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PERFORMANCE EVALUATION OF A TWO-STAGE AXIAL-FLOW TURBINE FOR TWO VALUES OF TIP CLEARANCE

by Milton G. Kofskey and William J. Nusbaum

Lewis Research Center
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SUMMARY

An experimental investigation was made of a two-stage turbine designed to drive an alternator for a 10-kilowatt-shaft-output space power system. Performance results were obtained for the turbine operating with the rotor tip clearance of 0.031 inch (0.079 cm) recommended for preliminary hot operation of the turbine. These results are compared with the results obtained with the design tip clearance of 0.013 inch (0.033 cm). Tests were made with cold argon as the working fluid over a speed range from 0 to 120 percent of design equivalent speed and at pressure ratios ranging from 1.08 to 1.55.

The results of the investigation indicated that the static and total efficiencies (based on turbine-inlet and collector-exit conditions) were 0.785 and 0.792, respectively, for a tip clearance of 0.031 inch (0.079 cm). These values represent a 4-percentage-point decrease in turbine efficiency when the tip clearance is increased.

This decrease in efficiency may be attributed to such factors as rotor blade tip unloading, increased throughflow over the blade tips in the clearance space, and the reduction of working blade area. The equivalent mass flow was 1.2 percent greater, and the equivalent torque was 4 percent lower when the tip clearance was increased.

The results based on two tip clearances indicated that the subject turbine, which was designed with high rotor reaction, was very sensitive to changes in tip clearance. There was a 3.7-percent decrease in turbine specific work for an increase in tip clearance of 1.0 percent of passage height. This decrease is substantially greater than the 1.75-percent decrease in turbine work obtained for a reference turbine of impulse design.

INTRODUCTION

In a current program at the NASA Lewis Research Center, the components of an exploratory Brayton-cycle space power system are being investigated. One of these components is a two-stage axial-flow turbine designed and built under contract to drive the alternator of a two-shaft, 10-kilowatt-shaft-output system which uses argon as the working fluid. The results of a cold-gas performance evaluation of the turbine are presented in reference 1. These tests were made with the design rotor blade tip clearance of 0.013 inch (0.033 cm). However, for preliminary hot operation (inlet total temperature, 1685° R (936.1° K)) of the turbine with gas bearings and with the alternator, an average tip clearance value of 0.031 inch (0.079 cm) was recommended by the contractor. This value represents an increase in tip clearance of about 1.5 percent of the annular passage height.

Results of previous tip-clearance investigations indicated that increases in clearance such as that just described can have a significant effect on the turbine efficiency. For example, the study of reference 2 indicated a 1.75-percent reduction in turbine efficiency when rotor blade tip clearance was increased 1 percent of annular passage height. This turbine operated under impulse conditions, whereas the subject turbine was designed with a large amount of rotor reaction in both stages. Accordingly, it might be expected that the tip-clearance effects would be greater for the subject turbine.

In view of the foregoing discussion and the objective of high efficiency for the intended application, it was considered important to determine the effect of increased tip clearance on the two-stage turbine performance. Accordingly, the investigation of reference 1 was extended to evaluate its performance at the larger tip clearance of 0.031 inch (0.079 cm). In addition, the sensitivity of performance to tip clearance could be determined by comparison with the results obtained at the small tip clearance.

Tests were made with argon at a turbine-inlet total temperature of 610° R (339° K) and an absolute inlet total pressure of 2.50 psi (1.723 N/cm^2). Data were obtained over a range of total- to static-pressure ratios of 1.08 to 1.55 and a range of speeds from 0 to 120 percent of equivalent design speed. The results of the investigation are presented in terms of mass flow, torque, and efficiency. These results are then compared with those obtained when operating the turbine with a rotor tip clearance of 0.013 inch (0.033 cm), as reported in reference 1. The effect of increased tip clearance on the performance of the subject turbine is then compared with that for an impulse-type turbine (ref. 2).

SYMBOLS

Δh specific work, Btu/lb (J/g)

N turbine speed, rpm

p	absolute pressure, psi (N/cm^2)
Re	Reynolds number, $w/\mu r_m$
r	radius, ft (m)
T	absolute temperature, ^oR (^oK)
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, (p_1'/p_5), ft/sec (m/sec)
W	relative gas velocity, ft/sec (m/sec)
w	mass flow, 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 $\gamma^*/\gamma \left[\left(\frac{\gamma+1}{2}\right)^{\gamma/(\gamma-1)} \Big/ \left(\frac{\gamma^*+1}{2}\right)^{\gamma^*/(\gamma^*-1)}\right]$
η_s	static efficiency (based on inlet-total- to exit-static-pressure ratio)
η_t	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) (kg/(m)(sec))
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)
id	ideal
m	mean radius
1	station at turbine inlet
2	station at first-stage stator exit

3 station at first-stage rotor exit

4 station at second-stage stator exit

5 station at second-stage rotor exit

6 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/cm² abs)

TURBINE DESCRIPTION

The two-stage axial-flow turbine was designed to drive the low-speed alternator of a 10-kilowatt-shaft-output space power system. Design-point values with argon as the working fluid, as well as the air-equivalent values are as follows:

