure 3. - Subject turbine tip static honeycomb shroud.

Honeycomb Static Rotor Shroud

A schematic of the honeycomb static shroud tested is shown in figure 2(a). Also shown in figure 2(b) is a schematic of the hot research honeycomb shroud for comparison. Both have hexagonal honeycomb cells with a width of 0.081 cm and depth of 0.305 cm. The Poroloy wire mesh designed for the hot turbine was not used in the test turbine because of prohibitive fabrication problems. It was felt that the loss resulting from the shroud coolant flowing radially inward into the blade passage was not affected by the omission of the Poroloy mesh. When the shroud coolant system was closed off, it was assumed that the Poroloy omission did not affect the loss associated with the flow in the blade channel surging into and out of the honeycomb shroud, providing that the effective honeycomb depth is taken to include the depth of the supply plenum (see fig. 2(a)).

The metering holes were sized to provide the same volumetric flow ratio of shroud coolant-to-primary flow between the test turbine and hot research turbine. This was felt to be the most reasonable correlating parameter to duplicate loss results. At this condition the shroud coolant-flow fraction (ratio of coolant flow-to-primary inlet flow) was 0.015, or 1.5 percent of the primary flow, which was specified as the design coolant flow for the test program.

A posttest photograph of the honeycomb shroud looking radially outward is shown in figure 3. The slight circumferential rub that can be seen occurred after test data were taken and did not influence the results.

APPARATUS

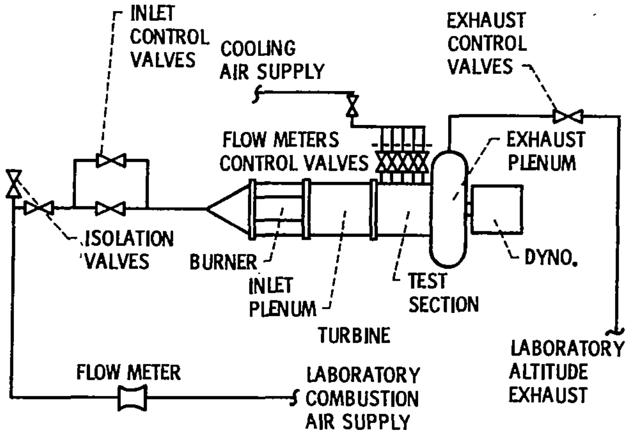
The apparatus consisted of the turbine, as described in the preceeding section, an absorption dynamometer, and an inlet and exhaust system with suitable flow control valves (fig. 4). A water-cooled electric dynamometer was used to absorb turbine power output and control its speed.

Air is supplied from the laboratory combustion air system through a venturi flowmeter located in a straight section of inlet piping. The inlet pressure is then reduced and controlled by suitable valves in the air line. The air is then directed to a natural gas vitiated air heater and then through the turbine test section. The turbine exit pressure was set by a control valve in the exhaust line. The air was then ducted into the laboratory altitude exhaust system.

In addition to the primary air flow, the turbine facility is provided with auxilliary cooling air circuits, each individually metered and controlled. One of these circuits was used to supply the honeycomb shroud coolant.

INSTRUMENTATION

A schematic cross section of the turbine showing measuring stations is shown in figure 5. Turbine inlet and exit locations are defined at stations 1 and 3, respectively. The turbine inlet temperature was determined by 25 shielded thermocouples located at station 0. There were five thermocouples, each located at the inlet edge of five of the 10 struts shown in figure 5. Total pressure was also measured at station 0 by five total pressure probes, which were located at the centers of five equal annular areas on the leading edges of five struts.



