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ABSTRACT

An extensive experimental investigation was made to determine the effect of varying the rotor tip clearance of a 12.77-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. For the recessed casing configuration, the optimum rotor blade height was found to be the one where the rotor tip diameter was equal to the stator tip diameter. The tip clearance loss associated with this optimum recessed casing configuration was less than that for the reduced blade height configuration.

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.

With small blade height turbines, geometric similarity with larger turbines becomes difficult to maintain. For example, rotor tip clearances of about 1 to 1.5 percent of the rotor blade height are commonly used in larger turbines. Use of this same tip clearance percentage in a small turbine would require an actual tip clearance of about 0.013 centimeter. This small clearance is 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 rotor blade height are required. Because it is necessary to operate with large 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 four of these investigations are discussed in references 1 to 4. 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 paper summarizes the results of an experimental investigation (ref. 6) to evaluate the effect of varying the rotor tip clearance of a 12.77-centimeter-tip-diameter, single-stage, axial-flow turbine. This turbine was the configuration used in reference 5 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 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 4.

SYMBOLS

p	absolute pressure, N/cm^2
RBH	reduced blade height
RC	recessed casing
R_x	rotor tip reaction $\left(W_{3t}^2 - W_{2t}^2\right)/W_{3t}^2$
r	radius, cm
S	shrouded
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
α	absolute gas flow angle measured from axial direction, deg
β	relative gas flow angle measured from axial direction, deg
η	static efficiency (based on stator inlet-total to rotor exit-static pressure ratio)
η'	total efficiency (based on stator inlet-total to rotor exit-total pressure ratio)

Subscripts:

cr	condition corresponding to Mach number of unity
t	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

TURBINE DESCRIPTION

The turbine used in this experimental investigation was a single-stage, axial-flow 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. 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. These were calculated using the computer program of reference 7. 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. 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 5.

TEST EQUIPMENT AND EVALUATION 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-aligning 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 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 a maximum rotor blade extension into the recess of about 0.035 centimeter, or 3.28 percent of the stator blade height. The rotor blade extension was then reduced in four increments of about 0.008 centimeter each until the rotor tip diameter was equal to the stator tip diameter. For each value of rotor blade extension, performance 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 this paper, 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.

The reduced blade height configuration was tested with the casing diameter set equal to the stator tip diameter and the rotor tip diameter 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.

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.

RESULTS AND DISCUSSION

Turbine Efficiency

Figure 7 shows the change in total efficiency with rotor blade extension for lines of constant rotor tip clearance. This data is at the design total pressure ratio. 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 7 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 with increasing rotor tip clearance. The optimum blade height with the recessed casing configuration was the one where the rotor blade tip diameter was equal to the stator tip diameter (zero blade extension). For this optimum blade height 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 of the stator blade height, 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 one-third increase in loss with amount of blade extension was attributed to pumping and windage 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 a change in rotor blade extension equal to 1 percent of the stator blade height. For the same change in rotor blade extension at a rotor tip clearance of 5 percent, this efficiency loss increased to about 1.0 percent.

Figure 8 shows the change in total efficiency with rotor tip clearance for the reduced blade height configuration. 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 equal to 1 percent of the stator blade height.

This decrease in total efficiency with the reduced blade height configuration was one-third 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 throughflow over the blade tip in the clearance space. For the recessed casing configuration, however, the rotor blade height remained constant as the tip clearance was varied. Thus, only the factors of blade tip unloading and throughflow over the blade tip affected turbine work.

It should be noted that the tip clearance losses mentioned previously in the discussion of figures 7 and 8 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.

Rotor Exit Flow Angle

Figure 9 shows the variation in rotor exit absolute flow angle for the optimum recessed casing and the reduced blade height configuration. This data was obtained from rotor exit radial surveys conducted at design total pressure ratio. Both figures 9(a) and (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 a 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 the reduced blade height configuration is shown in figure 10. 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 rotor 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 Recess

Figures 11 and 12 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 11 shows a schematic of the two different axial lengths of recesses tested. For all of the recessed casing configurations, the recess extended the entire width of the rotor hub (fig. 11(a)). However, for these additional tests, the axial length of the recess was reduced to be slightly greater than the rotor blade axial chord length (fig. 11(b)). This was done to determine if a shorter casing recess would result in a reduced tip clearance loss.