Design point (argon):	
Inlet total temperature, T_1', °R (°K)	1685 (936.11)
Inlet total pressure, p_1', psia (N/cm² abs)	8.45 (4.826)
Mass flow, w, lb/sec (kg/sec)	0.611 (0.277)
Turbine rotative speed, N, rpm	12 000
Total- to static-pressure ratio:	
Overall, p_1'/p_6	1.2540
Rotor exit, p_1'/p_5	1.2542
Total- to total-pressure ratio:	
Overall, p_1'/p_6'	1.2495
Rotor exit, p_1'/p_5'	1.2469
Blade-jet speed ratio, ν	0.465
Total to static efficiency, η_s:	
Overall	0.826
Rotor exit	0.825
Total to total efficiency, η_t:	
Overall	0.843
Rotor exit	0.850
Specific work, Δh, Btu/lb (J/g)	15.04 (35.010)
Reynolds number, $Re = w/\mu r_m$	49 500
Air equivalent:	
Mass flow, $\epsilon w \sqrt{\theta_{cr}}/\delta$, lb/sec (kg/sec)	1.537 (0.697)
Specific work, $\Delta h/\theta_{cr}$, Btu/lb (J/g)	5.96 (13.874)
Torque, $\tau\epsilon/\delta$, in.-lb (N-m)	108.12 (12.216)
Rotative speed, $N/\sqrt{\theta_{cr}}$, rpm	7 554
Total- to total-pressure ratio:	
Overall, $p_1'/p_{6,eq}'$	1.226
Rotor exit, $p_1'/p_{5,eq}'$	1.225
Total- to static-pressure ratio:	
Overall, $p_1'/p_{6,eq}$	1.232
Rotor exit, $p_1'/p_{5,eq}$	1.233
Blade-jet speed ratio, ν	0.465

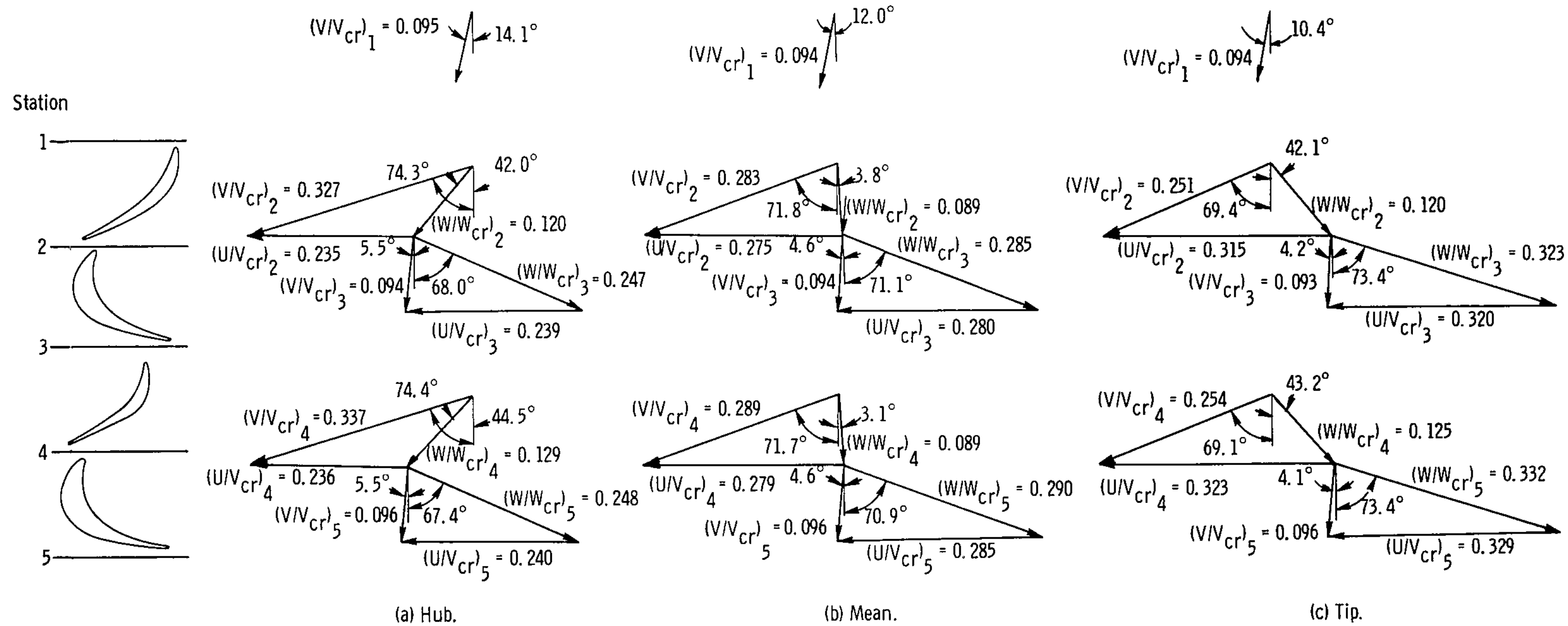

Figure 1. - Design velocity diagrams.

A detailed description of the aerodynamic and mechanical design of this turbine is given in reference 3.