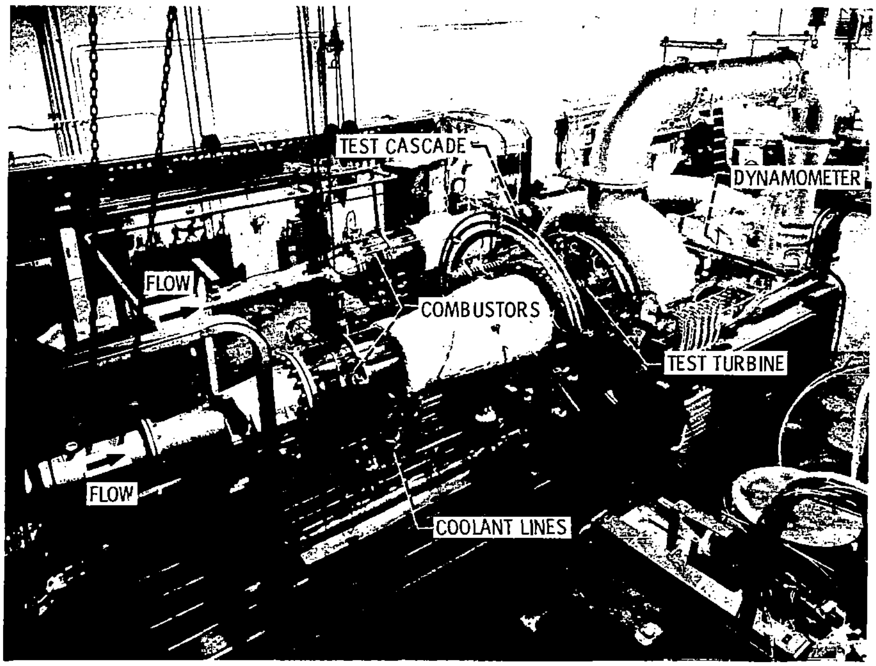


Figure 4. - Test facility.

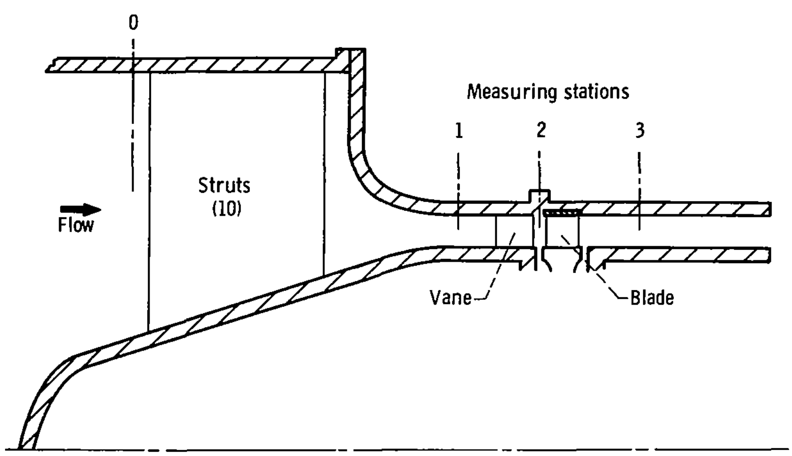


Figure 5. - Turbine test section measuring station locations.

The instrumentation at the turbine inlet (station 1) consisted of five static pressure taps, each located on the inner and outer walls. Station 1 is located 3.8 cm upstream of the stator vane leading edge.

The turbine outlet (station 3) is located 6.6 cm downstream of the rotor blade trailing edge. Instrumentation at the turbine outlet consisted of static pressure, total pressure, and flow angle measuring devices. Static pressures were measured by six static pressure taps each located on the inner and outer walls. Total pressures and flow angles were measured from four, self-alining, combination probes located at the center of four equal annular areas spaced around the circumference of the outlet plane.

The primary airflow was measured with a venturi tube and the gas flow was measured with a flow nozzle. The coolant air was measured with a flat-plate orifice. These three flow rate measurements required an upstream pressure measurement, upstream temperature measurement, and a measurement of differential pressure across the flowmeter element.

The speed of the turbine was measured by using an electronic counter in conjunction with a 60-tooth gear mounted on the turbine shaft. The primary torque measurement was a brushless rotating torque meter (fig. 6) mounted between the turbine and the dynamometer. A backup torque measurement was also obtained by measuring the reaction torque on the dynamometer stator, which was supported on hydrostatic trunion bearings.