Figure 12 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 the total efficiency between the two configurations over the range of rotor tip clearance investigated.

COMPARISON OF RESULTS WITH OTHER TIP CLEARANCE INVESTIGATIONS

A comparison of the efficiency loss with rotor tip clearance for various tip clearance investigations is shown in figure 13. 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 obtained by extrapolating the data to zero clearance. The legend indicates the type of turbine configuration (impulse or reaction), the degree of tip reaction, the type of tip clearance configuration (shrouded, recessed casing or reduced blade height), and the efficiency decrease for a change in rotor tip clearance equal to 1 percent of the stator blade height.

Included on this curve are the shrouded, 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, the shrouded, impulse turbine of reference 4, and the reduced blade height and optimum recessed casing configurations for the subject turbine.

Figure 13 shows that the tip clearance loss varied from a value of 0.25 for the impulse shrouded turbine of reference 4 to a value of 2.9 for the reaction turbine of reference 3 having a reduced blade height configuration. Two trends are noticeable from this figure. First, for a given level of tip reaction, the optimum tip clearance configurations from a standpoint of having the smallest tip clearance penalty were the shrouded turbines, followed by the recessed casing configurations, and then the reduced blade height configurations. Secondly, except for the reference 3 turbine, the tip clearance loss increased with an increase in the tip reaction for a given tip clearance configuration.

The reason for the much higher tip clearance loss for the reference 3 turbine was attributed to the mass flow characteristics of the turbine. 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 a change in tip clearance equal to 1 percent of the stator blade height. However, the reference 3 turbine experienced a 0.88 percent increase in mass flow for this same change in rotor tip clearance. Thus, for this turbine the mass flow through the stator increased significantly as the radial clearance increased and this, in turn, resulted in greater changes 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 due to the factors of reduced blade area, blade tip unloading, and throughflow over the blade in the clearance space 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 change in tip clearance equal to 1 percent of the stator blade height.

CONCLUDING REMARKS

The results presented in this paper showed two factors that have to be considered in predicting the tip clearance loss for a given design. The tip clearance loss varies over a wide range depending on the tip reaction and the type of tip clearance configuration chosen. In small turbine designs, where the level of tip reaction is generally high and a shrouded rotor configuration is not used, the results from this paper can be condensed to provide simple expressions for the tip clearance loss for reaction turbines having either a reduced blade height or a recessed casing configuration. Expressed as a change in efficiency from a zero tip clearance reference for a change in tip clearance equal to 1 percent of the stator blade height, these losses are 1.5 and 2.0 for a recessed casing and a reduced blade height configuration, respectively.

REFERENCES

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2 Szanca, E. M., Behning, F. P., and Schum, H. J., "Research Turbine for High-Temperature Core Engine Application. II - Effect of Rotor Tip Clearance on Overall Performance," NASA TN D-7639, 1974.

3 Holeski, D. E., and Futral, S. M., Jr., "Effect of Rotor Tip Clearance on the Performance of a 5-Inch Single-Stage Axial-Flow Turbine," NASA TM X-1757, 1969.

4 Haas, J. E.; et al.: Cold-Air Performance of a Tip Turbine Designed to Drive a Lift Fan. IV - Effect of Reducing Rotor Tip Clearance. NASA TP-1126, 1978.

5 Haas, J. E. and Kofskey, M. G., "Cold-Air Performance of a 12.766-Centimeter-Tip-Diameter Axial-Flow Cooled Turbine. I - Design and Performance of a Solid Blade Configuration," NASA TN C-7881, 1975.

6 Haas, J. E. and Kofskey, M. G., "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," NASA TP-1032, 1977.

7 Katsanis, T., "Fortran Program for Calculating Transonic Velocities on a Blade-to-Blade Stream Surface of a Turbomachine," NASA TN D-5427, 1969.