Velocity Diagrams

Design conditions of pressure, temperature, weight flow, and speed were used to calculate velocity diagrams to meet the design work requirement. There was assumed to be an equal work split between the two stages. Effects of both low Reynolds number and leakage over the rotor tips were included in the estimated losses through the blade rows. Velocity diagrams, shown in figure 1, were calculated at the hub, mean, and tip diameters. The diagrams indicate the turbine to be of low subsonic design with a small amount of turbine-exit whirl in the direction of rotation. Gas flow turning at the mean diameter is 67.3° and 67.8° for the first- and second-stage rotors, respectively. The velocity diagrams also show, that there is a large amount of rotor reaction in both stages.

Stator Blades

There were 44 stator blades in the first stage and 40 stator blades in the second stage. The solidity at the mean diameter was 1.57 and 1.56 for the first and second stages, respectively. The stators were designed with comparatively high convergence of the flow passage from the inlet to the throat. (Blade surface velocities are given in ref. 1.) Essentially the blades of both stators have a small amount of diffusion on the suction surface downstream of the throat and on the pressure surface near the leading edge.

Rotor Blades

The design included 36 rotor blades in each of the two stages, with tip diameters of 9.7 and 9.8 inches (24.6 and 24.9 cm) for the first and second stages, respectively. The low solidity of 1.2 at the tip diameters of both rotors resulted in a very short guided channel. Figure 2 shows the large amount of rotor blade twist required by the design velocity diagrams. The blades of both rotors have a small amount of diffusion on the suction surface downstream of the throat and on the pressure surface near the leading edge.

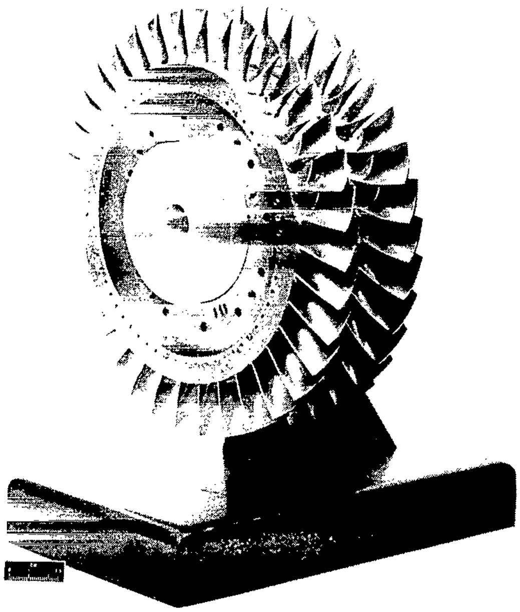

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Figure 2. - Two-stage turbine rotor assembly.

Turbine Assembly

In figure 3, a cross section of the turbine is presented schematically. The major dimensions of the turbine are given, together with the arrangement of the blading. The inlet guide vanes were designed to give a prerotation of the flow of approximately 12° from axial at mean diameter. This prerotation simulates the exit flow conditions from the compressor-drive turbine for the system. Figure 3 shows the sharp radially outward turn in the passage of the exhaust collector. Rotor blade tip clearances were initially 0.012 and 0.015 inch (0.030 and 0.038 cm) for the first and second stages, respectively. These clearances are referred to hereinafter as the 0.013-inch (0.033-cm) tip clearance. The tip clearance was then increased to an average value of 0.031 inch (0.079 cm) by grinding material from the rotor blade tips. This larger tip clearance was the operating clearance recommended by the contractor for preliminary hot operation in the power generation system. These tip clearances correspond to 1.06 and 2.47 percent of the average annular passage height for the respective tip clearances of 0.013 inch (0.033 cm) and 0.031 inch (0.079 cm).

