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EXPERIMENTAL PERFORMANCE OF A 5-INCH SINGLE-STAGE AXIAL-FLOW TURBINE DESIGNED FOR A BRAYTON CYCLE SPACE POWER SYSTEM

by Donald E. Holeski and Samuel M. Futral, Jr.

Lewis Research Center
Cleveland, Ohio

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ABSTRACT

The turbine investigated was designed to drive a six-stage, axial-flow compressor for a 10-kilowatt, two-shaft space power system. Tests were made using argon at an inlet pressure of 2.9 psia (2.0 N/cm^2 abs) and an inlet temperature of 582° R (323 K). Test results showed that the turbine efficiency was below the design value. Data indicated the rotor tip sections performed poorly. The exit diffuser performed well.

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SUMMARY

A single-stage, axial-flow turbine was tested to determine its performance at design Reynolds number. This turbine was designed to drive a six-stage, axial-flow compressor for a 10-kilowatt, two-shaft space power system. The tests were conducted with an inlet pressure of 2.9 psia (2.0 N/cm^2 abs) and an inlet temperature of 582° R (323 K). Argon was used as the working fluid. Tests were conducted over a range of speeds and pressure ratios.

The results of these tests showed that the turbine performance was below that of design. The turbine static and total efficiencies (based on rotor exit conditions) were 0.77 and 0.83. The design static and total efficiencies were 0.796 and 0.849. The equivalent specific work of 10.78 Btu per pound (25.09 J/g) was 4 percent below the design value. The equivalent mass flow of 1.13 pounds per second (0.513 kg/sec) was 7 percent higher than the design value. Radial surveys of exit flow angle indicated the rotor tip section performed poorly. Computed values of diffuser effectiveness showed the diffuser to perform well.

INTRODUCTION

The importance of high component efficiencies in Brayton cycle space power systems has led to a technology program in turbomachinery components at the Lewis Research Center. This program includes the testing of turbines, compressors, and an alternator designed for the 10-kilowatt, two-shaft system described in reference 1. This system employs argon as the working fluid circulated by a high-speed turbocompressor and has a low-speed turboalternator to generate electrical power at 400 hertz.

Both axial- and radial-flow turbine-compressor combinations were designed for this application. Performance of the radial-flow components has been evaluated and published (refs. 2 and 3). A two-stage, axial-flow, alternator-drive turbine has also been evaluated and reported (ref. 4).

The subject single-stage, axial-flow turbine was designed to drive a 3.5-inch (8.9-cm), six-stage, axial-flow compressor at 50 000 rpm and a pressure ratio of 2.3. The turbine has high reaction in both blade rows and a hub-tip radius ratio of 0.6. It was tested in a cold-gas test facility to obtain performance characteristics in the design working fluid, argon. Tests were conducted with an inlet pressure of 2.9 psia (2.0 N/cm^2 abs) and an inlet temperature of 582° R (323 K). These conditions provide a Reynolds number near the design value.

Experimental results include efficiency, specific work, and mass flow as they vary with pressure ratio and shaft speed. Also, design aerodynamic and mechanical information is presented to describe the turbine.

SYMBOLS

Δh	specific work, Btu/lb (J/g)
N	turbine speed, rpm
p	absolute pressure, psia (N/cm^2 abs)
Re	Reynolds number, $w/\mu r_m$
r	radius, ft (m)
T	absolute temperature, $^\circ$R (K)
U	blade velocity, ft/sec (m/sec)
V	absolute gas velocity, ft/sec (m/sec)
V_j	ideal jet speed corresponding to total- to static- pressure ratio across turbine, ft/sec (m/sec)
W	relative gas velocity, ft/sec (m/sec)
w	mass flow rate, lb/sec (kg/sec)
α	absolute gas flow angle measured from axial direction, deg
γ	ratio of specific heats
δ	ratio of inlet total pressure to U.S. standard sea-level pressure

ϵ function of γ used in relating parameters to those using air inlet conditions at U.S. standard sea-level conditions,

$$\frac{\gamma^*}{\gamma}\left[\frac{\left(\frac{\gamma+1}{2}\right)^{\gamma/\gamma-1}}{\left(\frac{\gamma^*+1}{2}\right)^{\gamma^*/\gamma^*-1}}\right]$$

η static efficiency (based on inlet-total- to exit-static-pressure ratio)

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 temperature, $(V_{cr}/V^*_{cr})^2$

μ gas viscosity, lb/(ft)(sec) (N-sec/m^2)

ν blade-jet speed ratio, U_m/V_j

τ torque, in. -lb (N-m)

Subscripts:

cr condition corresponding to Mach 1

eq air equivalent (U.S. standard sea level)

m mean radius

1 station at turbine inlet

1a, 1b, 1c stations between turbine inlet and stator inlet (see fig. 3)

2 station at stator inlet

3 station at stator exit

4 station at rotor exit

4a, 4b, 4c stations between rotor exit and exhaust-pipe flange (see fig. 3)

5 station at exhaust-pipe flange

Superscripts:

' absolute total state

* U.S. standard sea-level conditions (temperature, 518.67° R (288.15 K); pressure, 14.696 psia (10.128 N/cm^2 abs))

TURBINE DESCRIPTION

As indicated in the INTRODUCTION, the single-stage, axial-flow turbine was designed to drive a six-stage, axial-flow compressor. The turbine and compressor are components of the 10-kilowatt, two-shaft space power system. The aerodynamic and mechanical design and fabrication of this turbine was performed by Pratt & Whitney Aircraft Company; reference 5 is the final report on the design. Photographs of the turbine are shown in figures 1 and 2. Figure 2 shows only 7 of the 30 stator blades required for assembly. A cross-sectional sketch of the turbine is shown in figure 3.

Listed in table I are the design-point values for the turbine. Values are tabulated for both argon and air as the working fluids. The air values are corrected to U. S. standard sea-level conditions.

The equivalent pressure ratios were calculated from equivalent specific work in air. The assumption was made that the efficiency would be the same for argon or air as the working fluid.

