NASA Technical Paper 1032

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COLD-AIR PERFORMANCE OF A 12.766-CENTIMETER-TIP-DIAMETER AXIAL-FLOW COOLED TURBINE

III - EFFECT OF ROTOR TIP CLEARANCE ON OVERALL PERFORMANCE OF A SOLID BLADE CONFIGURATION

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SUMMARY

An experimental investigation was made to investigate the effect of varying the rotor tip clearance of a 12.766-centimeter-tip-diameter, single-stage, axial-flow turbine.

Two tip clearance configurations, one with a recess in the casing and the other with a reduced rotor blade height, were investigated at design equivalent speed over a range of tip clearance from about 2.0 to 5.0 percent of the stator blade height.

The optimum configuration with a recess in the casing was the one where the rotor tip diameter was equal to the stator tip diameter (zero blade extension). For this configuration there was an approximate 1.5 percent decrease in total efficiency for an increase in tip clearance of 1 percent of stator blade height.

For the reduced blade height configurations there was an approximate 2.0 percent decrease in total efficiency for an increase in tip clearance of 1 percent of stator blade height.

INTRODUCTION

Advanced small turboshaft engines in the 1.00- to 4.50-kilogram-per-second, 250- to 1100-kilowatt class are being designed to operate at cycle pressure ratios of 10 to 1 or higher, with turbine inlet temperatures as high as 1550 K. The high compressor pressure ratio, together with the small mass flow, results in a turbine design with a small annulus area, and, therefore, a small blade height. A high turbine inlet temperature requires the use of blade cooling air, and therefore, the stator and rotor blade profiles must be longer and thicker than desired from an aerodynamic standpoint to provide adequate space for cooling passages. Long chord lengths and small blade heights result in a low aspect ratio design.

With these small blade height, low aspect ratio turbines, geometric similarity with larger turbines becomes difficult to maintain. For example, rotor tip clearances of about 1 to 1.5 percent of the stator blade height as commonly used in larger turbines would require tip clearances of about 0.013 centimeter. These small clearances are generally not practical in these small turbines because of manufacturing and engine buildup tolerance limits, as well as thermal growth and rotor dynamics considerations, particularly during engine startup and shutdown. As a result, rotor tip clearances of about 2.5 percent of the stator blade height are required. Because it is necessary to operate with larger tip clearance ratios, the tip clearance losses are more severe. To help understand these loss effects, a determination of the penalty associated with varying the radial clearance is required.

Over the past 15 years, several investigations have been conducted at the NASA Lewis Research Center to determine the rotor tip clearance penalty for various turbine designs. The results from three of these investigations are discussed in references 1 to 3.

The turbine used in reference 1 was of near impulse rotor design. In this investigation three rotor tip clearance configurations were tested. First, a shrouded rotor blade tip with a labyrinth seal in the turbine casing recess to reduce the through-flow leakage was tested. For this configuration there was a 0.31 percent decrease in static efficiency for a 1 percent increase in rotor tip clearance. Second, a recessed-casing configuration was tested, in which the rotor blade height was greater than the stator blade height by about 2.0 percent and the rotor tip clearance was obtained by a recess in the turbine casing above the rotor. For this configuration there was a 0.91 percent decrease in static efficiency for a 1 percent increase in rotor tip clearance. Finally, a reduced blade-height configuration was investigated. For this type of configuration the turbine casing wall was maintained at a diameter equal to the stator tip diameter, and the rotor tip clearance was obtained by reducing the rotor tip diameter. For this configuration there was a 1.75 percent decrease in static efficiency for a 1 percent increase in rotor tip clearance.

The turbines used in the investigations of references 2 and 3 were designed with levels of tip reaction R_x of 0.805 (ref. 2) and 0.834 (ref. 3). For these investigations, only reduced blade height configurations were tested. For the reference 2 turbine, the total efficiency decreased by 1.7 percent for a 1 percent increase in rotor tip clearance, whereas, for the reference 3 turbine, the total efficiency decreased by 3.2 percent for a 1 percent increase in rotor tip clearance.

Since the previous tip clearance investigations with turbines having high levels of tip reaction (refs. 2 and 3) were conducted only with reduced blade height configurations, a decision was made to conduct an experimental investigation to determine the tip clearance losses associated with both reduced blade height and recessed casing configurations for a turbine having a level of tip reaction of about 0.9.

This report describes the results of an experimental investigation to evaluate the effect of varying the rotor tip clearance of a 12.766-centimeter-tip-diameter, single-stage, axial-flow turbine. This turbine was the configuration used in reference 4 and had a value of tip reaction of 0.890. In this investigation, the rotor tip clearance was obtained by use of a recess in the casing over the rotor blade and also by use of a reduced blade height. The recessed casing configuration was tested with the rotor blade extended into the recess by about 0.035 centimeter or 3.28 percent of the stator blade height. The rotor blade tip diameter was then reduced in four increments of about 0.015 centimeters each until the rotor tip diameter was equal to the stator tip diameter. For each rotor diameter data were taken at four values of rotor tip clearance between 2.0 and 5.0 percent of the stator blade height. Next, the reduced blade height configuration was tested. For these tests, the casing diameter was set equal to the stator tip diameter and the rotor tip clearance was changed by reducing the rotor blade tip diameter. In this part of the investigation, a total of three rotor tip clearances were tested over a range from about 2.0 to 5.0 percent of the stator blade height.

All tests were conducted at design equivalent speed with nominal turbine inlet conditions of 8.27 newtons per square centimeter and 320 K. For each rotor tip clearance tested, data were obtained over a range of total to static pressure ratio from 1.6 to 4.0. A rotor exit radial survey of absolute flow angle was also made for each configuration near the design total pressure ratio of 2.77. The effect of tip clearance on performance is presented in terms of efficiency, rotor reaction, and rotor exit absolute flow angle. In addition, the effect of clearance on efficiency for the subject turbine is compared to the experimental results from the turbines of references 1 to 3.