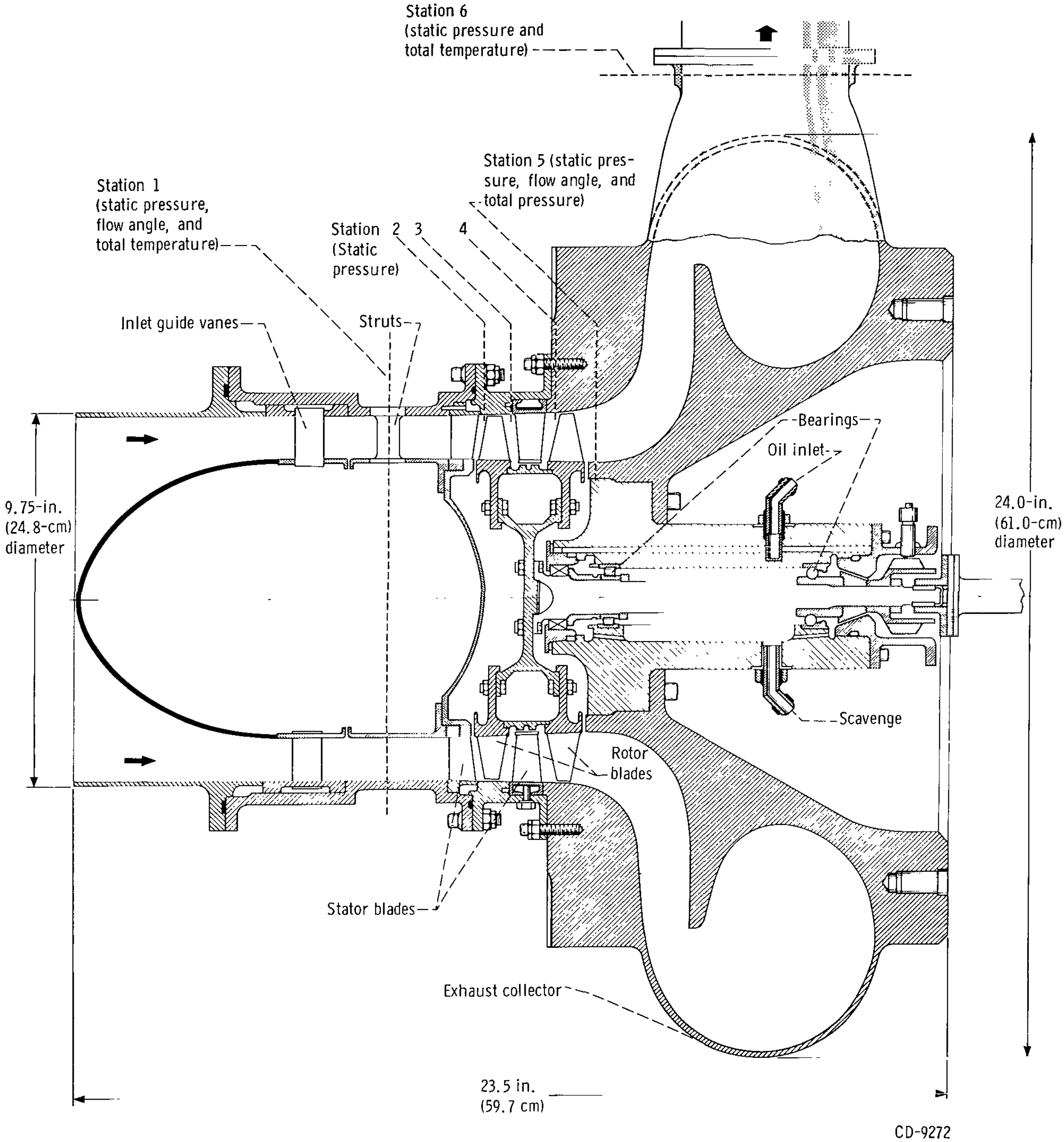


Figure 3. - Cross section of turbine.

APPARATUS, INSTRUMENTATION, AND PROCEDURE

The apparatus consisted of the turbine described in the preceding section, an airbrake dynamometer to absorb and measure the power output of the turbine, and an inlet and exhaust piping system with flow controls. The arrangement of the apparatus is shown schematically in figure 4. Pressurized argon was used as the driving fluid for the turbine. Argon was passed into the turbine through an electric heater, a filter, a weight-flow measuring station, and a remotely controlled pressure-regulating valve.

The airbrake dynamometer was used to absorb and measure the power output of the turbine and, at the same time, control the speed. The torque force was measured with a commercial strain-gage load cell. The rotational speed was measured with an electronic counter in conjunction with a magnetic pickup and a shaft-mounted gear.

The instrument measuring stations are shown in figure 3. The following instrumentation was included at the turbine inlet (station 1): eight static pressure taps (four on each of the inner and outer walls), a self-alining probe for the flow angle measurement; two total temperature rakes (each containing three thermocouples). At station 5, immediately downstream of the second-stage-rotor trailing edge, the instrumentation consisted of eight static pressure taps (four each at the inner and outer walls) and a self-alining probe to measure flow angle, total temperature, and total pressure. Four static pressure taps and two total temperature rakes were used at station 6. The temperature rakes were used to determine turbine total efficiency by temperature drop across the

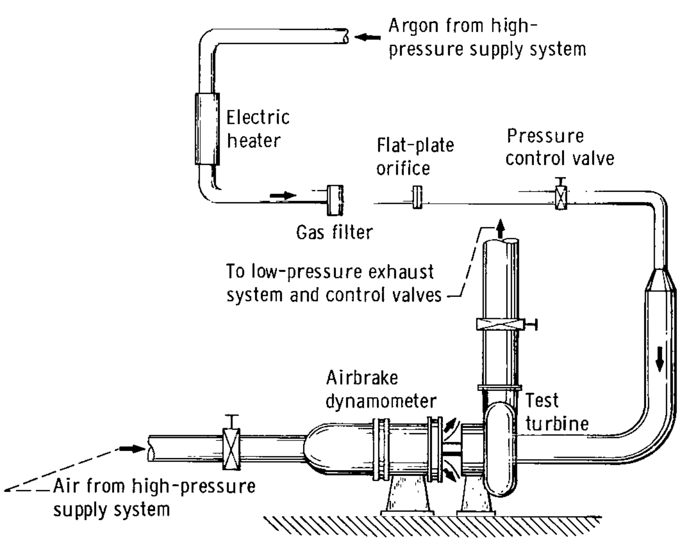


Figure 4. - Schematic of experimental equipment. CD-9274

turbine; these results were used as a check on turbine total efficiency as calculated from torque, mass flow, and speed measurements.

Overall performance was based on measurements taken at stations 1 and 6. The efficiency based on collector-exit conditions, station 6, is of importance for the space power system since the exhaust collector is part of the turbine package for the system. Comparison of turbine work and efficiency for the two tip clearances was based on measurements at stations 1 and 5 (turbine inlet and rotor exit). These comparisons were used to determine the degree of performance sensitivity to tip clearance.