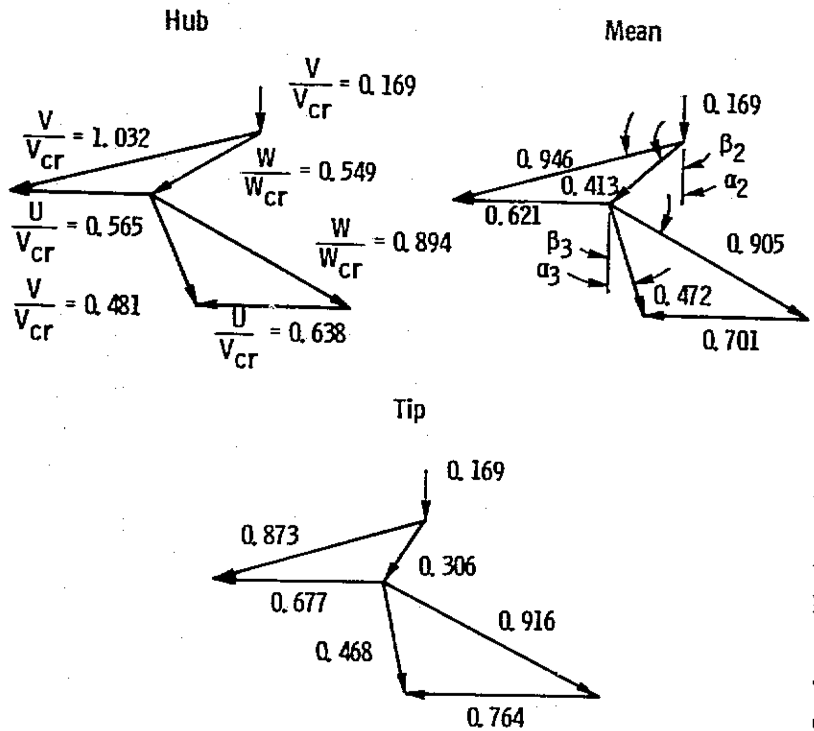
TABLE I. - TURBINE DESIGN CONDITIONS

Parameter	Engine	Equivalent
Turbine inlet temperature, K	1478	288.2
Turbine inlet pressure, N/cm^2	91.2	10.1
Mass flow rate, kg/sec	0.950	0.246
Rotative speed, rpm	70 000	31 460
Specific work, J/g	307.6	62.1
Torque, N-m	39.86	4.64
Power, kW	292.2	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'	0.89	0.85
Work factor, $\Delta V_u/U$	1.67	1.67

TABLE II. - TURBINE DESIGN

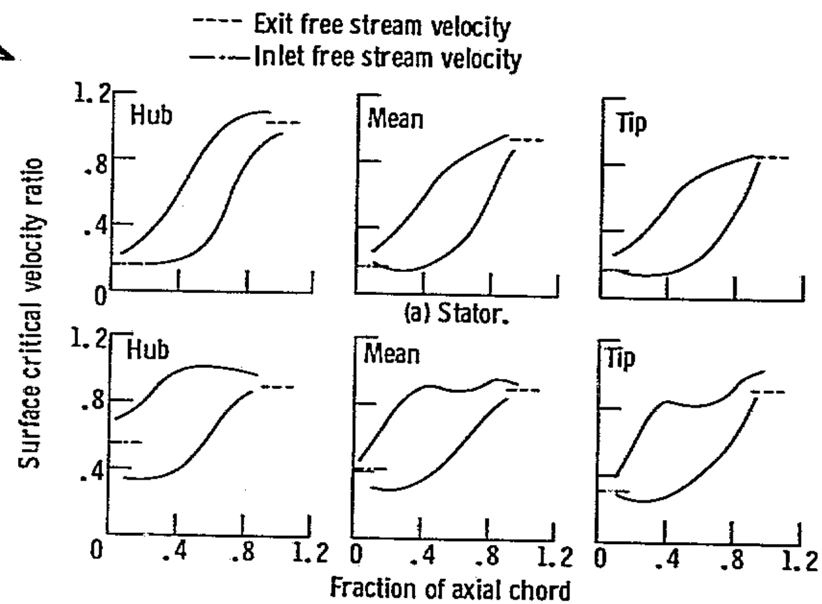
PHYSICAL PARAMETERS

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	1.60	1.68
Aspect ratio	1.00	1.00
Number of blades	56	59
Radius ratio	.835	.835
Blade pitch, cm	.657	.624