SYMBOLS

A	area, cm^2
Δh	specific work, J/g
N	rotative speed, rpm
p	absolute pressure, N/cm^2
R	gas constant, J/(kg)(K)
RBH	reduced blade height
RC	recessed casing

R_x tip reaction, $\left(W_{3t}^2 - W_{2t}^2\right)/W_{3t}^2$

r radius, cm

T absolute temperature, K

U blade velocity, m/sec

V absolute gas velocity, m/sec

ΔV_u change in absolute tangential velocity, m/sec

W relative gas velocity, m/sec

w mass flow, kg/sec

α absolute gas flow angle measured from axial direction, deg

β relative 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, p_1'/p^*

ϵ function of γ used in relating parameters to those using air inlet conditions, $\left(\frac{0.740}{\gamma}\right)\left(\frac{\gamma + 1}{2}\right)^{\gamma/(\gamma-1)}$

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 temperature to critical velocity at U. S. standard sea-level temperature, $(V_{cr}/V_{cr}^*)^2$

τ torque, N-m

Subscripts:

cr condition corresponding to Mach number of unity

eq equivalent

t rotor tip

0 zero rotor tip clearance

1 station at turbine inlet (fig. 5)

2 station at stator exit (fig. 5)

3 station at rotor exit (fig. 5)

4 station at turbine exit (fig. 5)

Superscripts:

' absolute total state

* U.S. standard sea-level conditions (temperature, 288.15 K; pressure, 10.13 N/cm^2

TURBINE DESCRIPTION

The turbine used in this experimental investigation was the solid blade version of a single-stage, axial-flow cooled turbine designed to drive a two-stage, 10-to-1 pressure ratio compressor with a mass flow of 0.907 kilogram per second, a rotative speed of 70 000 rpm, and a turbine inlet temperature of 1478 K. A list of both the engine design conditions and the equivalent design conditions for this turbine are presented in table I. A more detailed description of the turbine design is presented in reference 4. The turbine was designed with both the stator and rotor blading being untwisted and untapered. Table II lists some of the physical parameters for this turbine. An aspect ratio of 1.00 was selected. The solidities were 1.61 and 1.70 for the stator and rotor, respectively, at the mean section. There were 56 stator blades and 59 rotor blades.

Figure 1 shows the velocity diagrams as calculated at the hub, mean, and tip diameters. It can be seen that the stator discharge angle was a constant 74.2° and the rotor discharge angle was a constant 61.9° from hub to tip.

The blade surface velocities at the hub, mean, and tip diameters are shown in figure 2 for the stator and rotor. The figure shows that there was no large diffusion predicted for any of the three blade sections.

Figure 3(a) is a photograph of the stator assembly, and figure 3(b) is a photograph of the rotor and shaft assembly. These photographs show some of the design features of this turbine. The stator assembly shown in figure 3(a) was the same one used in the investigation of reference 4.

APPARATUS, INSTRUMENTATION, AND TEST PROCEDURE

The apparatus used in this investigation consisted of the subject turbine, an airbrake dynamometer used to absorb and measure the power output of the turbine, an inlet and exhaust piping system including flow controls, and appropriate instrumentation. A schematic of the experimental equipment and instrument measuring stations is shown in figure 4. A cross-sectional view of the turbine is shown in figure 5.

Instrumentation at the turbine inlet (station 1) measured static pressure and total temperature. Static pressures were obtained from eight taps with four on the inner wall and four on the outer wall. The inner and outer taps were located opposite each other at 90° intervals around the circumference at a distance approximately two axial chord lengths upstream of the stator. The temperature was measured with three thermocouple rakes, each containing three thermocouples at the area center radii of three equal annular areas.

At station 2, two static pressure taps were located 180° apart on the outer wall.

At station 3, approximately three axial chord lengths downstream of the rotor, the static pressure, total pressure, total temperature, and flow angle were measured. The static pressure was measured with eight taps with four each on the inner and outer walls. These inner and outer wall taps were located opposite each other at 90° intervals around the circumference. A self-alining probe was used for measurement of total pressure, total temperature, and flow angle.

There were four total temperature rakes, each containing three thermocouples, at station 4 located about 16 axial chord lengths downstream from the rotor exit. Temperatures from these rakes were used to calculate a turbine temperature efficiency. This efficiency was used to check the turbine torque efficiency as calculated from torque, speed, and mass flow measurements. The difference in the temperature and torque efficiencies was within 1 percentage point. Torque efficiency is presented in this report.

The rotational speed of the turbine was measured with an electronic counter in conjunction with a magnetic pickup and a shaft-mounted gear. Mass flow was measured with a calibrated critical flow nozzle. An airbrake dynamometer absorbed the power output of the turbine. Torque was measured by the airbrake, which was mounted on air trunion bearings. The torque load was measured with a commercial strain-gage load cell.

In order to obtain aerodynamic performance, friction torque was added to dynamometer torque. The friction torque from the bearings, seal, and coupling windage was obtained by driving the rotor and shaft over the range of speeds covered in this investigation. In order to eliminate disk windage and blade pumping and churning losses from the friction torque, the turbine cavity was evacuated to a pressure to approximately 0.013 newtons per square centimeter. A friction torque value of 0.140 newton-meter was obtained at equivalent design rotative speed. This value of friction torque corresponded to about 3.0 percent of the work obtained at equivalent design rotative speed and pressure ratio.

In this investigation, the rotor tip clearance was obtained by use of a recess in the casing over the rotor blades and also by use of a reduced blade height. Figure 6 shows a schematic of these two configurations. The recessed casing configuration was tested with the rotor blade extended into the recess by 0.035 centimeter, or 3.28 percent of the stator blade height. The rotor blade tip diameter was then reduced in four increments of about 0.015 centimeter each until the rotor tip diameter was equal to the stator tip diameter. For each rotor diameter, data were taken at four values of rotor tip clearance between 2.0 and 5.0 percent of the stator blade height. The tip clearance was varied by varying the casing recess depth. In these tests, the recess extended the entire width of the rotor hub. In addition, when the rotor blade was extended into the recess by 1.70 percent of the stator blade height, tests were made to determine the effect on tip clearance loss of shortening the axial length of the recess.