Performance data were taken at an inlet total temperature of 610° R (339° K) and a total pressure of 2.50 psia (1.723 N/cm^2 abs). These values of temperature and pressure correspond to a Reynolds number of 47 400 at design equivalent speed and design pressure ratio. This value approximates the design value of 49 500. Reynolds number is defined herein as $Re = w/\mu r_m$. Data were taken over a range of equivalent inlet total- to exit-static-pressure ratios p_1'/p_5 from 1.08 to 1.55 and over a range of speeds from 0 to 120 percent of design equivalent speed. A rotor-exit radial survey of flow angle, total pressure, and total temperature was made at design equivalent speed and design pressure ratio.

Friction torque of the bearings and seals was obtained by measuring the amount of torque required to drive the shaft and rotor over the range of speeds covered in this investigation. In obtaining the friction torque, rotor windage losses were minimized by evacuating the air from the turbine to about 10 to 20 microns of mercury (0.00013 to 0.00026 N/cm^2) by sealing off the inlet and outlet pipes to the turbine. A friction torque value of about 1 inch-pound (0.113 N-m) was obtained at the design equivalent speed of 7200 rpm. This torque value corresponds to about 5.5 percent of the turbine torque obtained at design pressure ratio and design equivalent speed. The friction torque was added to shaft torque to determine the aerodynamic performance.

The turbine was rated on the basis of both total and static efficiencies from measurements taken at the turbine inlet (station 1) and the rotor exit (station 5), as well as at station 6 which is the exit of the collector. The total pressures were calculated from weight flow, static pressure, total temperature, and flow angle. In the calculation of total pressure at station 6, the flow was assumed to be normal to the plane defined by that station. The total temperature value used in the calculation of total pressure for stations 5 and 6 was determined from the measured inlet total temperature and the specific work as obtained from torque measurements.

The absolute values of the pressures at the various stations were measured by the use of manometer tubes that contained a fluid with a specific gravity of 1.04 and which were used as absolute manometers. The reference side of each manometer tube was evacuated to 10 microns of mercury (0.00013 N/cm^2). All other data were recorded by an automatic digital potentiometer, and all data were processed through an electronic digital computer. Equivalent pressure ratio was calculated from equivalent specific work and from the value of efficiency obtained from the tests in argon.

It is estimated that the probable error involved in obtaining the efficiency at design Reynolds number and at design equivalent speed and pressure ratio was less than 1 percent. This estimate is based on the discussion in reference 4 which concerns the accuracy of data taken with the same or similar instrumentation. This reference states that the probable error of a single observation at design Reynolds number was about 1 percent. The quoted values of efficiency for the subject turbine were based on faired curves and should accordingly have a probable error less than 1 percent.

RESULTS AND DISCUSSION

As described in the previous section, the subject turbine was investigated at nominal inlet conditions of 2.50 psia (1.723 N/cm^2) and 610° R (339° K). Data were taken over a range of speeds from 0 to 120 percent of design equivalent and at pressure ratios p_1'/p_5 from 1.08 to 1.55.

Performance results are presented for the turbine operating with a rotor tip clearance of 0.031 inch (0.079 cm), which is the recommended tip clearance value for preliminary hot operation. The performance values obtained at this clearance are compared with those obtained at the design 0.013-inch (0.033-cm) tip clearance, as reported in reference 1. As an aid in comparing performance for the two tip clearances, table I lists the performance results obtained at design equivalent speed and pressure ratio for both tip clearances, as well as the design-point values.

TABLE I. - PERFORMANCE VALUES

Performance parameters	Experimental		Design
	Design tip clearance	Increased tip clearance	
Equivalent mass flow, $\epsilon w\sqrt{\theta_{cr}}/\delta$ lb/sec (kg/sec)	1.477 (0.670)	1.495 (0.678)	1.537 (0.697)
Equivalent torque, $\tau\epsilon/\delta$, in.-lb (N-m)	104.44 (11.80)	100.19 (11.32)	108.12 (12.216)
Static efficiency, $\eta_{s,1-5}$	0.825	0.782	0.825
Total efficiency, $\eta_{t,1-5}$	0.845	0.800	0.850
Static efficiency, $\eta_{s,1-6}$	0.826	0.785	0.826
Total efficiency, $\eta_{t,1-6}$	0.835	0.792	0.843

Performance results are also presented on the basis of change in efficiency with rotor tip clearance expressed as a percentage of annular passage height. Included is a comparison with the results of a tip-clearance investigation of an axial-flow turbine of impulse design (ref. 2).

All data, with the exception of the radial surveys, are shown in terms of air equivalent values. Blade-jet speed ratio in every case is based on turbine-inlet-total and rotor-exit-static conditions.