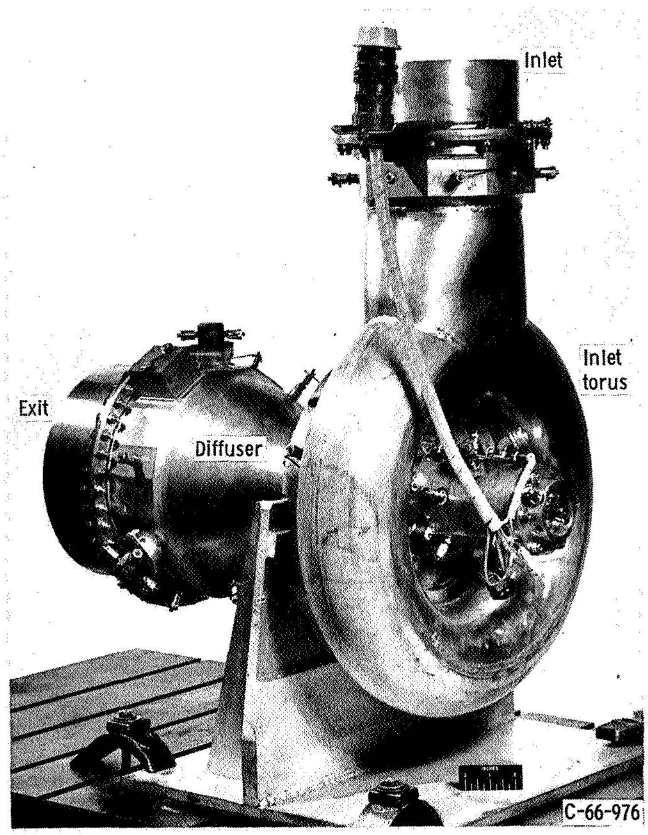


Figure 1. - Research turbine.

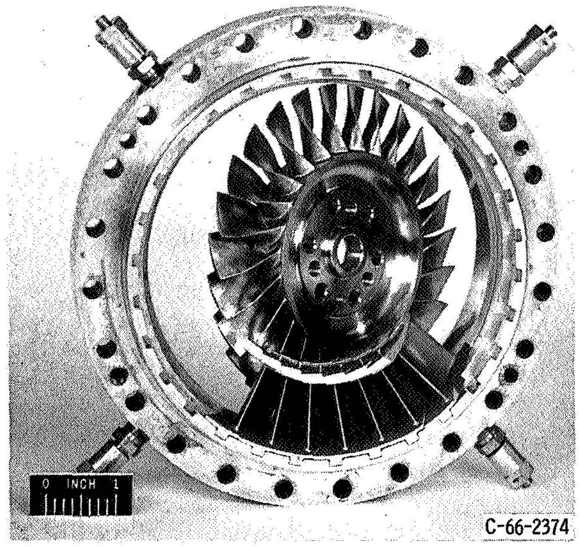


Figure 2. - Turbine stator and rotor. Rotor diameters: tip, 5.137 inches (13.05 cm); mean, 4.100 inches (10.41 cm); hub, 3.063 inches (7.780 cm). Stator diameters: tip, 5.137 inches (13.05 cm); mean, 4.150 inches (10.54 cm); hub, 3.163 inches (8.034 cm).

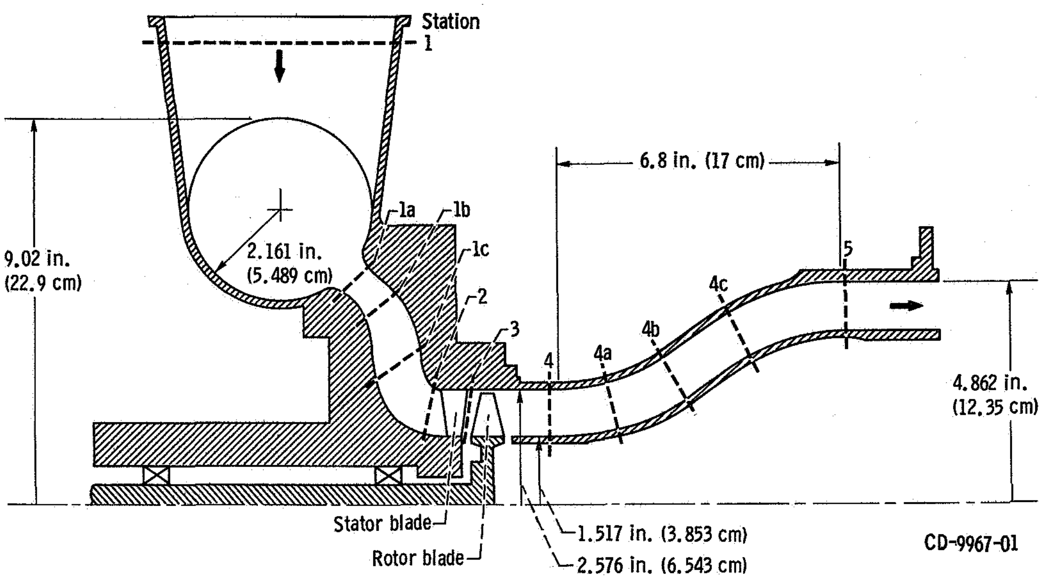


Figure 3. - Cross section of turbine.

TABLE I. - TURBINE DESIGN VALUES

Design parameters	Argon			Air equivalent (a)		
	Symbol	Value		Symbol	Value	
Inlet total temperature, °R (K)	T_1'	1950	(1083)	---------	------	------
Inlet total pressure, psia (N/cm^2 abs)	p_1'	13.20	(9.10)	---------	------	------
Mass flow rate, lb/sec (kg/sec)	w	0.611	(0.277)	$w\epsilon\sqrt{\theta_{cr}}/\delta$	1.06	(0.48)
Turbine speed, rpm	N	50 000		$N/\sqrt{\theta_{cr}}$	29 260	
Specific work, Btu/lb (J/g)	Δh	32.82	(76.34)	$\Delta h/\theta_{cr}$	11.24	(26.14)
Torque, in.-lb (N-m)	τ	35.8	(4.04)	$\tau\epsilon/\delta$	36.3	(4.10)
Ratio of inlet total pressure to rotor-exit total pressure	p_1'/p_4'	1.543		$\left(p_1'/p_4'\right)_{eq}$	1.482	
Ratio of inlet total pressure to rotor-exit static pressure	p_1'/p_4	1.592		$\left(p_1'/p_4\right)_{eq}$	1.524	
Ratio of inlet total pressure to diffuser-exit total pressure	p_1'/p_5'	1.549		$\left(p_1'/p_5'\right)_{eq}$	1.487	
Ratio of inlet total pressure to diffuser-exit static pressure	p_1'/p_5	1.557		$\left(p_1'/p_5\right)_{eq}$	1.494	
Rotor-exit total efficiency	η_{1-4}'	0.849		η_{1-4}'	0.849	
Rotor-exit static efficiency	η_{1-4}	0.796		η_{1-4}	0.796	
Overall total efficiency[b]	η_{1-5}'	0.842		η_{1-5}'	0.842	
Overall static efficiency[b]	η_{1-5}	0.833		η_{1-5}	0.833	
Reynolds number	Re	93 100		Re	93 100	
Ratio of blade speed to jet speed	ν	0.621		ν	0.621	

[a] U.S. standard sea-level conditions.

[b] Measured from torus-inlet flange to diffuser-exit flange.