Finally, the reduced blade height configuration was tested. For these tests, the casing diameter was set equal to the stator tip diameter, and the rotor tip diameter was machined to attain the desired tip clearance. In this part of the investigation, a total of three rotor tip clearances were tested over a range of about 2.0 to 5.0 percent of the stator blade height.

Table III presents a list of the 27 total tip clearance configurations tested in this program. In this report, the term ''rotor blade extension'' will refer to the percent, relative to the stator blade height, that the rotor blade tip radius is extended beyond the stator blade tip radius.

All tests were conducted at design equivalent speed with nominal turbine inlet conditions of 8.27 newtons per square centimeter and 320 K. In this report, the turbine was rated on the basis of both total and static efficiency. The total pressures used in determining these efficiencies were calculated from mass flow, static pressure, total temperature, and flow angle from the following equation:

$$p' = p\left\{\frac{1}{2} + \frac{1}{2}\left[1 + \frac{2R(\gamma - 1)}{\gamma}\left(\frac{w\sqrt{T'}}{pA \cos \alpha}\right)^2\right]^{1/2}\right\}^{\gamma/(\gamma-1)}$$

In the calculation of turbine inlet total pressure, the flow angle was assumed to be zero.

RESULTS AND DISCUSSION

Performance results from an experimental investigation to evaluate the effect of varying the rotor tip clearance of a 12.766-centimeter-tip-diameter, single-stage, axial-flow turbine are presented. The rotor tip clearance was obtained by use of a recess in the casing above the rotor blades and also by use of a reduced blade height. Performance tests were made with air as the working fluid with nominal turbine inlet conditions of 8.27 newtons per square centimeter and 320 K. For each tip clearance tested, data were obtained at design equivalent speed over a range of total to static pressure ratio from 1.6 to 4.0. In addition, a rotor exit radial survey was made for each configuration near the design total pressure ratio. The effect of tip clearance on performance is presented in terms of efficiency, rotor reaction, and rotor exit absolute flow angle. In addition, the effect of clearance on efficiency for the subject turbine is compared to the experimental results from three reference turbines.

MASS FLOW

Figure 7 shows the variation of equivalent mass flow with total to static pressure ratio for both the recessed casing and reduced blade height configurations. The figure indicates that the stator was choked over most of the pressure ratio range at a mass flow rate of 0.231 kilogram per second. This was the same value of mass flow obtained in reference 4 and was about 6.1 percent smaller than design due to an undersized stator throat area. As expected, both configurations showed identical mass flow characteristics over a range of tip clearance.

TURBINE EFFICIENCY

Recessed Casing Configuration

Figure 8 shows the change in total efficiency with rotor blade extension for lines of constant rotor tip clearance. This figure was obtained from a crossplot of efficiency and tip clearance for lines of constant rotor blade extension. The changes in total efficiency shown in this figure are referenced to an efficiency obtained by extrapolating the zero blade extension data to zero rotor tip clearance. The dotted lines shown for zero and 1-percent rotor tip clearance were extrapolated from the measured data.

Figure 8 indicates a trend of decreasing efficiency with increasing rotor tip clearance for a constant rotor blade extension. The efficiency also decreases with increasing rotor blade extension for a constant tip clearance. In addition, the slope of the lines of constant rotor tip clearance increase slightly with increasing rotor tip clearance. The optimum recessed casing configuration from the standpoint of having the highest level of efficiency was the one where the rotor blade tip diameter was equal to the stator tip diameter (zero blade extension). For this optimum configuration there was an approximate 1.5 percent decrease in total efficiency for an increase in tip clearance of 1 percent of stator blade height.

For a blade extension of 3.5 percent there was an approximate 2.0 percent decrease in total efficiency for an increase in tip clearance of 1 percent of stator blade height. This slight increase in loss with an increase in blade extension was attributed to pumping losses that occurred due to the extended portion of the rotor blade rotating in a region of relatively low momentum fluid.

At zero rotor tip clearance there was an approximate 0.3 percent decrease in total efficiency for each percent increase in rotor blade extension. At a rotor tip clearance of 5 percent, this efficiency loss increased to about 1.0 percent for each percent increase in rotor blade extension.

Reduced Blade Height Configuration

Figure 9 shows the change in total efficiency with rotor tip clearance for the reduced blade height configurations. This data is also at the design total pressure ratio. The dotted line in this figure indicates that the data was extrapolated to zero rotor tip clearance. There was an approximate 2.0-percent decrease in total efficiency with an increase in tip clearance of 1 percent of stator blade height.

This decrease in total efficiency with the reduced blade height configurations was slightly larger than that for the optimum recessed casing configuration. As indicated in reference 1 the factors affecting turbine work for the reduced-blade-height configuration consist of reduced blade area for doing turbine work, blade tip unloading (flow over the rotor blade tip from pressure to suction surface), and through flow over the blade tip in the clearance space. The recessed-casing configuration would be expected to have higher efficiencies (over the range of tip clearance investigated) than the reduced-blade-height configuration, since the rotor blade height remained constant and only the factors of blade tip unloading and through flow over the blade tip affected turbine work.

It should be noted that the tip clearance losses mentioned previously in the discussion of figures 8 and 9 were obtained by linearly extrapolating the data from approximately 2.0 percent rotor tip clearance to zero rotor tip clearance. Closer examination of the data indicated that the trend in efficiency with tip clearance was actually slightly parabolic in nature with the efficiency increase becoming greater as zero tip clearance was approached. This trend was also noted in the data of references 1 and 2. However, since no data was obtained below about 2.0 percent tip clearance, the actual trend in efficiency could not be accurately determined. Therefore, a linear curve fit of the data was made. These resulting tip clearance losses for the recessed casing and reduced blade height configurations would approximately represent average tip clearance losses for the data over the range of tip clearance investigated.

Rotor Exit Flow Angle

Figure 10 shows the variation in rotor exit absolute flow angle for the optimum recessed casing and the reduced blade height configurations. This data was obtained from rotor exit radial surveys conducted at design total pressure ratio. Both figures 10(a) and 10(b) show a trend of reduced flow turning as the radial clearance increased. The largest changes occurred in the blade portion between the midspan and the tip, indicating that a change in tip clearance affects the flow conditions over a large portion of the blade. As the tip clearance is increased, there is greater unguided throughflow over the rotor tip and greater tip leakage flow over the blade tip from the suction to the pressure surface.