Overall Performance With Increased Tip Clearance

The mass flow characteristics of the subject turbine operating with increased tip clearance are shown in figure 5. At design pressure ratio (0.811) and speed, the mass flow was 1.495 pounds per second (0.678 kg/sec). This mass flow is 1.2 percent higher than the 1.477 pounds per second (0.670 kg/sec) obtained with the turbine operating with the smaller tip-clearance value. This increase in mass flow is caused by increased flow in the enlarged rotor clearance space. The variation of mass flow with pressure ratio for lines of constant speed is typical of turbines of subsonic design. The curves

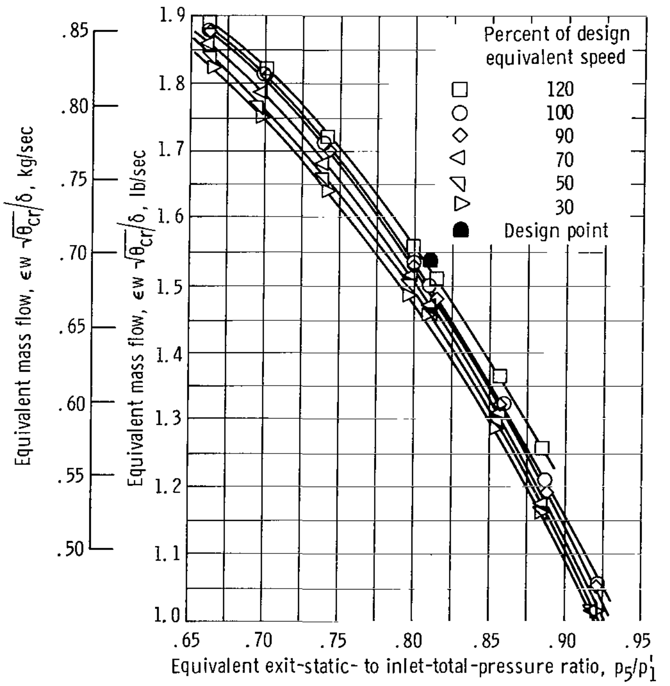


Figure 5. - Variation of mass flow with pressure ratio and speed for two-stage operation with 0.031-inch (0.079-cm) tip clearance.

shown in figure 5 are similar to the curves obtained with the smaller tip clearance (ref. 1).

The variation of mass flow with speed for a given pressure ratio (decreasing mass flow with decreasing speed) results primarily from the low axial velocity component through the turbine. This low axial velocity results in high incidence losses as the speed is varied from design.

The variation of equivalent torque $\tau\epsilon/\delta$ with speed and pressure ratio is shown in figure 6. A torque value of 100.19 inch-pounds (11.32 N-m) was obtained at design

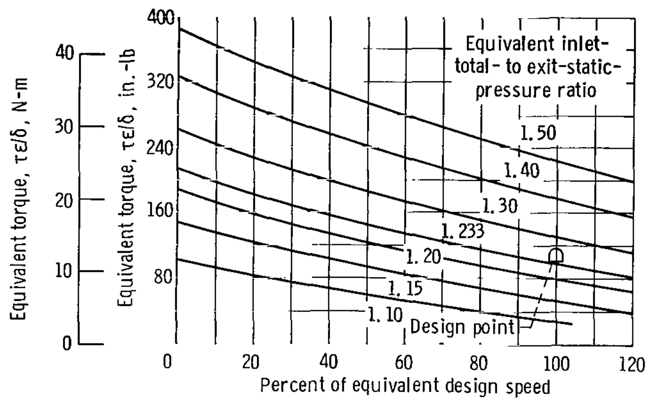
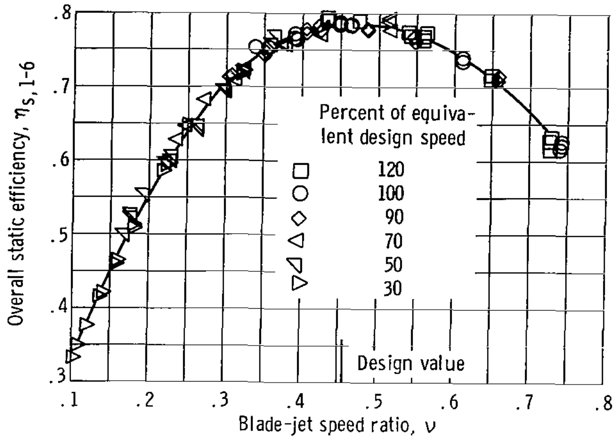


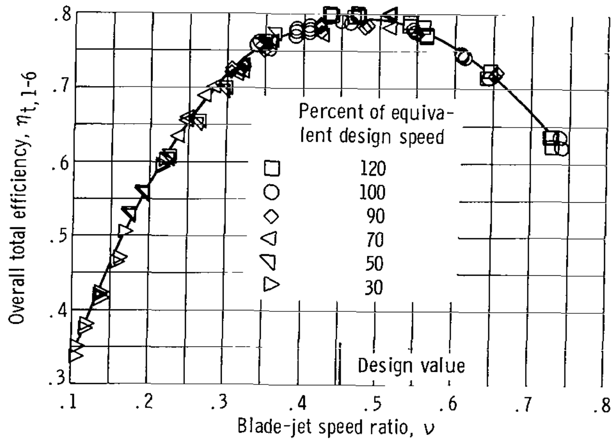
Figure 6. - Variation of torque with equivalent speed and pressure ratio for 0.031-inch (0.079-cm) tip clearance.

speed and a pressure ratio of 1.233. This torque value is 4 percent lower than the 104.44 inch-pounds (11.80 N-m) obtained with the smaller tip clearance. Zero-speed torque at design pressure ratio was 213.3 inch-pounds (24.10 N-m), which is 2.1 times the torque value obtained at design speed and pressure ratio. A 2.2 ratio was obtained for the smaller tip clearance.