Velocity Diagrams

The turbine design-velocity diagrams were computed on the basis of near free vortex flow and with a small variation of work from hub to tip. The loss coefficients used were adjusted for Reynolds number on the basis of a 1/5 power law. A tip leakage allowance of 3 percent of the flow was made in calculating the rotor exit total temperature. The design velocity diagrams are shown in figure 4 and are labeled in terms of critical-velocity ratios. A tabulation of velocities, in feet per second, with pressures and temperatures in the turbine is contained in table 2 of reference 5.

Stator Blades

The stator for this turbine was designed for conventional converging flow passages. Figure 2 shows the stator ring with some of the blades mounted. The profiles of the blades are shown in figure 4. The stator contained 30 blades. The hub, mean, and tip diameters are given in figure 2. The solidity (ratio of chord to spacing) at the mean radius is 1.621. Design blade surface velocities for hub, mean, and tip sections are shown in figures 5, 6, and 7 of reference 5.

Rotor Blades

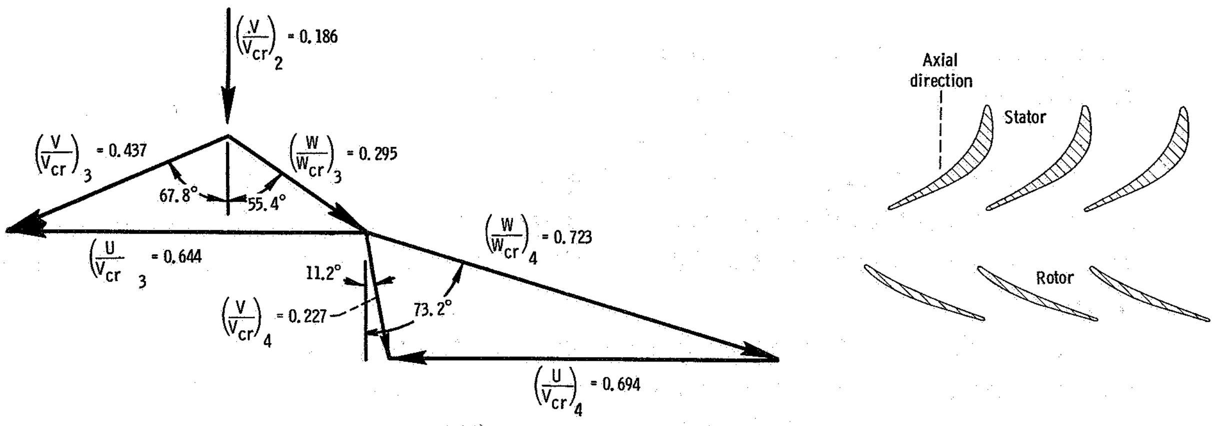
The rotor of this turbine had 26 blades with a solidity of 1.466 at the mean diameter. The hub, mean, and tip diameters are given in figure 2. The blades had considerable twist, as seen in figure 4, which shows the blade profiles for hub, mean, and tip sections. The twist is also noticeable in the photograph of figure 2. The design surface velocity distribution for the blading is given in figures 9 and 10 of reference 5.

Turbine Assembly

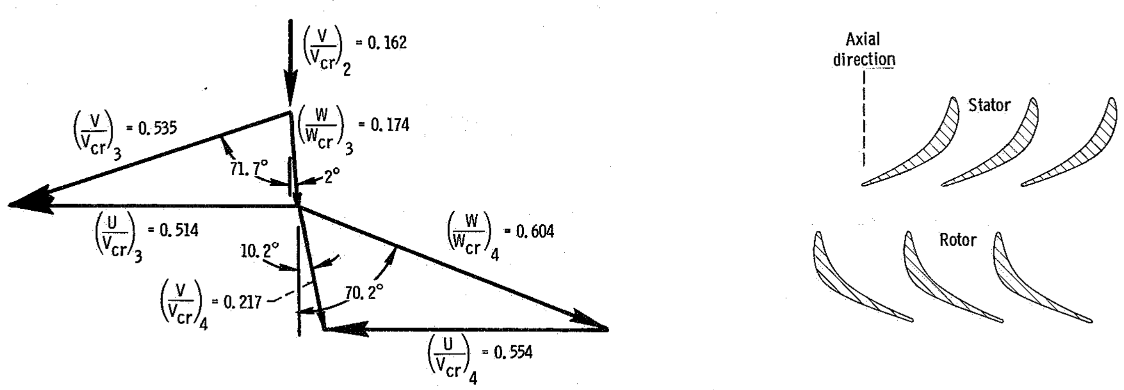
A cross-sectional view of the turbine flow passages is shown in figure 3. The station numbers, as well as some of the major dimensions of the turbine are given in the figure.

The rotor blade height was 1.037 inches (2.634 cm) with an average tip clearance of 0.012 inch (0.030 cm) or about 1 percent of the blade height.

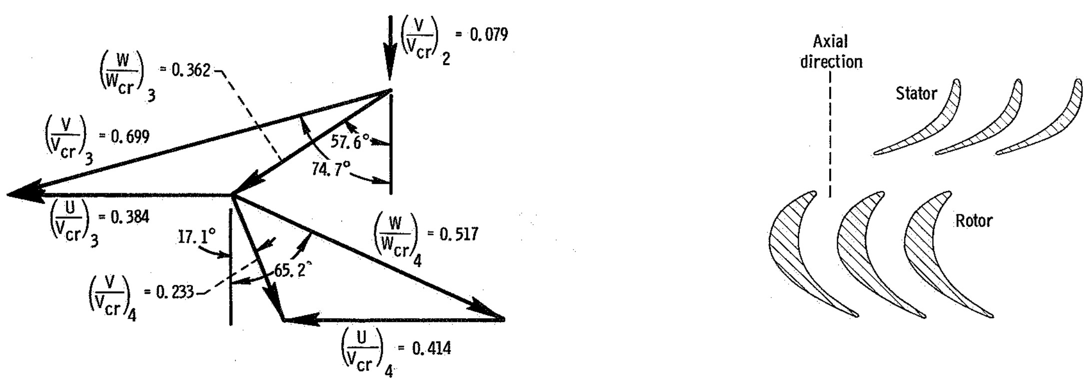
The shaft was supported by a ball bearing on the coupling end and a roller bearing on the rotor end. The bearing housing was sealed by a carbon face seal on the rotor end



(a) Tip.



(b) Mean.



(c) Hub.

Figure 4. - Design velocity diagrams and blade profiles.

and a labyrinth seal on the coupling end. Lubrication was accomplished by means of an air-oil mist system. This system was used because of the reduced bearing drag as compared to a solid-jet oil system.

The stator throat, or minimum flow area, was 4.50 square inches (29.0 cm^2), which was 1.4 percent larger than the design area. The rotor throat, or minimum flow area, was 4.67 square inches (30.1 cm^2), which was 3 percent larger than the design value.