Static Pressure Comparison

The tip static pressure variation through the turbine at design total pressure ratio for various clearances for both the optimum recessed casing and reduced blade height configurations is shown in figure 11. For each configuration the static pressure at the turbine inlet and turbine exit was set at constant values for all clearances. Increasing the tip clearance resulted in a drop in the static pressure measured at the stator exit outer wall taps, and thus, the tip reaction decreased. This trend was expected since the amount of unguided flow area over the blade tip increased as the tip clearance increased.

Effect of Changing the Axial Length of the Recessed Casing

Figures 12 and 13 show the results of additional tests that were conducted to investigate the effect of changing the axial length of the recessed casing. These tests were conducted at a blade recess equal to 1.7 percent of stator blade height. Figure 12 shows a schematic of the two different axial recessed clearance configurations tested. As was mentioned previously, for all of the recessed casing configurations, the recess extended the entire width of the rotor hub (fig. 12(a)). However, for these additional tests, the axial length of the recess was reduced to be just beyond the rotor blade trailing edge (fig. 12(b)). This was done to determine if a shorter casing recess would result in a reduced tip clearance loss.

Figure 13 shows the results from this investigation. These data were obtained at design total pressure ratio. This figure indicates that there was essentially no difference in efficiency between the two configurations over the range of rotor tip clearance investigated. Thus, in this investigation the length of the recessed casing did not affect the level of efficiency.

COMPARISON OF RESULTS WITH OTHER TIP CLEARANCE INVESTIGATIONS

A comparison of the efficiency loss with radial clearance for various tip clearance investigations is shown in figure 14. The comparisons were made on the basis of static efficiency since this was the only efficiency available for reference 1. The efficiencies are expressed as a fraction of the efficiency with zero tip clearance. The legend indicates the type of turbine configuration (impulse or reaction), the degree of tip reaction, the type of tip clearance configuration (recessed casing or reduced blade height), and the efficiency decrease for each percent increase in tip clearance.

Included on this curve are the recessed casing and reduced blade height configurations from the single-stage impulse turbine of reference 1, the reduced blade height configurations from the single-stage reaction turbines of references 2 and 3, and the reduced blade height and optimum recessed casing configurations from the subject turbine.

Two trends are noticeable from this figure. First, the tip clearance loss associated with the recessed casing configuration was less than that associated with the reduced blade height configuration for both the impulse turbine (ref. 1) and the subject turbine. Secondly, except for the reference 3 turbine, the tip clearance loss increased with an increase in the tip reaction.

The reason for the much higher tip clearance loss for the reference 3 turbine was attributed to the mass flow characteristics of the turbine. As was mentioned previously, the subject turbine had a choked stator and the mass flow remained constant as the tip clearance was changed. Similarly, the reference 2 turbine experienced only a 0.22 percent increase in mass flow for every 1 percent increase in radial clearance. However, the reference 3 turbine experienced a 0.88 percent increase in mass flow for every 1 percent increase in radial clearance. Thus, for this turbine the mass flow through the stator increased significantly as the radial clearance increased and this, in turn, resulted in a change in the velocity diagrams. Therefore, part of the higher tip clearance loss for the reference 3 turbine could be due to detrimental reaction and rotor incidence effects. If the actual tip clearance loss were to agree with the loss for the subject and reference 2 turbines, these incidence and reaction effects would have had to cause an approximate 1 percent loss in efficiency for every 1 percent increase in tip clearance.

SUMMARY OF RESULTS

An experimental investigation was made to determine the effect of varying the rotor tip clearance of a 12.766-centimeter-tip-diameter, single-stage, axial-flow reaction turbine. In this investigation, the rotor tip clearance was obtained by use of a recess in the casing above the rotor blades and also by use of a reduced blade height. The recessed casing configuration was tested with the rotor blade extended into the recess by 0.035 centimeter, or 3.28 percent of the stator blade height. The rotor blade tip diameter was then reduced in four increments of about 0.015 centimeter each until the rotor tip diameter was equal to the stator tip diameter. For each rotor diameter, data were taken at four values of rotor tip clearance between 2.0 and 5.0 percent of the stator blade height.

For the tests with the reduced blade height configuration the casing tip diameter was set equal to the stator tip diameter. In this part of the investigation, a total of three tip clearances were tested over a range from about 2.0 to 5.0 percent of the stator blade height. The results of this investigation may be summarized as follows:

1. The optimum recessed casing configuration from the standpoint of having the highest level of efficiency was the one where the rotor tip diameter was equal to the stator tip diameter (zero blade extension). For this configuration there was an approximate

1.5 percent decrease in total efficiency for an increase in tip clearance of 1 percent of stator blade height.

2. For the reduced blade height configuration there was an approximate 2.0 percent decrease in total efficiency for an increase in tip clearance of 1 percent of stator blade height.

3. The mass flow was choked at a value of 0.231 kilogram per second over most of the turbine pressure ratio range investigated. This same level of mass flow was obtained for both configurations over a range of tip clearance.

4. A comparison of the tip clearance loss for the subject turbine with three reference turbines indicated that, in general, the tip clearance loss increased with increasing tip reaction.