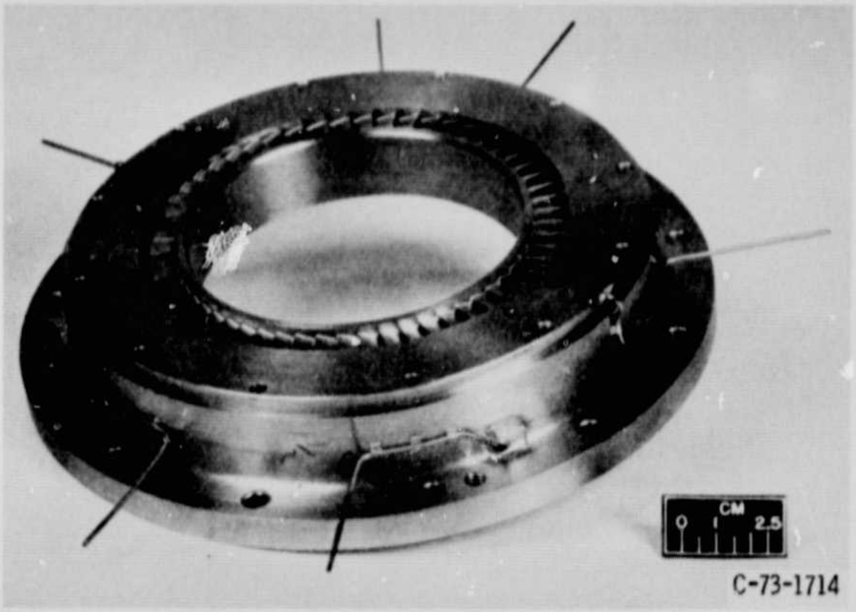
	Hub	Mean	Tip
Radius, cm			
Station 1	5.33	5.86	6.38
Station 2	5.33	5.86	6.38
Station 3	5.33	5.86	6.38
Angle, deg			
α_2, deg	74.2	74.2	74.2
β_2, deg	56.7	48.2	34.4
α_3, deg	-23.3	-17.5	-11.6
β_3, deg	-61.9	-61.9	-61.9

Figure 1. - Design velocity diagrams.

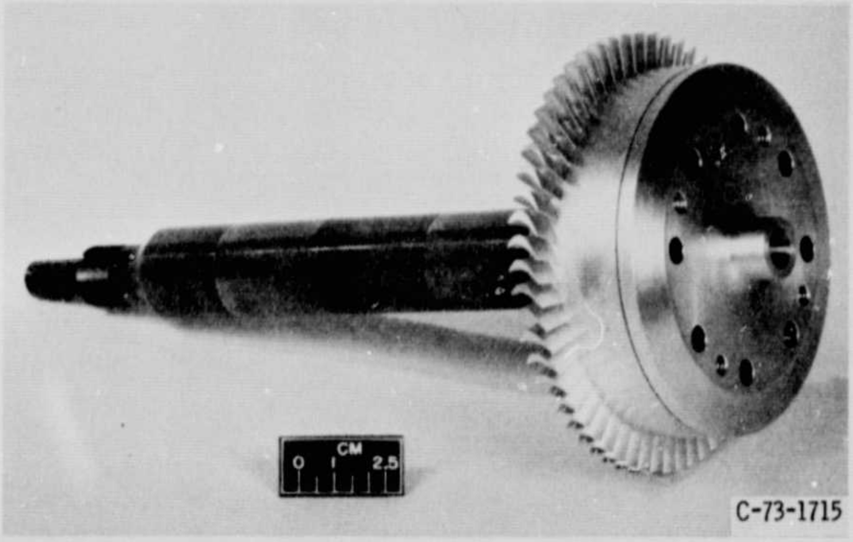


(b) Rotor.