The diffuser was designed to be a transition piece connecting two turbines. Therefore, its entrance and exit diameters were fixed. Entrance and exit diameters are shown in figure 3. There is a 1.2-inch (3.0-cm) straight section at the diffuser entrance and a 2-inch (5-cm) straight section at the exit. The middle section, where the area varies, is about 6.8 inches (17 cm) long. The diffuser was designed to diffuse at a rate consistent with a 4° half-angle conical diffuser. The annular entrance area was 13.28 square inches (85.68 cm^2) and the annular exit area was 32.26 square inches (208.1 cm^2).

APPARATUS

The apparatus consisted of the turbine (described in the preceding section), an airbrake dynamometer, and an inlet and exhaust piping system. The arrangement of the apparatus is shown schematically in figure 5. Pressurized argon was used as the

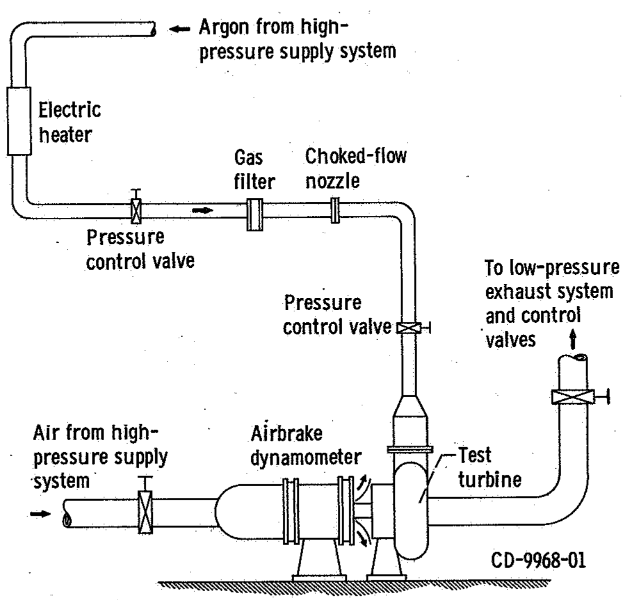


Figure 5. - Experimental equipment.

driving fluid for the turbine. The argon was heated by an electric heater and was filtered. The argon then passed through a weight-flow measuring station that consisted of a calibrated choked-flow nozzle. The argon, after passing through the turbine, was exhausted into the laboratory exhaust system. A pressure-control value upstream of the turbine regulated the turbine-inlet pressure. With a fixed inlet pressure, a remotely controlled valve in the exhaust line was used to obtain the desired pressure ratio across the turbine.

The airbrake dynamometer was used to absorb and measure the power output of the turbine. The dynamometer was cradle mounted on air bearings for torque measurements. The turbine torque was obtained by measuring the force on the dynamometer torque arm. A strain-gage-type load cell measured the force. An electronic counter in conjunction with a magnetic pickup and a shaft-mounted gear measured the turbine speed. The dynamometer also controlled the turbine speed.

INSTRUMENTATION

The instrument measuring stations are shown in figure 3. The instrumentation at station 1 consisted of 4 static-pressure taps and one total-temperature rake with three thermocouples. In the radial section between the torus and stator inlet there were 10 static-pressure taps. These taps were spaced at 1.5-inch (3.8-cm) intervals in two rows approximately 180° apart and are labeled station 1a, 1b, and 1c. At station 2 (stator inlet), there were four static taps in the inner wall and four in the outer wall. Downstream of the stator, at station 3, there were four static taps in a cavity forward of the rotor and four in the outer wall. At station 4, immediately downstream of the rotor, the instrumentation consisted of eight static-pressure taps (four each at inner and outer walls) and a self-alining probe for flow angle and total pressure measurements. In the diffuser were 16 static taps spaced at 2-inch (5.1-cm) intervals in two rows approximately 180° apart. These taps are labeled 4a, 4b, and 4c. At station 5 the instrumentation was the same as at station 4. Static taps were manifolded wherever the pressures were thought to be the same.

All turbine pressures were measured by manometers and recorded by hand. These manometers contained a fluid with a specific gravity near 1 and a low vapor pressure at room temperature. The manometers were calibrated to an accuracy of 0.002 psia (0.001 N/cm^2 abs). Pressure measurements were taken with the reference side of each manometer evacuated to a pressure of 10 millitorr (1.33 N/m^2).

All data except manometer data were recorded on an integrating digital data recorder. A digital computer processed all data.

PROCEDURE

In the turbine test program, constant-speed runs were made at speeds of 30, 50, 70, 90, 100, and 110 percent of design speed. Argon was used as the working fluid. For all speeds, the inlet total pressure was held at 2.9 psia (2.0 N/cm^2 abs), and the inlet total temperature was held at 582° R (323 K). These inlet conditions correspond to a Reynolds number of about 95 000 at the design operating point. The Reynolds number as used in this report is defined as $Re = w/\mu r_m$, where w is the turbine mass flow, μ in the gas viscosity at the turbine-inlet total conditions, and r_m is the mean radius of the rotor. The exit pressure was varied to obtain ratios of inlet total pressure to rotor-exit static pressure $\left(p_1'/p_4\right)_{eq}$ from 1.2 to 2.3. Inlet total pressure was calculated from measured static pressure, total temperature, area, and mass flow rate. Data resulting from the measurements were used to calculate the turbine efficiencies before and after diffusion.

Bearing and seal friction torque was measured and added to shaft torque for the determination of turbine performance. The friction torque was obtained by measuring the amount of torque required to rotate the shaft and rotor over the speed range. Rotor windage losses were ignored during friction tests because the turbine was evacuated to a pressure of 150 millitorr (20 N/m^2). A friction torque value of 0.74 inch-pound (0.08 N-m) was obtained at equivalent design speed. This is 9 percent of the measured turbine torque at inlet total pressure of 2.9 psia (2.0 N/cm^2 abs).

Blade-jet speed ratio, used in the presentation of results, was calculated in all cases from the ratio of turbine-inlet total pressure to the rotor-exit static pressure $\left(p_1'/p_4\right)_{eq}$

Turbine performance was based on measurements taken at the torus inlet (station 1) and the rotor exit (station 4). Overall performance for the turbine with diffuser was based on measurements taken at the torus inlet (station 1) and the diffuser exit (station 5). It should be noted that values presented in the design report of Pratt & Whitney Aircraft Company, reference 5, were taken from stator inlet (station 2) to rotor exit (station 4).

RESULTS AND DISCUSSION

The design-point performance results from tests in terms of air-equivalent values are given in table II. Also shown in table II are the values used in the turbine design. A comparison of these values shows that the turbine did not perform as well as was expected. While the mass flow rate was higher than design value, the efficiency was below that designed for.