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TABLE I. - TURBINE DESIGN CONDITIONS

Parameter	Engine	Equivalent
Turbine inlet temperature, T_1', K	1478	288.2
Turbine inlet pressure, p_1', N/cm^2	91.2	10.1
Mass flow rate, w, kg/sec	0.907	0.246
Rotative speed, N, rpm	70 000	31 460
Specific work, Δh, J/g	307.3	62.1
Torque, τ, N-m	38.02	4.64
Power, kW	292.6	15.3
Total to total pressure ratio, p_1'/p_3'	2.57	2.77
Total to static pressure ratio, p_1'/p_3	2.92	3.16
Total efficiency, η'	0.85	0.85
Work factor, $\Delta V_u/U$	1.67	1.67

TABLE II. - TURBINE DESIGN

PHYSICAL PARAMETERS

Parameter	Stator	Rotor
Actual chord, cm	1.052	1.052
Axial chord, cm	.721	.968
Leading edge radius, cm	.051	.028
Trailing edge radius, cm	.010	.013
Radius, cm		
Hub	5.331	5.331
Mean	5.857	5.857
Tip	6.383	6.383
Blade height, cm	1.052	1.052
Solidity	1.61	1.70
Aspect ratio	1.00	1.00
Number of blades	56	59
Radius ratio	.835	.835
Blade pitch, cm	.653	.619

TABLE III. - TIP CLEARANCE CONFIGURATIONS TESTED

Configuration	Rotor tip diameter, cm	Rotor tip clearance, cm	Rotor tip clearance, percent of stator blade height	Rotor blade extension, percent of stator blade height
Recessed casing				
1	12.794	0.022	2.07	3.28
2	↓	.029	2.80	↓
3		.039	3.77	
4		.059	5.67	
5	12.774	.018	1.70	2.31
6	↓	.028	2.68	↓
7		.038	3.65	
8		.048	4.62	
[a]9	12.761	.024	2.31	1.70
[a]10	↓	.033	3.17	↓
[a]11		.042	4.02	
[a]12		.051	4.87	
13	12.761	.024	2.31	1.70
14	↓	.033	3.17	↓
15		.042	4.02	
16		.051	4.87	
17	12.743	.020	1.95	.85
18	↓	.030	2.92	↓
19		.041	3.89	
20		.051	4.87	
21	12.725	.020	1.95	0
22	↓	.030	2.92	↓
23		.041	3.89	
24		.051	4.87	
Reduced blade height				
25	12.685	0.019	1.83	---
26	12.649	.037	3.53	---
27	12.616	.053	5.12	---

[a]Tests made with short axial recessed clearance (fig. 12).

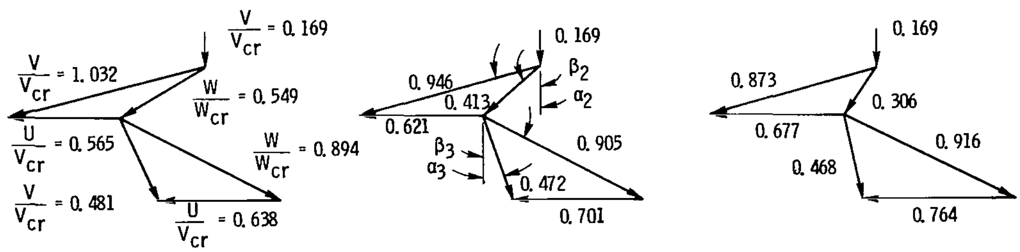
	Hub	Mean	Tip

Station

1, Stator entrance

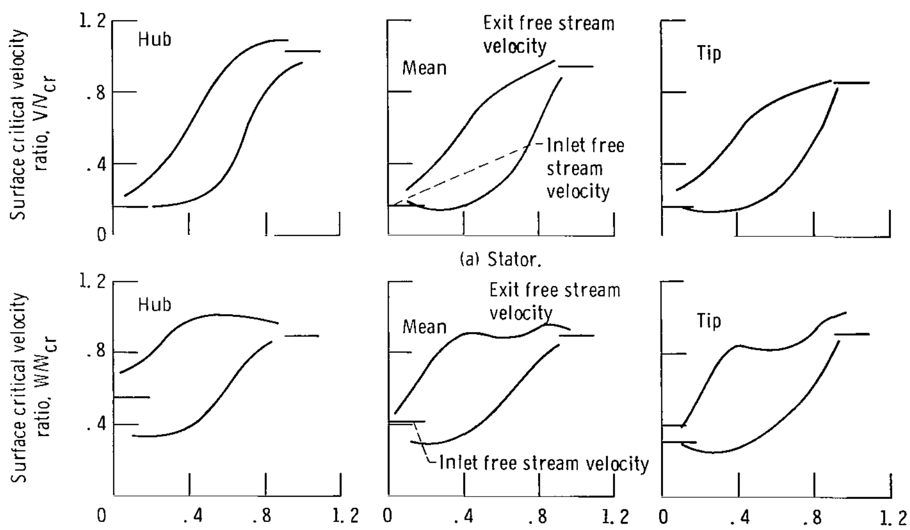
2, Stator exit

3, Rotor exit



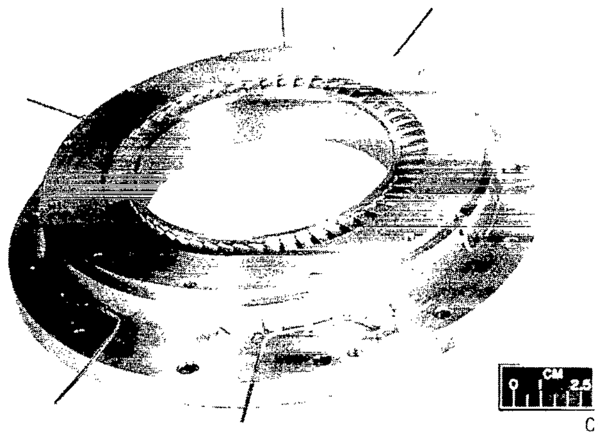
	Hub	Mean	Tip
Radius (cm) at -			
Station 1	5.33	5.86	6.38
Station 2	5.33	5.86	6.38
Station 3	5.33	5.86	6.38
α_2, deg	74.2	74.2	74.2
β_2, deg	56.7	48.2	34.4
α_3, deg	-23.3	-17.5	-11.6
β_3, deg	-61.9	-61.9	-61.9

Figure 1. - Design velocity diagrams.



(b) Rotor.

Figure 2. - Design blade surface velocity distributions at hub, mean, and tip.