PROCEDURE

Turbine performance was obtained for both series of tests at nominal primary air inlet conditions of 461 K and 28.6 N/cm^2. For both the solid and the honeycomb shroud with no coolant tests, data were obtained at design equivalent speed (7832 rpm) and over a range of calculated inlet-to-exit-total pressure ratio from 1.5 to 2.4. For the honeycomb shroud with coolant tests, data were obtained at design equivalent speed and pressure ratio (1.813) conditions and for shroud coolant flow fractions from 0 to 0.025.

The actual specific work output of the turbine was determined from the speed, torque, and mass flow measurements.

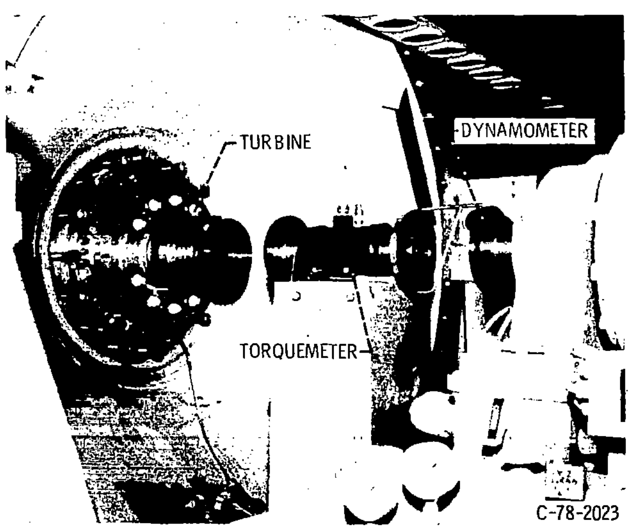


Figure 6. - Inline brushless torquemeter.

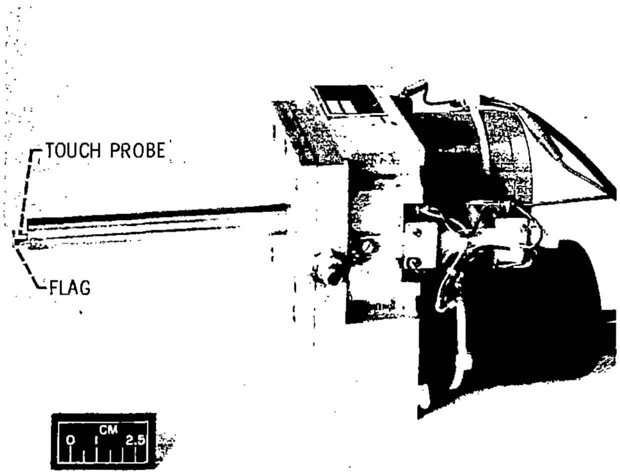


Figure 7. - Blade tip clearance probe.

Turbine efficiency was rated on the basis of inlet-to-exit-total pressure ratio. Turbine ideal specific work output was based on inlet and exit total pressures calculated from measurements of mass flow, static pressure, total temperatures, and flow angle. Outlet total temperature was calculated from measurements of inlet total temperature, torque, speed, and mass flow. When honeycomb shroud coolant flow was used, the ideal power of the coolant was included and was based on measurements of coolant mass flow, coolant inlet total pressure and temperature, and rotor exit total pressure.