Design operating point as used in the following discussion is defined as turbine

TABLE II. - AIR-EQUIVALENT PERFORMANCE VALUES

Performance parameter	Design	Experimental
Mass flow rate, $w\epsilon\sqrt{\theta_{cr}}/\delta$, lb/sec (kg/sec)	1.06 (0.481)	1.13 (0.513)
Specific work, $\Delta h/\theta_{cr}$, Btu/lb (J/g)	11.24 (26.14)	10.78 (25.09)
Torque, $\tau\epsilon/\delta$, in.-lb (N-m)	36.3 (4.10)	37.1 (4.19)
Rotor-exit total efficiency, η'_{1-4}	0.849	0.83
Rotor-exit static efficiency, η_{1-4}	0.796	0.77
Overall total efficiency[a], η'_{1-5}	0.842	0.82
Overall static efficiency[a], η_{1-5}	0.833	0.81
Reynolds number, Re	93 100	95 000

[a]Measured from torus-inlet flange to diffuser-exit flange.

operation at equivalent speed $N/\sqrt{\theta_{cr}}$ of 29 260 rpm and equivalent pressure ratio $\left(p_1'/p_4\right)_{eq}$ of 1.524.

Mass Flow

The mass flow characteristics of the turbine are shown in figure 6. Equivalent mass flow $w\epsilon\sqrt{\theta_{cr}}/\delta$ is plotted against equivalent exit-static- to inlet-total-pressure

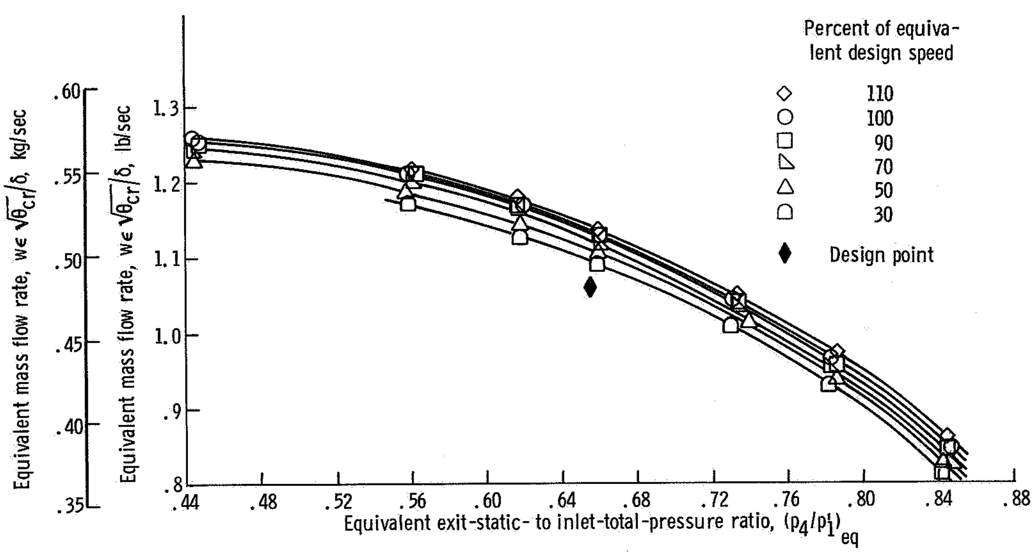


Figure 6. - Variation of mass flow rate with pressure ratio and speed.

ratio $\left(p_4/p'_1\right)_{eq}$ for lines of constant speed. At design pressure ratio of 0.656 the mass flow rate was 1.13 pounds per second (0.513 kg/sec). This value is 7 percent higher than the design value of 1.06 pounds per second (0.481 kg/sec). Part of this excess flow may be attributed to larger-than-design passages in both the stator and rotor. These areas were mentioned in the TURBINE DESCRIPTION section. The passage-to-passage variation in the stator flow area was large - approximately 11 percent of the average passage area.

The mass flow curves show a decrease in mass flow with decreasing speed at constant pressure ratio. The variation may be caused by changes in rotor incidence losses. Figure 6 also shows the turbine was nearing choked conditions at the minimum pressure ratio $\left(p_4/p'_1\right)_{eq}$ of 0.445.

Torque and Specific Work

Variation of torque with speed for lines of constant pressure ratio is shown in figure 7. Each pressure-ratio curve shows a linear variation of torque from zero speed to 110 percent of design speed. At the design operating point an equivalent torque $\tau\epsilon/\delta$ of 37.1 inch-pounds (4.19 N-m) was obtained. The value is 2 percent higher than the design value of 36.3 inch-pounds (4.10 N-m). This results from the excessive mass flow predominating over the lower-than-design efficiency (as will be discussed later). This value of equivalent torque is a faired value and was obtained by cross-plotting the data of figure 7. This was done in order to obtain the equivalent torque at exactly design pressure ratio and equivalent speed. Figure 7 also shows the torque values at zero

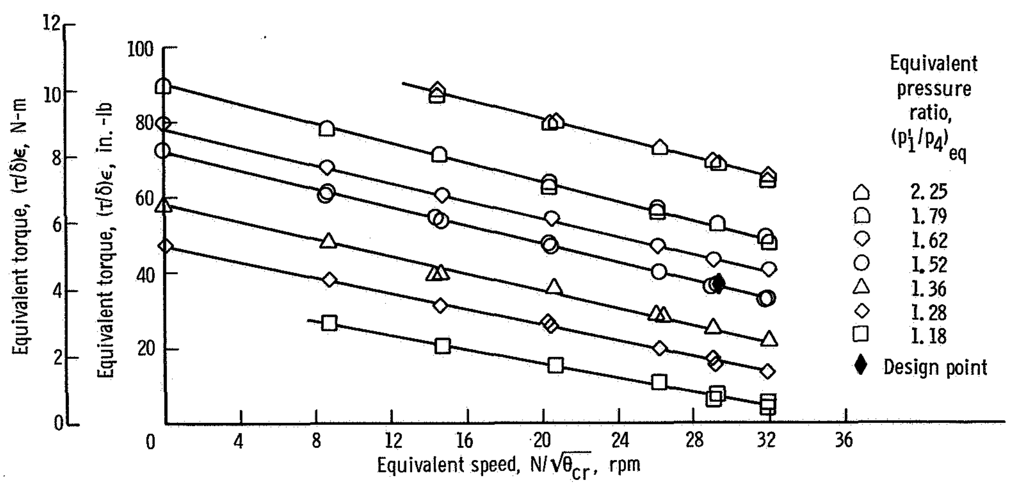


Figure 7. - Variation of torque with speed and pressure ratio.