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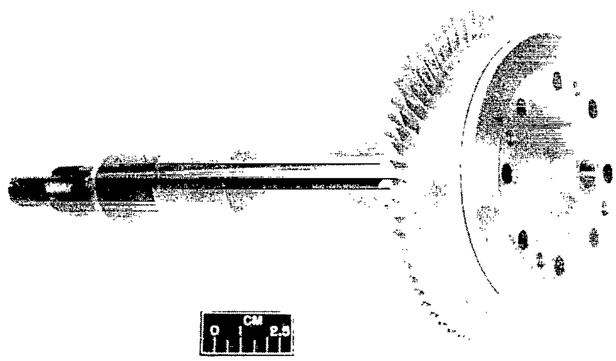
(a) Stator assembly.



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(b) Rotor and shaft assembly.

Figure 3. - Turbine test hardware.

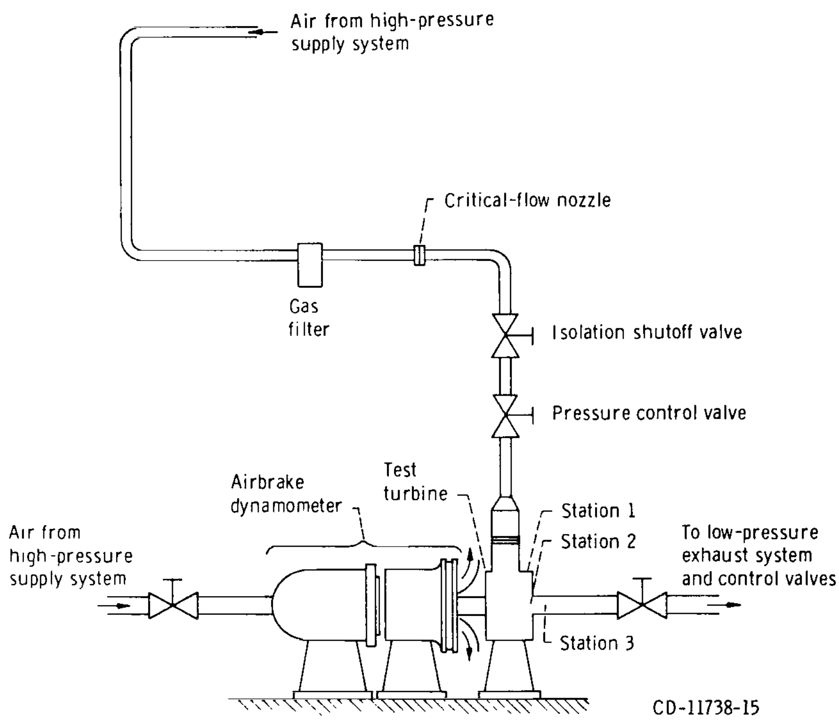


Figure 4. - Experimental equipment.

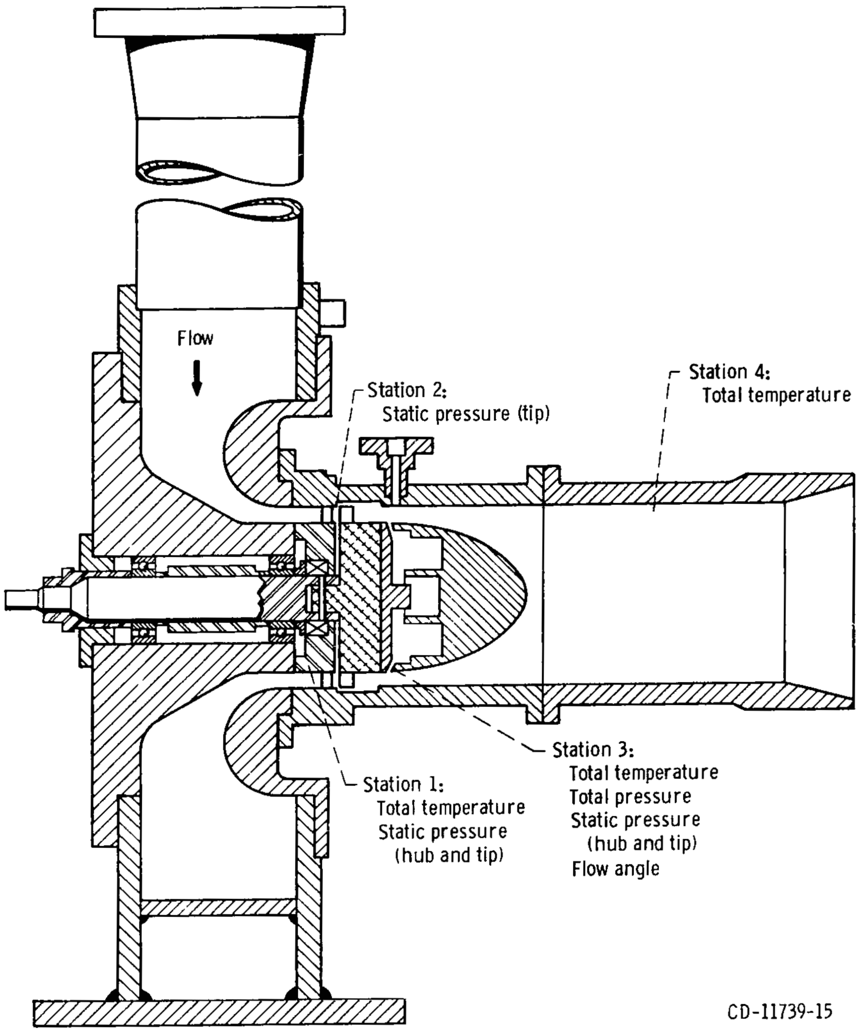


Figure 5. - Schematic of turbine.