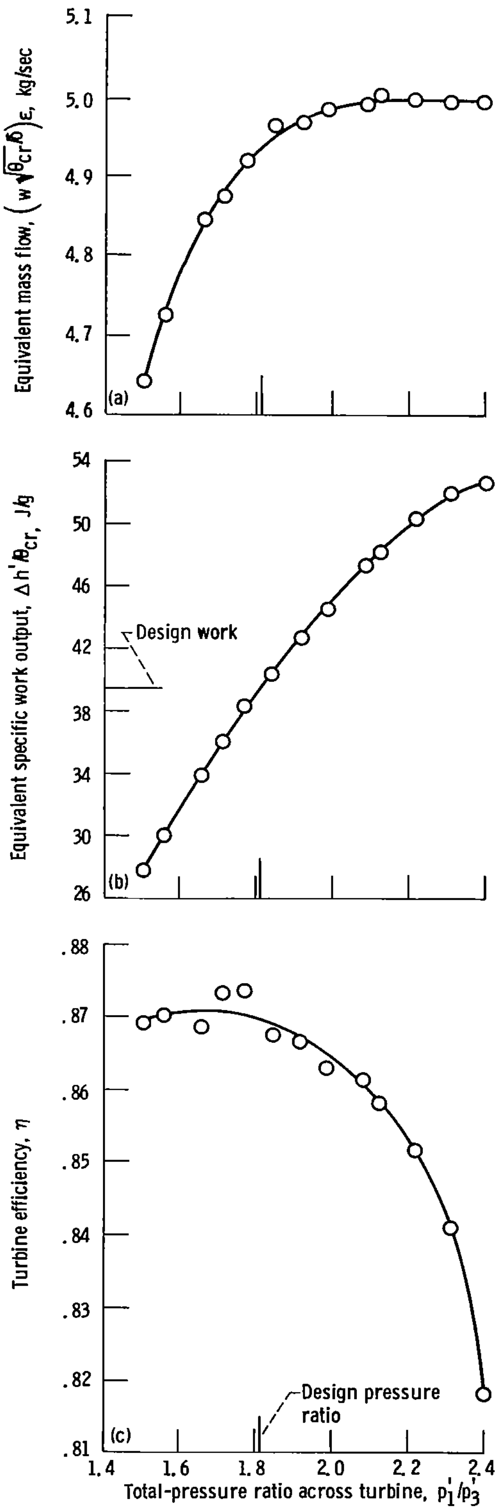
Corrections were made for calibrated turbine bearing losses and changes in rotor tip clearance. The rotor tip clearance was measured with a self-calibrating touch probe (fig. 7).

RESULTS AND DISCUSSION

The results of this investigation are presented in three parts: The first section covers the basic performance of the turbine with the solid static shroud; the second covers the results obtained with the use of the honeycomb static shroud with and without coolant; the third presents a comparison of the performance of the subject turbine (50.8 cm diameter) with honeycomb shroud and the small starter turbine (13.5 cm diameter) with honeycomb shroud (ref. 1). Performance is expressed in terms of specific work output, mass flow, and efficiency. The data are shown in terms of equivalent air conditions; for simplicity, the term "equivalent" will be implied and not used in the discussion.