The variation of static and total efficiency with blade-jet speed ratio is shown in figure 7. For these curves, both static and total efficiencies are based on conditions at the turbine inlet and the collector exit. As shown in figure 7(a), the static efficiency was 0.785 at design speed and a design blade-jet speed ratio of 0.465. Therefore, increasing the tip clearance resulted in approximately a 4-percentage-point drop in efficiency, since a value of 0.826 was obtained for the smaller tip clearance. The figure also shows that peak efficiency was obtained at design blade-jet speed ratio. Figure 7(b) shows that a total efficiency value of 0.792 was obtained at design speed and design blade-jet speed ratio. This efficiency value is also approximately 4 percentage points lower than the total efficiency value of 0.835 obtained with the smaller tip clearance.



(a) Static efficiency.

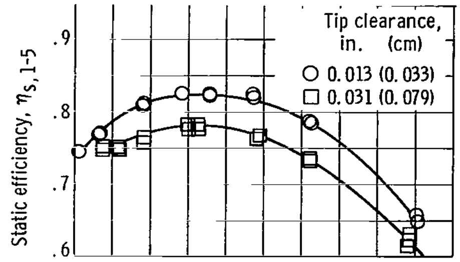


(b) Total efficiency.

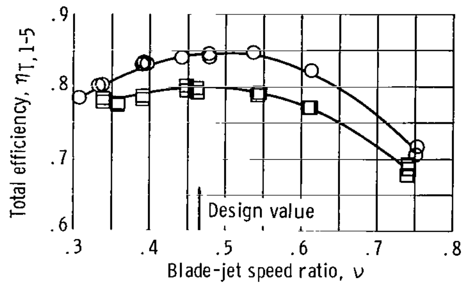
Figure 7. - Variation of efficiency with blade-jet speed ratio for 0.031-inch (0.079-cm) tip clearance.

Effect of Tip Clearance on Blading Performance

Efficiency characteristics of the turbine were also obtained for conditions based on the turbine inlet and the rotor exit (stations 1 and 5) in order to determine the effects of tip clearance on the turbine blading performance. Figure 8 presents the variation of efficiency at design equivalent speed with blade-jet speed ratio for the two values of tip clearance investigated. Figure 8(a) shows that at design blade-jet speed ratio there again was a 4-percentage-point decrease in static efficiency when the tip clearance was increased from 0.013 to 0.031 inch (0.033 to 0.079 cm). Similarly, figure 8(b) shows that, for design blade-jet speed ratio, there was also a corresponding 4-percentage-point decrease in total efficiency when the tip clearance was increased.



(a) Static efficiency.



(b) Total efficiency.

Figure 8. - Variation of efficiency with blade-jet speed ratio at design equivalent speed and for two values of rotor blade tip clearance.

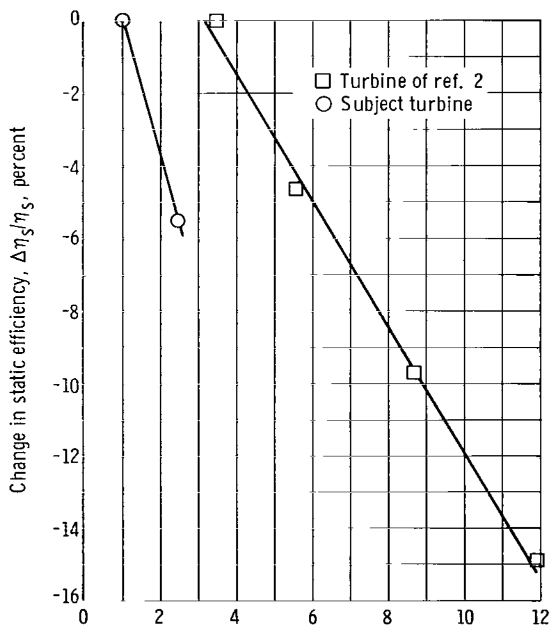


Figure 9. - Effect of rotor tip clearance on performance at design equivalent speed and design pressure ratio.

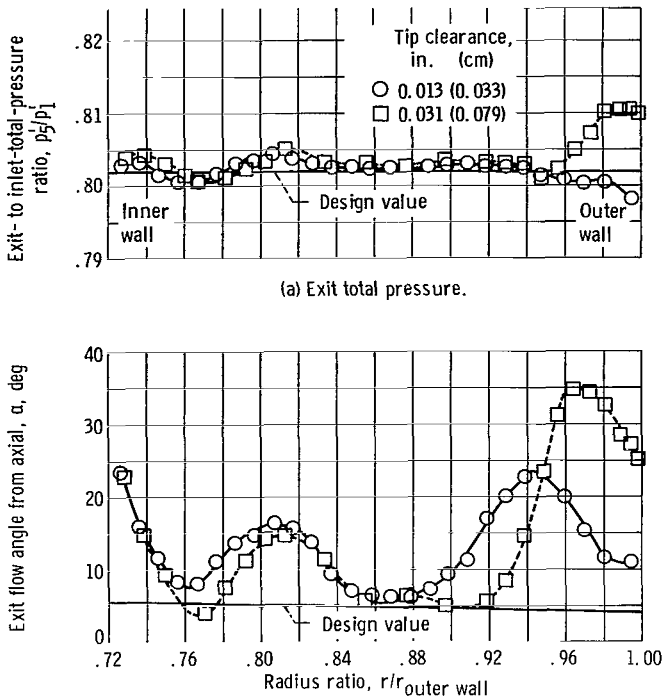
This decrease in efficiency may be attributed to such factors as rotor blade tip unloading, increased throughflow over the blade tips in the clearance space, and the reduction of working blade area. As was stated in the section Turbine Assembly, tip clearance was increased by grinding material from the rotor blade tips.