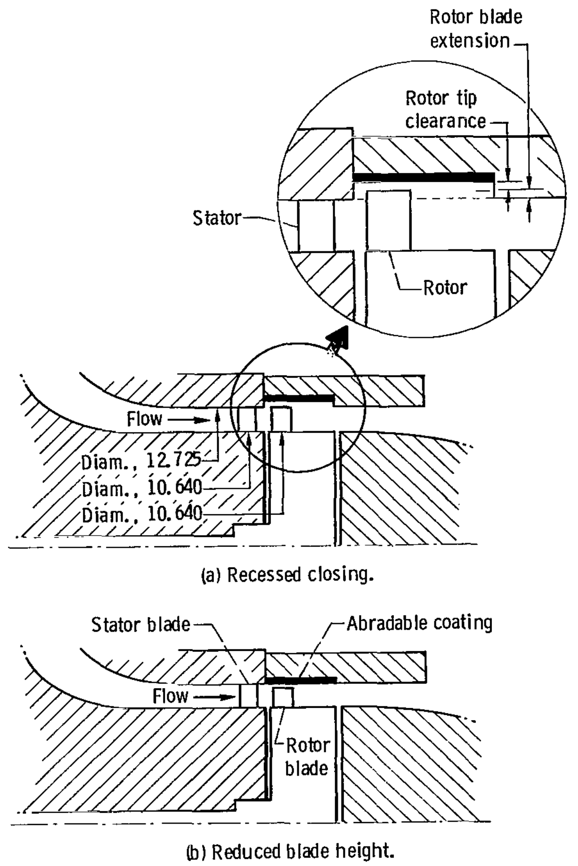


Figure 6. - Schematic of tip clearance configurations investigated. (Dimensions in cm.)

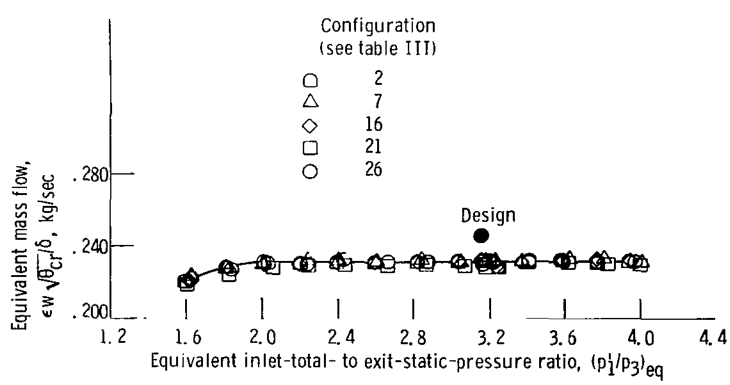


Figure 7. - Variation of mass flow with pressure ratio for recessed casing and reduced-blade height configurations.

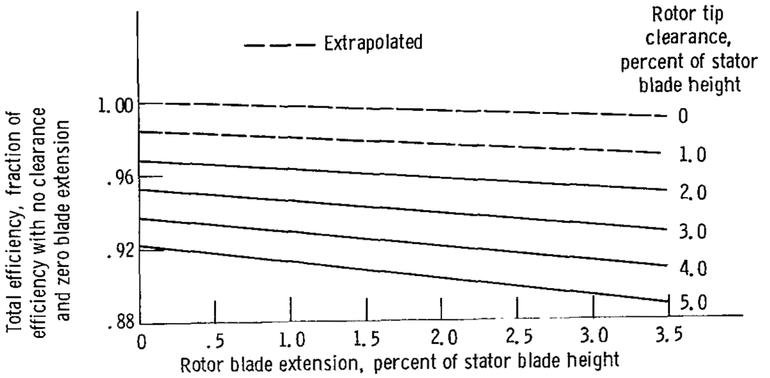


Figure 8. - Change in total efficiency with rotor blade extension for lines of constant rotor tip clearance. Data at design equivalent speed and design total pressure ratio.

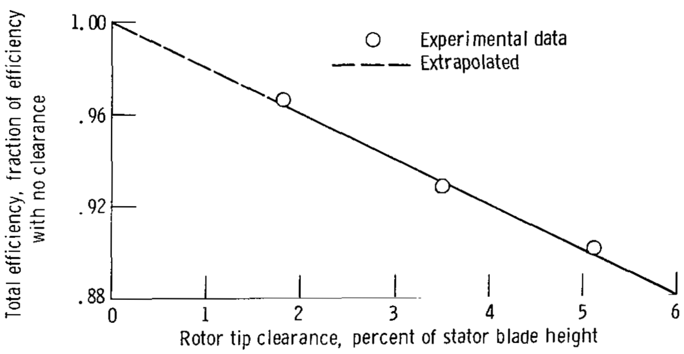
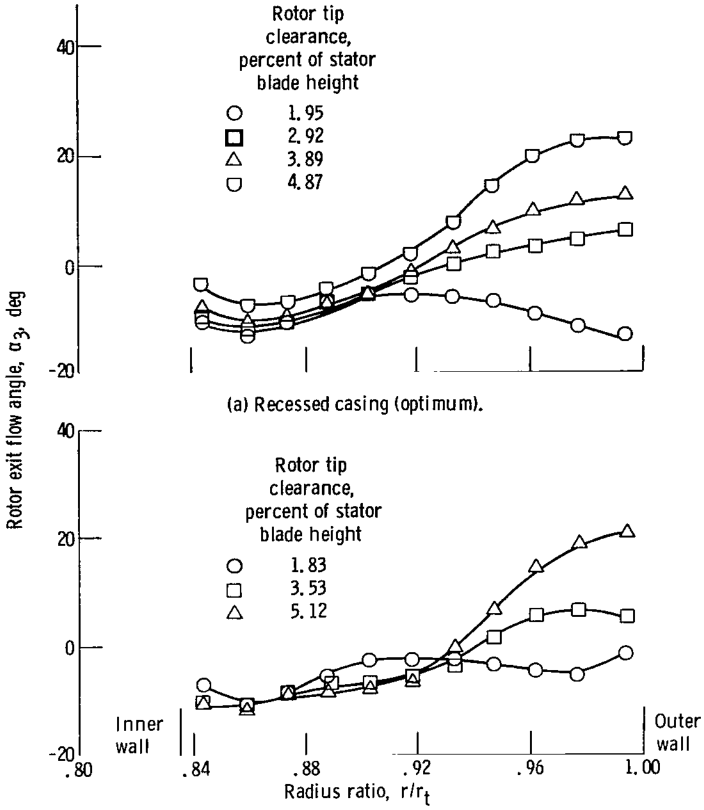
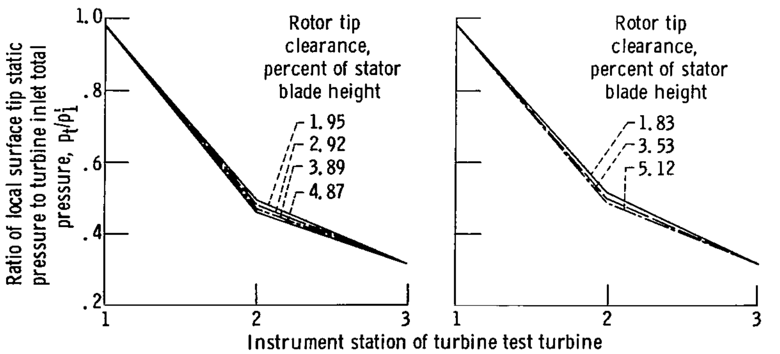