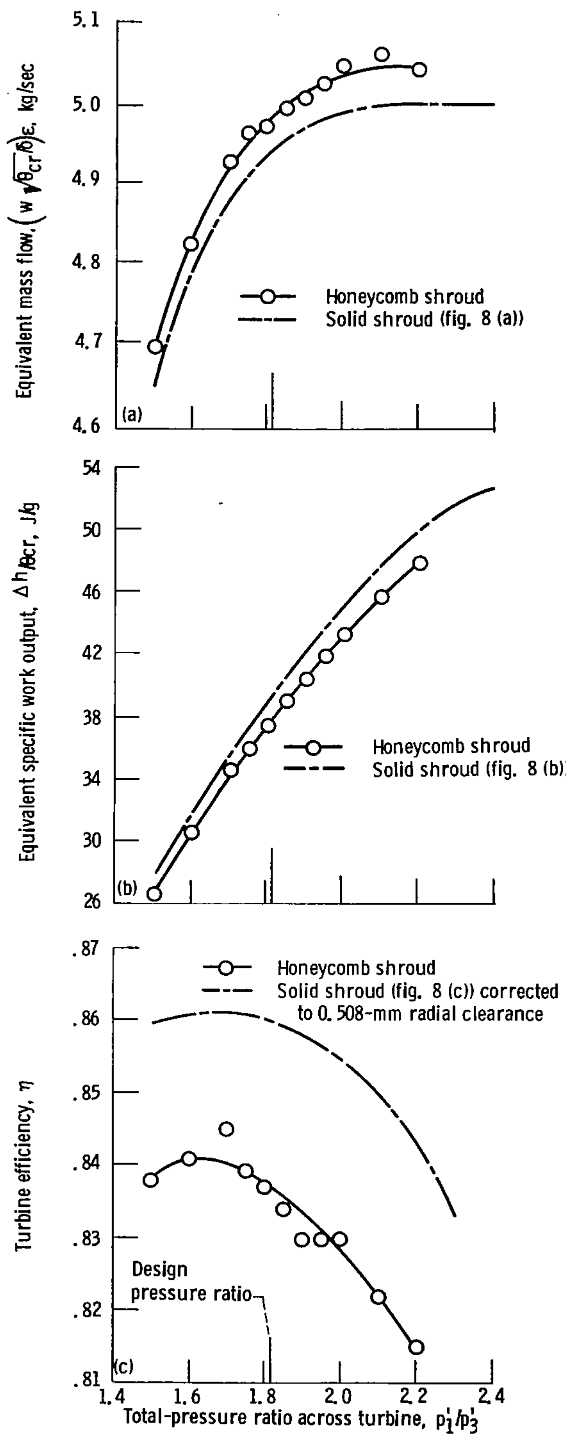
Turbine Performance With Solid Shroud

The overall performance of the turbine with a solid rotor tip static shroud is shown in figure 8. All data were taken at design speed. Design work output (fig. 8(b)) was obtained at an overall pressure ratio of 1.813, which is shown as a tick on subsequent curves and labeled "design pressure ratio." A substantial margin in work is seen to be available before limiting loading occurs and is typical of a turbine designed for a moderate



(a) Mass flow rate.
(b) Specific work output.
(c) Turbine efficiency.

Figure 8. - Performance of turbine with solid rotor shroud at design speed.



(a) Mass flow rate.
(b) Specific work output.
(c) Turbine efficiency.

Figure 9. - Effect of honeycomb shroud on turbine performance with no coolant flow at design speed.

Mach number level. The mass flow at design pressure ratio is 4.94 kg/sec, which is about 1 percent lower than the choking mass flow of 5.00 kg/sec, and 2 percent higher than the design value of 4.83 kg/sec (fig. 8(a)). The efficiency of the turbine at design pressure ratio is 0.87 and falls off to about 0.82 at the highest pressure ratio tested (2.4; fig 8(c)). The efficiency value of 0.87 obtained at design pressure ratio is the same as that obtained when tested with solid vanes and blades and a porous rotor static tip shroud that was cooled with about 1 percent coolant fraction (ref. 5). The shroud coolant of the reference turbine was estimated to have an insignificant effect on efficiency. The duplication of basic performance data was therefore considered to be very good.

Turbine Performance With Honeycomb Shroud

Clearance correction. - The actual running clearances remained constant with pressure ratio at design speed for the solid shroud turbine and the uncooled honeycomb shroud turbine at 0.254 and 0.508 mm, respectively. The solid shroud efficiency data (fig. 8(c)) were corrected to the same clearance as the honeycomb shroud turbine using the following equation (from ref. 7):

$$\frac{\eta - \eta_{ref}}{\eta_{ref}} = -1.7 \frac{C - C_{ref}}{h_b}$$

The correction amounted to reducing the solid shroud data by 1 efficiency point.

Honeycomb shroud with no coolant. - The effect of the honeycomb shroud with no coolant on overall performance is shown in figure 9. The mass flow through the turbine (fig. 9(a)) at design pressure ratio increased by 1 percent when the honeycomb shroud replaced the solid shroud. This is partly explained by the 0.7-percent increase in annular area caused by the increased clearance of the honeycomb shroud and attendant increase in tip leakage air. The corresponding work output (fig. 9(b)) is 37.91 J/g, which is 4.1 percent lower than the design value of 39.54 J/g. A part of this difference is attributable to the difference in tip clearance. The figure indicates a consistent reduction in work output due to the honeycomb over the entire range of pressure ratio tested. This deficit is reflected in the efficiency comparison of figure 9(c). The solid shroud efficiency data in this figure have been corrected to the same clearance as the honeycomb shroud data. As mentioned previously, the correction amounted to a reduction in solid shroud efficiency of one point. At design pressure ratio, the honeycomb shroud turbine efficiency (fig. 9(c)) is 0.837, which is 2.3 points lower than the solid shroud turbine efficiency of 0.860. Although there is some scatter in the honeycomb data, this level of efficiency degradation persisted over the range of pressure ratio tested.