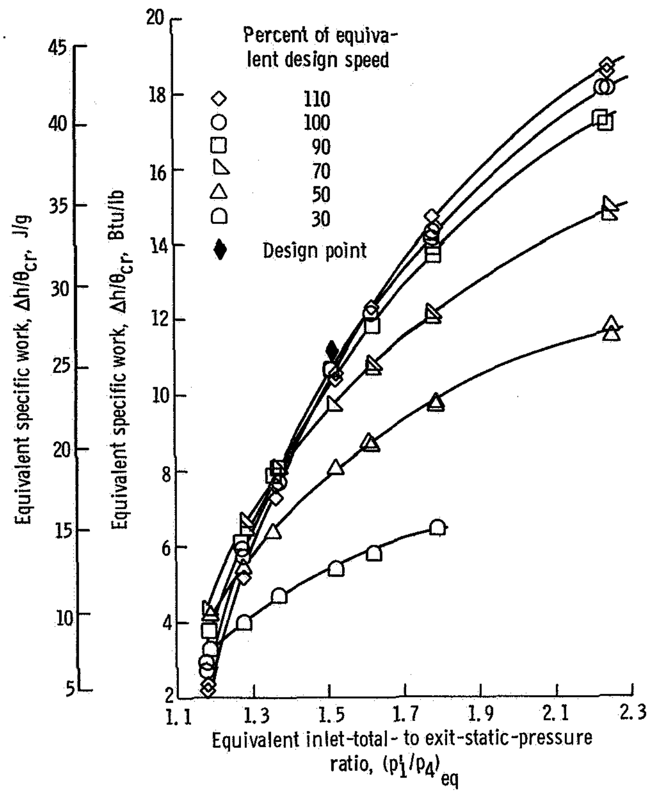


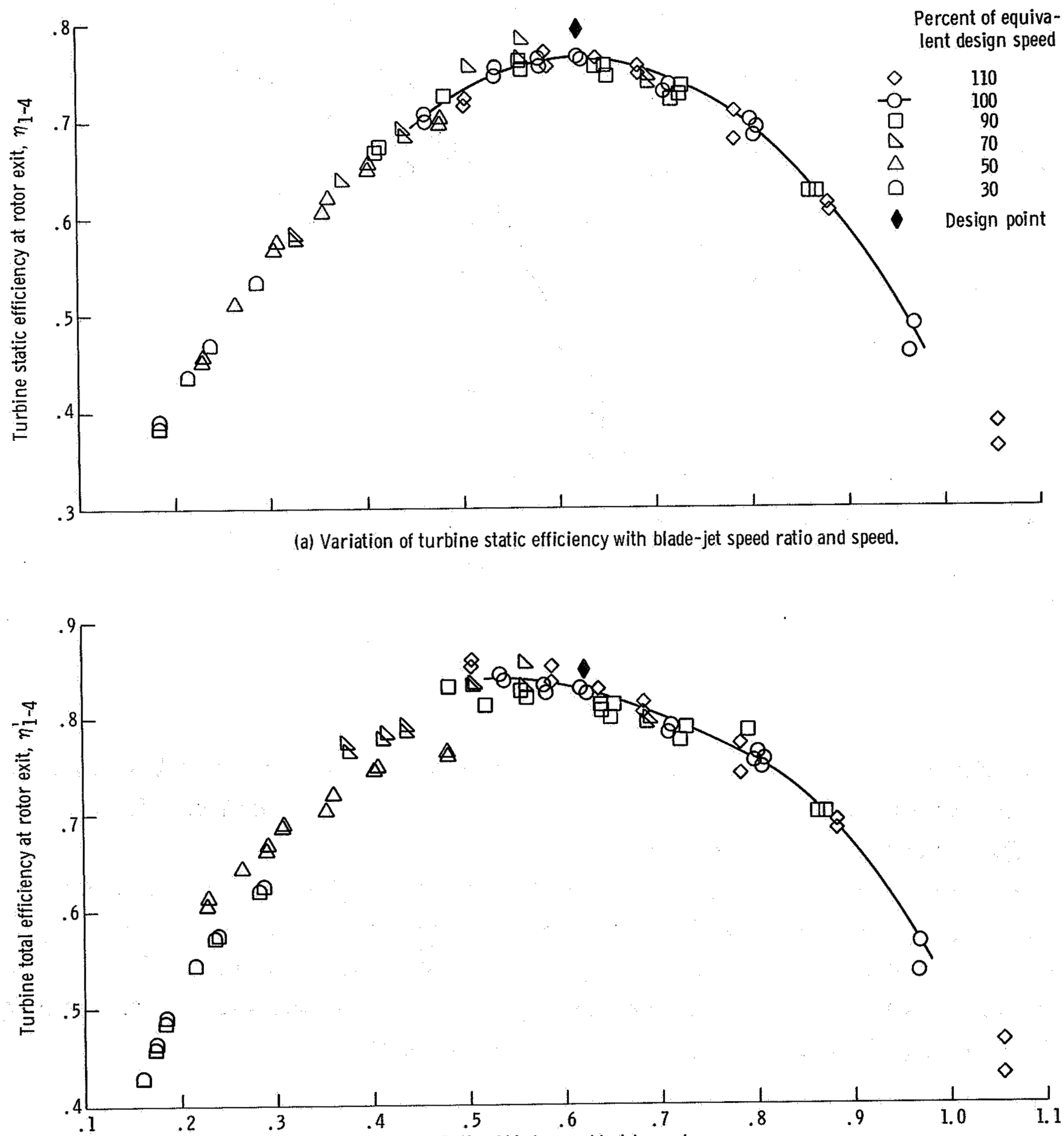
Figure 8. - Variation of equivalent specific work with pressure ratio and speed.

speed. At design pressure ratio the zero-speed torque is about 2 times the value obtained at equivalent design speed. This is reasonable for axial turbines with design velocity diagrams of the type used for this turbine.

The variation of specific work with pressure ratio for lines of constant speed is shown in figure 8. At the design operating point an equivalent specific work $\Delta h/\theta_{cr}$ of 10.78 Btu per pound (25.09 J/g) was obtained. This value is 4 percent lower than the design value of 11.24 Btu per pound (26.14 J/g). Figure 8 shows that limiting loading was not reached at the highest pressure ratios investigated.