Figure 9. - Change in total efficiency with rotor tip clearance for the reduced blade height configurations. Data at design equivalent speed and design total pressure ratio.



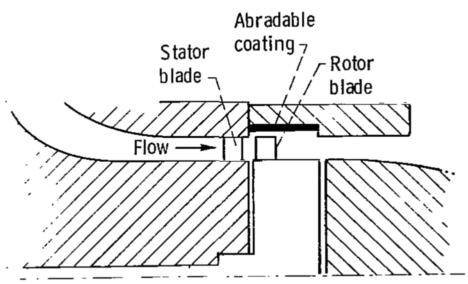
(b) Reduced blade height.

Figure 10. - Variation of rotor exit flow angle with radius ratio. Data at design equivalent speed and total pressure ratio.

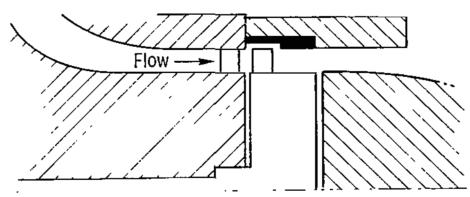


(a) Recessed casing (optimum). (b) Reduced blade height.

Figure 11. - Comparison of static pressure variation through turbine for various radial clearances at design total pressure ratio.



(a) Long axial recessed clearance.



(b) Short axial recessed clearance.

Figure 12. - Schematic of two different axial recessed clearance configurations tested.

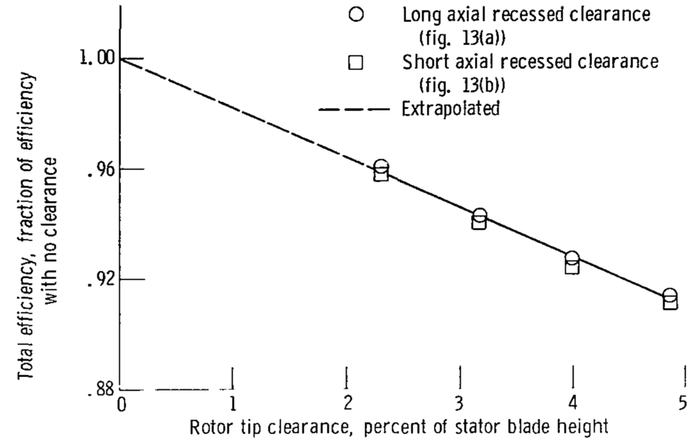
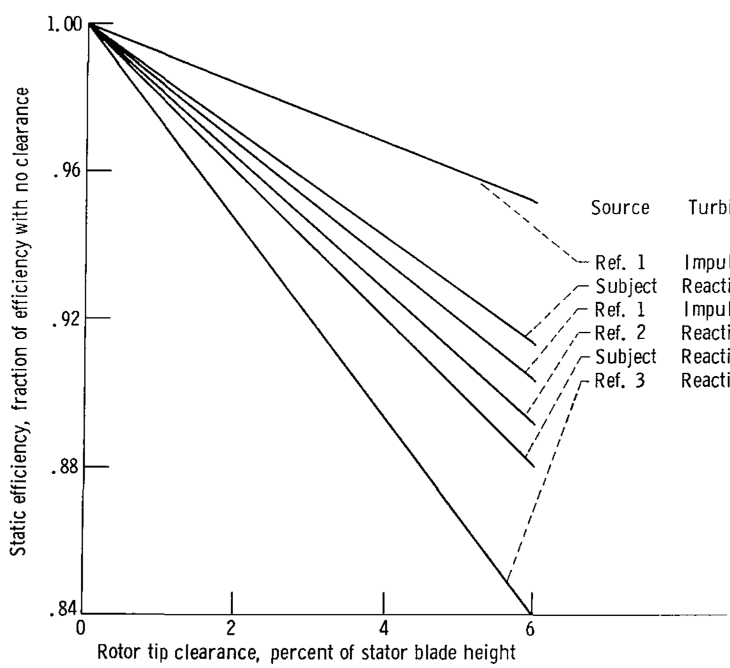


Figure 13. - Comparison of change in total efficiency with rotor tip clearance for two different axial recessed clearance configurations. Data obtained at design equivalent speed and design total pressure ratio.



Source	Turbine	Type of configuration	R_x	$\frac{[1-(\eta/\eta_0)]\times 100}{\%\ \text{tip clearance}}$
Ref. 1	Impulse	RC	0	0.8
Subject	Reaction	RC	.890	1.5
Ref. 1	Impulse	RBH	0	1.6
Ref. 2	Reaction	RBH	.805	1.8
Subject	Reaction	RBH	.890	2.0
Ref. 3	Reaction	RBH	.834	2.7

Figure 14. - Effect of rotor tip clearance on performance for various turbines.

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15. Supplementary Notes

16. Abstract

An experimental investigation was made to investigate the effect of varying the rotor tip clearance of a 12.766-centimeter-tip-diameter single-stage, axial-flow reaction turbine. Two tip clearance configurations, one with a recess in the casing and the other with a reduced rotor blade height, were investigated at design equivalent speed over a range of tip clearance from about 2.0 to 5.0 percent of the stator blade height.

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