Tests were also conducted with a minimum (essentially zero) coolant flow. It was felt that with coolant off, there was a possibility that the turbine working fluid could flow into the coolant shroud system at the blade inlet and back into the blade passage at the blade outlet where the static pressure is lower. The results obtained with minimum coolant flow were indistinguishable from those obtained with the coolant flow turned off, thus bypass effect was considered negligible.

Effect of honeycomb shroud coolant. - The effect of shroud coolant on turbine efficiency at design speed and pressure ratio is shown in figure 10.

Also shown for comparison as a single data point is the solid shroud turbine efficiency value of 0.860. Two sets of data points are shown for the honeycomb shroud turbine; the upper curve (circles) is from data as measured. These data were then corrected to a constant radial clearance of 0.508 mm and resulted in the lower curve, which is used for discussion. The radial clearance changed due to shrinkage of the shroud as relatively cold air was introduced as coolant and varied from 0.508 at no coolant to 0.330 mm at 2.2-percent coolant. Again, using reference 7 as a guide, this amounted to an expected efficiency change of 0.7 point at the maximum coolant involved. Reference 7 (solid shroud turbine) was used to correct for clearance change in lieu of any known clearance data available for honeycomb shrouds.

Figure 10 indicates that, when coolant was initially added, the efficiency first increased a slight amount and then decreased linearly at a rate of 1.2 efficiency points per percent of coolant. The initial increase in performance is probably caused by the coolant filling the honeycomb cells and retarding the tendency of the working fluid to surge into and out of the honeycomb area. The decreasing efficiencies probably result from the available energy of the coolant being charged to the turbine.

At the design coolant fraction of 0.015, the corrected efficiency was 0.832, which is one-half a point lower than the value of 0.837 obtained with no coolant. The overall combined effect of the honeycomb plus 1.5 percent coolant is therefore a penalty in efficiency of 2.8 points compared with operation with a solid shroud. This is the expected penalty associated with the use of the cooled honeycomb shroud for the hot 50.8-cm-diameter research turbine described in the Introduction.

Comparison with starter turbine. - It was suggested in reference 1 that the reduced efficiency encountered with the use of a honeycomb shroud was due primarily to the energy loss associated with gas being transported into and out of the honeycomb cells as a result of the blade-to-blade pressure differential at the tip section. The data showed that the magnitude of the loss increased with increases in cell depth and rotative speed. The results of the subject turbine are compared with those of the starter turbine (ref. 1) in an attempt to gain further insight into the geometric and operating factors that influence the magnitude of honeycomb losses.

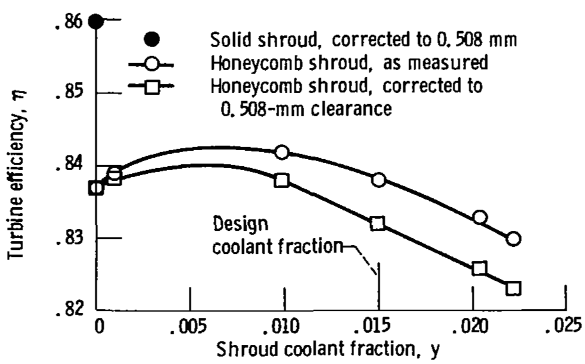


Figure 10. - Effect of honeycomb shroud coolant on turbine efficiency at design speed and pressure ratio.

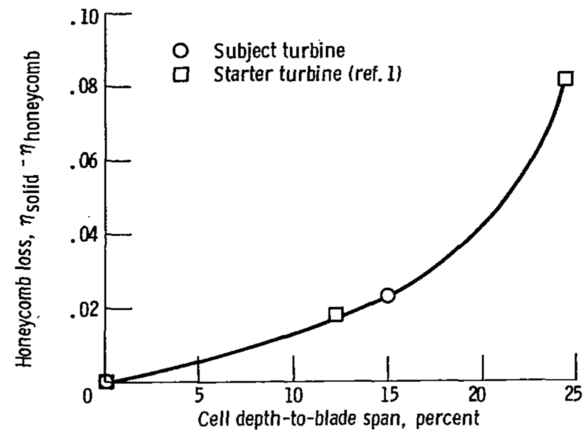


Figure 11. - Comparison of honeycomb losses for subject and starter turbines at design speed and pressure ratio.