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



(a) STATOR ASSEMBLY.



(b) ROTOR AND SHAFT ASSEMBLY.

Figure 3. - Turbine test hardware.

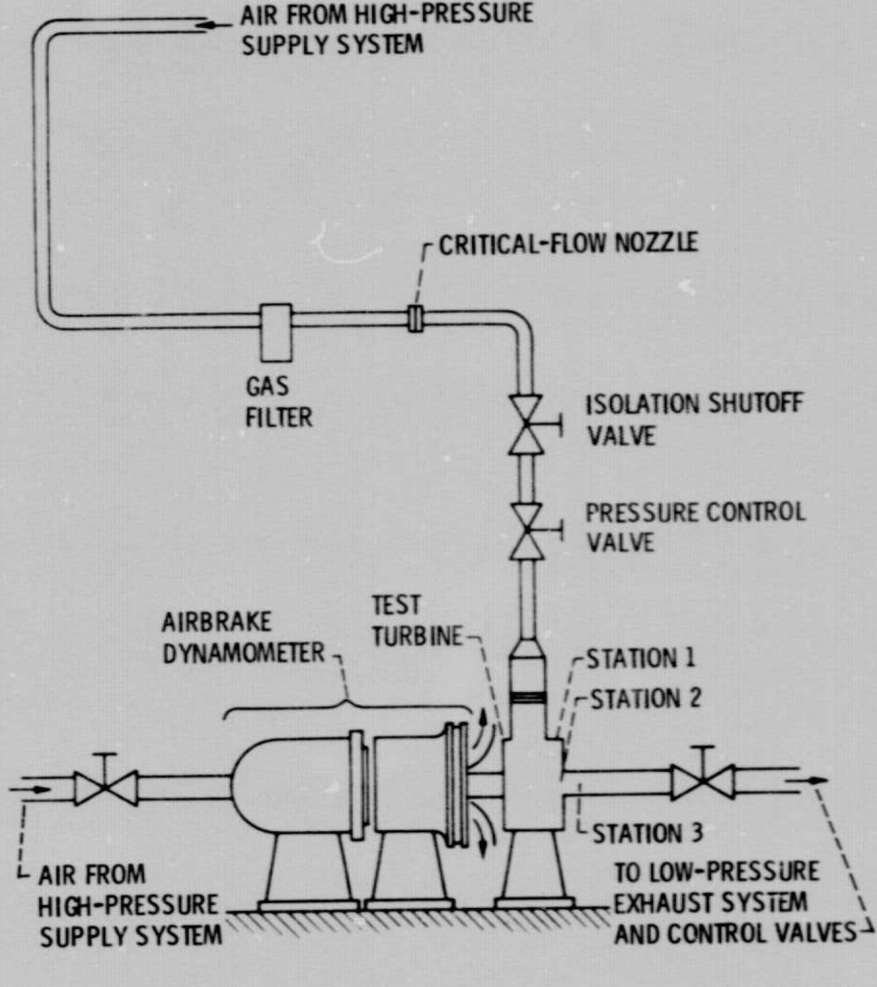


Figure 4. - Experimental equipment.