Efficiency

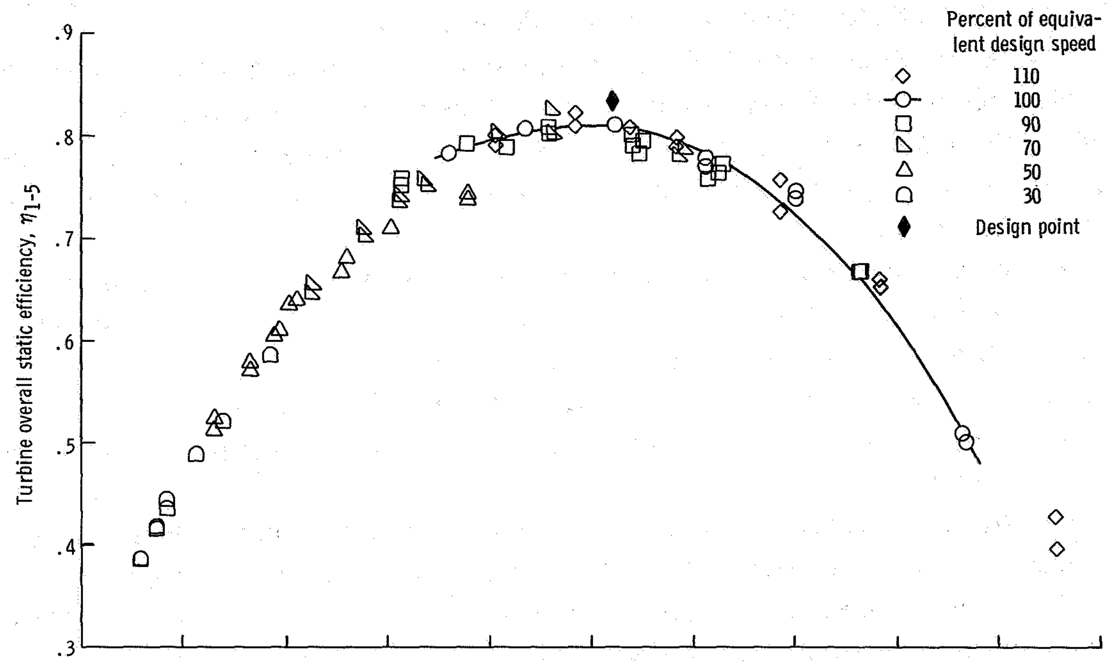
Turbine efficiencies over the range of blade-jet speed ratios investigated are shown in figures 9 and 10. Figure 9 presents static and total efficiencies based on rotor-exit conditions. At design blade-jet speed ratio ($\nu = 0.621$), the static efficiency



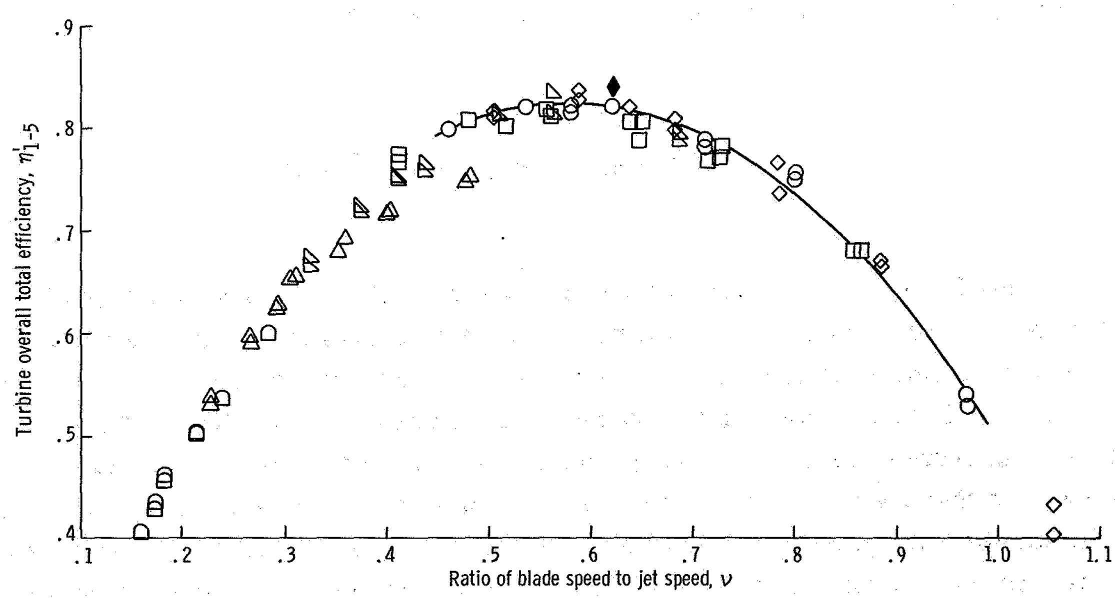
(a) Variation of turbine static efficiency with blade-jet speed ratio and speed.

(b) Variation of total efficiency with blade-jet speed ratio and speed.

Figure 9. - Performance characteristics at rotor exit.



(a) Variation of turbine overall static efficiency with blade-jet speed ratio and speed.



(b) Variation of turbine overall total efficiency with blade-jet speed ratio and speed.

Figure 10. - Overall performance characteristics.

η_{1-4} obtained was 0.77. This value is about 3 percentage points below the value of 0.796 assumed in the design. The total efficiency η'_{1-4} at design blade-jet speed ratio is 0.83. This is also below the design value of 0.849.

Figure 10 presents static and total efficiencies based on diffuser-exit conditions. At design blade-jet speed ratio, the static efficiency η_{1-5} obtained was 0.81. This value is about 2 percentage points below the value of 0.833 assumed in the design. The total efficiency η'_{1-5} at design blade-jet speed ratio is 0.82. This is also below the design value of 0.842.

At the rotor exit there is a difference between static and total efficiency of 6 percentage points. This difference represents the kinetic energy level at the rotor exit and agrees with the design values. At the diffuser exit the difference in static and total efficiency is 1 percentage point. This also agrees with design values. Also, at design blade-jet speed ratio the overall total efficiency is 1 percentage point lower than that before diffusion.

The diffuser performance also can be represented in terms of pressure recovery. A parameter commonly used is the ratio of static-pressure rise through the diffuser to the diffuser-inlet impact pressure and is called diffuser effectiveness. A diffuser effectiveness of 0.70 was computed from the experimental data. This compares favorably with the design value of 0.71. An ideal diffuser effectiveness for isentropic incompressible diffusion with the same area ratio is 0.83.

Internal and Exit Flow Conditions

The turbine was also instrumented to measure static pressure in the inlet radial section and in the exit diffuser. Although this instrumentation is not needed to determine turbine performance, it is of general interest. Figure 3 shows the location of these additional static taps, which have already been discussed in the INSTRUMENTATION section. At design operating point, pressure measurements were made with all taps. The ratio of these static pressures to the inlet total pressure is plotted as a function of station number in figure 11. The figure shows that there is a small pressure drop in the inlet radial section, as had been assumed in the design. The figure also shows the static-pressure recovery that occurs in the exit diffuser. The difference in inner-wall and outer-wall static pressure is small.