In figure 9, the results of the change in tip clearance in terms of change in efficiency (expressed as a percentage of the efficiency obtained at the lowest tip-clearance value) are compared with tip clearance as a percentage of the annular passage height. Included in the figure are the results obtained from the turbine of reference 2. However, the reference turbine was designed for impulse conditions, that is, for approximately constant relative velocity through the rotor while the subject turbine had a comparably high amount of reaction across the rotors. Figure 9 indicates that there was considerably more effect of tip-clearance change on efficiency for the subject turbine. The slopes of the curves are -1.75 for the reference turbine and -3.7 for the subject turbine. Therefore, for the subject turbine, specific work decreased at the rate of 3.7 percent for an increase in tip clearance of 1.0 percent of passage height. This decrease is substantially greater than the 1.75-percent decrease for the reference turbine.

Comparison of Rotor-Exit Conditions

Figure 10 shows the results of a radial survey taken at design equivalent speed and design pressure ratio for both tip clearances investigated. Figure 10(a) indicates that there was no significant difference in exit total pressure for both clearances until the region of radius ratio from 0.95 to 1.00 (or outer wall) was reached. In this region, there is an appreciably greater total pressure for the larger tip-clearance case. Therefore, the velocity was higher in this region for the larger tip clearance since the static pressure was essentially the same for both tip clearances. The higher velocity is to be expected since the mass flow increased in this region for the larger tip clearance. Figure 10(b), similarly, shows that the rotor-exit flow angle is about the same for both clearances from the inner wall to a radius ratio of about 0.90. The increased tip clearance caused considerably more underturning, as denoted by positive angles, in the region near the outer wall. This larger angle was probably caused by the decreased blade length and the increased flow in the clearance space.

Figure 11 shows the variation of total temperature drop against radius ratio for design equivalent speed and design pressure ratio. This figure indicates that local turbine work was reduced over a considerable portion of the blade (radius ratios of 0.80 to 0.97) when the tip clearance was increased. In terms of local total efficiency, however, the greatest difference of about 5 points in efficiency occurs in the radius ratio range of 0.94 to 0.98. Again, the largest changes would be expected at the outer wall since this



(a) Exit total pressure.

(b) Exit flow angle.

Figure 10. - Effect of increased tip clearance on exit total pressure and flow angle at design equivalent speed and design pressure ratio.

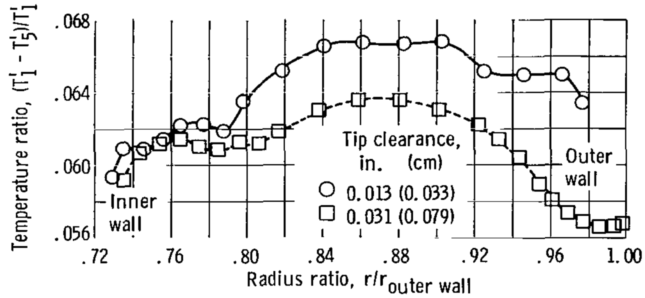


Figure 11. - Effect of increased tip clearance on turbine-exit total temperature at design equivalent speed and design pressure ratio.

is the region where rotor blade material was removed for increased tip clearance and there was greater flow over the blade tips in the larger clearance space.

SUMMARY OF RESULTS

The performance of a two-stage turbine was investigated at two values of tip clearance. Performance characteristics were first described for the turbine operating with the rotor tip clearance of 0.031 inch (0.079 cm) which was recommended for preliminary hot operation. These results were then compared with those obtained in a reference investigation in which the design tip clearance of 0.013 inch (0.033 cm) was used. The results of this investigation are summarized as follows:

1. Static and total efficiencies (based on turbine-inlet and collector-exit conditions) were 0.785 and 0.792, respectively, for operation at design equivalent speed and pressure ratio with a tip clearance of 0.033 inch (0.079 cm). These values represent a 4-percentage-point decrease in efficiency when compared with the results obtained when the turbine was investigated at the design tip clearance of 0.013 inch (0.033 cm).

This decrease in efficiency may be attributed to such factors as rotor blade tip unloading, increased throughflow over the blade tips in the clearance space, and the reduction of working blade area.

2. At design equivalent speed and pressure ratio, the equivalent mass flow was 1.495 pounds per second (0.0678 kg/sec) which represents a 1.2-percent increase in mass flow when the tip clearance was increased. The corresponding equivalent torque was 100.19 inch-pounds (11.32 N-m), and this value represents a 4-percent decrease in torque with the larger tip clearance.

3. The results based on the two tip clearances indicate that the subject turbine, which was designed with rotor reaction, was very sensitive to changes in tip clearance. Turbine specific work decreased at the rate of 3.7 percent for an increase in tip clearance of 1.0 percent of passage height. This decrease is substantially greater than the 1.75 percent decrease in turbine work obtained for a reference turbine of impulse design.

Lewis Research Center,
National Aeronautics and Space Administration,
Cleveland, Ohio, October 13, 1967,
120-27-03-13-22.

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