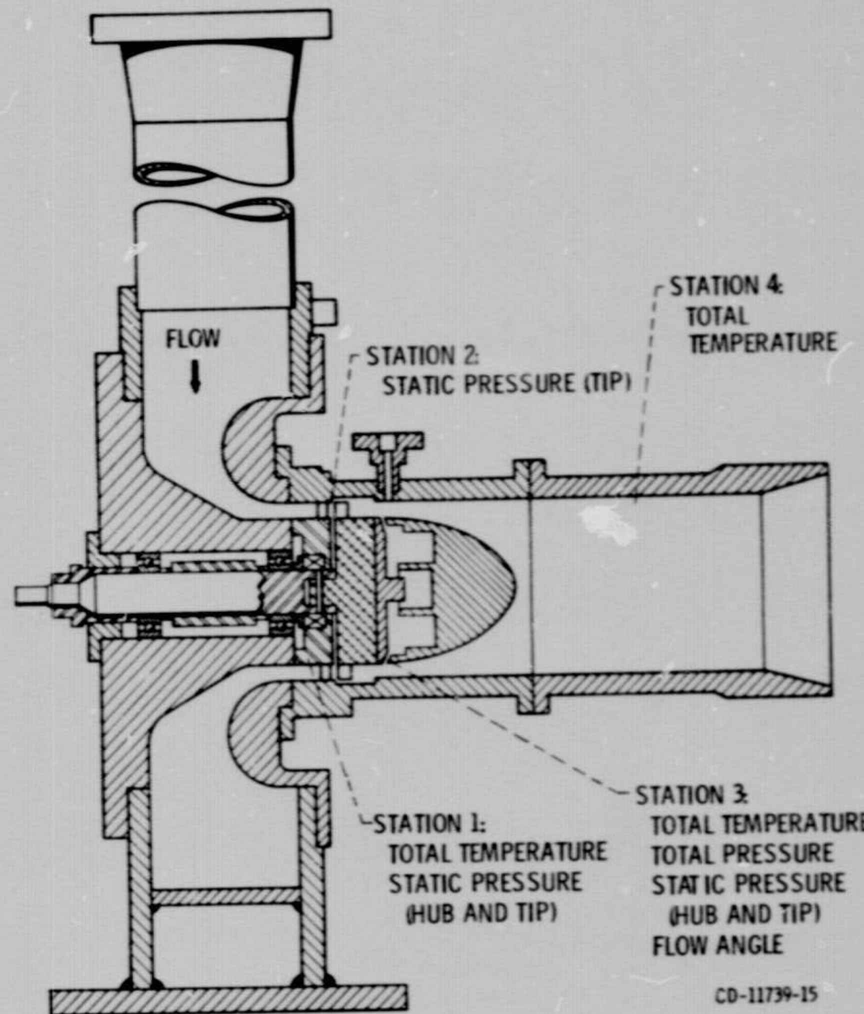
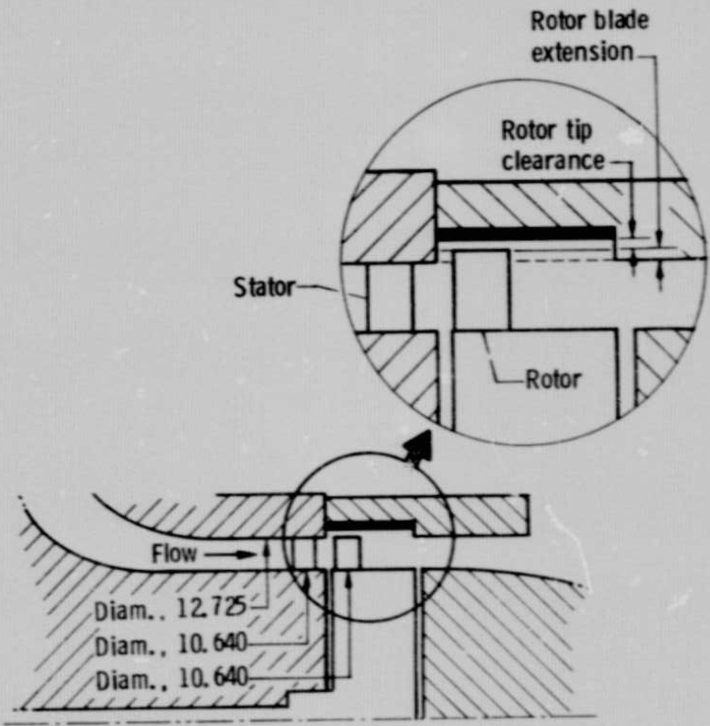
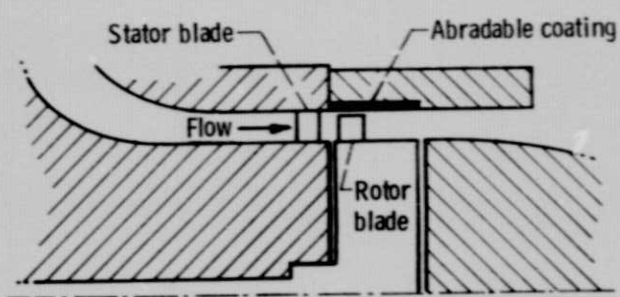