Radial surveys of flow angle and total pressure were taken at design speed and pressure ratio. Figure 12 presents radial survey results at the rotor exit. Figure 12(a) shows the variation in flow angle with radius ratio. Symbols are shown which represent

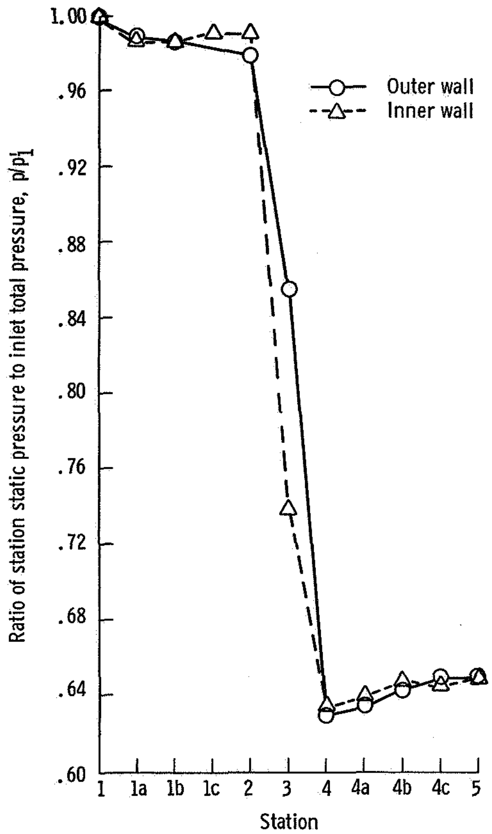
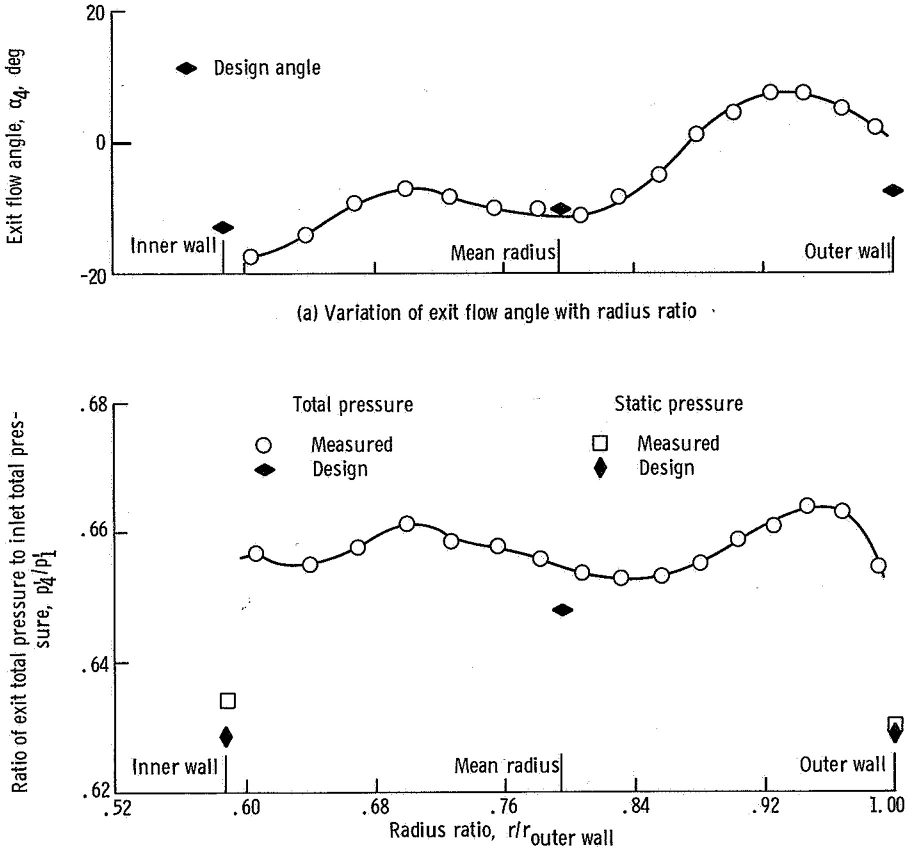


Figure 11. - Variation of static pressure through turbine at design speed.

design flow-angle variation. The curve shows a large change in flow angle occurring in the tip region of the rotor. Figure 12(b) shows the variation in total pressure with radius ratio. The measured values of total pressure were divided by the turbine-inlet total pressure in order to minimize any effects of changes in turbine-inlet total pressure during the tests. Also shown is a symbol representing the average design total pressure at this station. Symbols are also shown for design and measured wall static pressures. The figure shows that the largest variation in measured total pressure occurs near the rotor tip. Calculations of dynamic head across the passage indicated that the rotor-exit tip velocity was high. Figure 12(b) also shows rotor-exit flow angle at the tip was far from design. From this it can be concluded that the tip portion of the rotor performed poorly and contributed to the poor turbine performance that was measured.

A radial survey of flow angle and total pressure taken at the diffuser exit verified the data taken at the rotor exit. The data showed the flow angle to vary from about -17° at the inner wall to $+7^{\circ}$ at the outer wall. The design flow angle varied from -14° at the inner wall to -10° at the outer wall. The design values at the diffuser exit were



(a) Variation of exit flow angle with radius ratio

(b) Variation of total pressure with radius ratio.

Figure 12. - Survey results at rotor exit at design equivalent speed and pressure ratio.

obtained from reference 4 and are the same as the inlet values for the two-stage, axial-flow turbine. The measured flow-angle variation is large compared to the design values. Data also showed the measured radial variation of total pressure to be considerably less than that at the rotor exit. Thus, by decelerating the flow, the diffuser reduced the pressure variation that was present at the diffuser inlet; this is caused by the low velocity level in the exit of the diffuser due to the area increase and by the total pressure loss in the diffuser.

SUMMARY OF RESULTS

A single-stage, axial-flow turbine was tested to determine its performance at design Reynolds number. This turbine was designed to drive a six-stage, axial-flow compressor for a two-shaft, 10-kilowatt space power system. Results of these tests showed that the turbine did not perform as well as was expected. The results are summarized as follows.

1. An equivalent specific work of 10.78 Btu per pound (25.09 J/g) was obtained at design operating point. This is 4 percent below design value.

2. The turbine static and total efficiencies (based on rotor-exit conditions) were measured to be 0.77 and 0.83. The design static and total efficiencies were 0.796 and 0.849.

3. An equivalent mass flow rate of 1.13 pounds per second (0.513 kg/sec) was obtained at design operating point. This is 7 percent higher than the design value.

4. Radial survey results of flow angle and total pressure at the turbine exit showed the rotor tip section to perform poorly.

5. The ratio of diffuser static-pressure recovery to diffuser-inlet impact pressure was 0.70. This value compares favorably with the design value of 0.71.

Lewis Research Center,
National Aeronautics and Space Administration,
Cleveland, Ohio, May 28, 1968,
129-01-05-19-22.

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