The geometry of the subject turbine is shown along with that of the starter turbine in table IV. A comparison of the two columns for the starter turbine indicates that reducing the relative cell depth by one-half had a dramatic effect on loss - a decrease from 8.2 points for the open honeycomb to 1.8 points for the half-filled honeycomb. The corresponding relative cell depth (honeycomb cell depth-to-blade span ratio) of the subject turbine is 15 percent, as compared with 24.4 and 12.2 percent for the starter turbine's open and half-filled cells, respectively. The indicated efficiency penalty for the subject turbine is 2.3 points, which falls between the two values for the starter turbine. These data (fig. 11) indicate a correlation curve in the data from the two turbines. It is apparent that the dominant influence on honeycomb associated losses is the relative cell depth, or ratio of the honeycomb depth to the blade span. Other factors such as blade-passing frequency, reaction, blade loading, and cell width are felt to have only a secondary influence on the honeycomb losses.

SUMMARY OF RESULTS

An experimental investigation was conducted on a 50.8-cm-diameter turbine to determine the losses associated with the use of a cooled honeycomb static tip shroud over the rotor. The honeycomb had the same dimensions and was designed for the same volumetric flow ratio (coolant-to-primary flow) as a 50.8-cm-diameter hot research turbine (2200 K) to be tested at Lewis Research Center. Tests were conducted with and without shroud coolant flow and also with a solid shroud to obtain base performance. The results were also compared with those from a small 13.5-cm-diameter starter turbine that used a honeycomb shroud. The results are summarized as follows:

1. At design equivalent speed and pressure ratio, the efficiency of the subject honeycomb shroud turbine with no shroud coolant was 0.837, which is 2.3 points lower than the value of 0.860 obtained with the solid shroud installed.

2. As shroud coolant was activated, the efficiency increased a small amount and then decreased linearly at a rate of 1.2 points per percent of coolant. At the design coolant rate of 1.5 percent of primary flow, the efficiency was 0.832 for a combined penalty (honeycomb plus coolant) of 2.8

points. The same honeycomb and coolant effects are expected to occur for the hot (2200 K) version of this turbine.

3. A comparison with the results of the starter turbine indicated that the magnitude of honeycomb losses appears to be primarily dictated by the ratio of cell depth-to-blade span and, to a much lesser degree, by other geometric and operating characteristics.

Lewis Research Center
National Aeronautics and Space Administration,
Cleveland, Ohio, September 29, 1982

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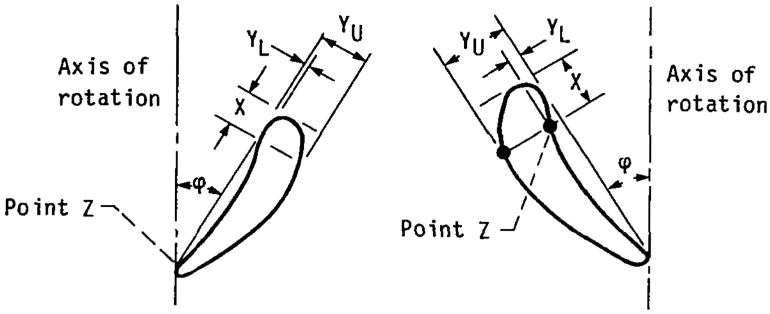
TABLE I. - TURBINE DESIGN OPERATING VALUES

Parameter	Actual	Equivalent	Test conditions
Tip diameter, D_t, cm	50.8	50.8	50.8
Inlet total temperature, T_0', K	2200	288.2	461.1
Inlet total pressure, p_0' N/cm^2	386.1	10.13	28.54
Mass flow rate, w, kg/sec	63.82	4.828	10.72
Turbine rotative speed, N, rpm	16687	6194	7832
Specific work output, $\Delta h'$, J/g	287.2	39.54	63.21
Mean blade speed, U_m, m/sec	410.6	152.4	192.7