Figure 5. - Schematic of turbine.



(a) Recessed casing.



(b) Reduced blade height.

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

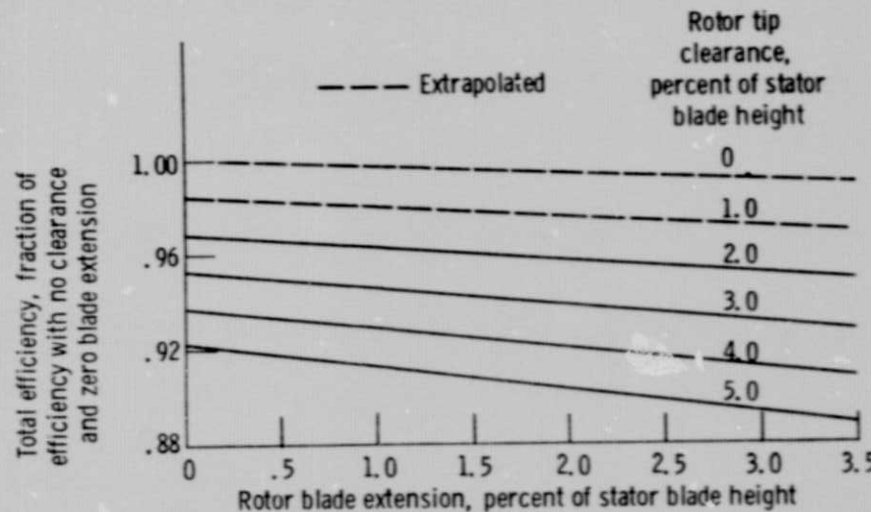


Figure 7. - Change in total efficiency with rotor blade extension for lines of constant rotor tip clearance. Data at design equivalent speed and design total pressure ratio.