TABLE II. - TEST TURBINE GEOMETRY

Stator	
Mean diameter, D_{mv}, cm	46.99
Vane height, h_v, cm	3.81
Axial chord, c_{xv}, cm	3.81
Axial solidity, $(c_{xv}/s_v)_m$	0.929
Aspect ratio, h_v/c_{xv}	1.000
Number of vanes	36
Leading edge radius, cm	0.508
Trailing edge radius, cm	0.089
Rotor	
Mean diameter, D_{mb}, cm	46.99
Blade height, h_b, cm	3.81
Axial chord, c_{xb}, cm	3.43
Axial solidity, $(c_{xb}/s_b)_m$	1.487
Aspect ratio, h_b/c_{xb}	1.111
Number of blades	64
Leading edge radius, cm	0.300
Trailing edge radius, cm	0 089

TABLE III. - AIRFOIL COORDINATES



Vane		
Point Z	X Y	5.4636 0.0890
φ	44°37'	
X	Y_L	Y_U
0	0.5080	0.5080
.128	------	.8510
.254	------	1.0058
.382	------	1.1226
.508	------	1.2142
.636	------	1.2878
.762	.0636	1.3462
.890	.1194	1.3920
1.016	.1652	1.4274
1.144	.2058	1.4492
1.270	.2414	1.4682
1.398	.2668	1.4732
1.524	.2922	1.4758
1.778	.3302	1.4630
2.032	.3606	1.4326
2.286	.3760	1.3818
2.540	.3836	1.3234
2.794	.3810	1.2574
3.048	.3734	1.1888
3.302	.3530	1.1100
3.556	.3302	1.0312
3.810	.2972	.9448
4.064	.2616	.8510
4.318	.2210	.7468
4.572	.1804	.6376
4.826	.1320	.5182
5.080	.0812	.3912
5.334	.0254	.2590
5.552	.0890	.0890

Blade		
Point Z	X Y	1.5440 0.7220
φ	24°	
X	Y_L	Y_U
0	0.2988	0.2988
.128	------	.6604
.254	------	.8458
.382	------	.9754
.508	.0838	1.0896
.762	.2642	1.2446
1.016	.4014	1.3310
1.270	.5030	1.3590
1.524	.5614	1.3386
1.778	.5842	1.2828
2.032	.5792	1.2014
2.286	.5436	1.0972
2.540	.4776	.9754
2.794	.3912	.8382
3.048	.2922	.6908
3.302	.1854	.5258
3.556	.0712	.3430
3.810	.0890	.0890

TABLE IV. - COMPARISON OF TWO TURBINES WITH HONEYCOMB SHROUDS

	Subject turbine	Starter turbine[a]	
	Honeycomb, full-depth	Honeycomb, full-depth	Honeycomb, half-depth
Rotor tip diameter, cm	50.8	13.5	13.5
Honeycomb			
Cell width, cm	0.081	0.160	0.160
Cell depth,[b] cm	0.572	0.380	0.190
Blade			
Span, cm	3.81	1.56	1.56
Cell depth/blade span, percent	15.0	24.4	12.2
Blade passing frequency, blades/sec	8400	21 700	21 700
Turbine efficiency:			
Solid shroud	0.860	0.868	0.868
Honeycomb shroud	0.837	0.786	0.850
Penalty, $\Delta\eta$	0.023	0.082	0.018

[a]From ref. 1.
[b]Includes supply manifold (see fig. 2).

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15. Supplementary Notes

16. Abstract

A 50.8-cm-tip-diameter turbine equipped with a rotor tip shroud of hexagonal cell (or honeycomb) cross section has been tested in warm air (416 K) for a range of shroud coolant to primary flow rates. Test results were also obtained for the same turbine operated with a solid shroud for comparison. The results showed that the combined effect of the honeycomb shroud and the coolant flow was to cause a reduction of 2.8 points in efficiency at design speed, pressure ratio, and coolant flow rate. With the coolant system inactivated, the honeycomb shroud caused a decrease in efficiency of 2.3 points. These results and those obtained from a small reference turbine indicate that the dominant factor governing honeycomb tip shroud loss is the ratio of honeycomb depth to blade span. The loss results of the two shrouds could be correlated on this basis. The same honeycomb and coolant effects are expected to occur for the hot (2200 K) version of this turbine.

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