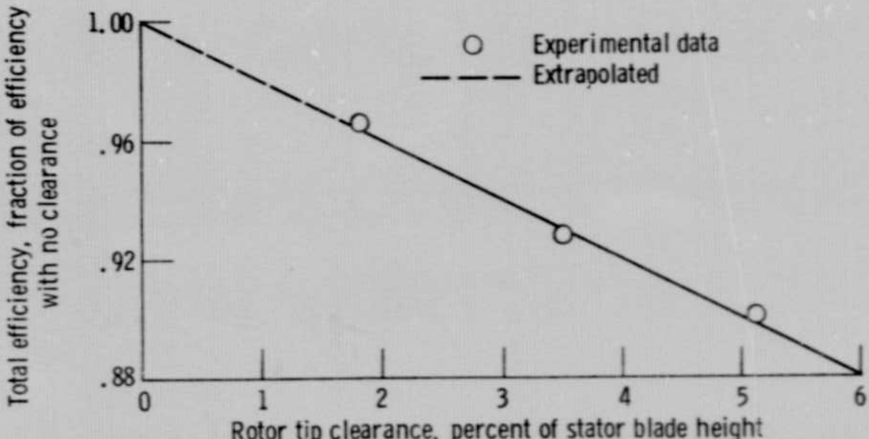
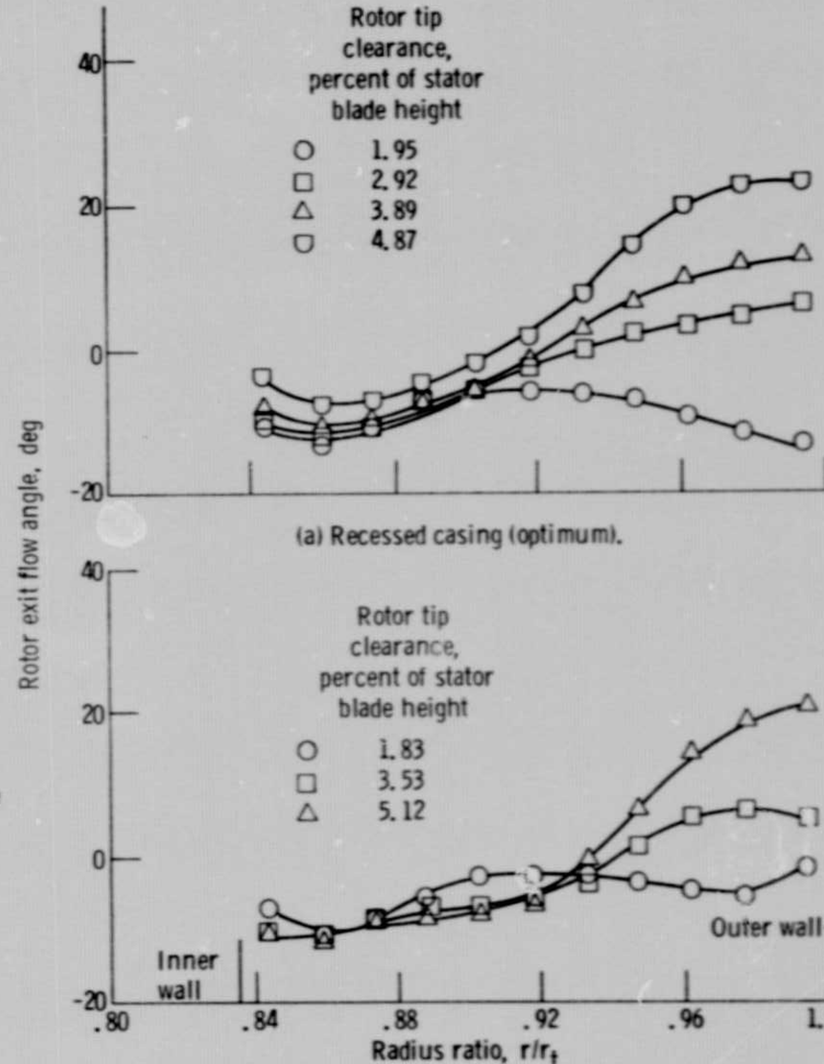


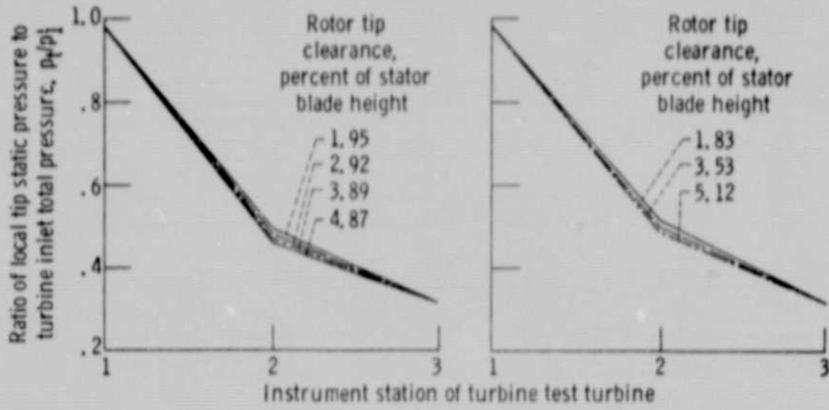
Figure 8. - Change in total efficiency with rotor tip clearance for the reduced blade height configuration. Data at design equivalent speed and design total pressure ratio.



(a) Recessed casing (optimum).

(b) Reduced blade height.

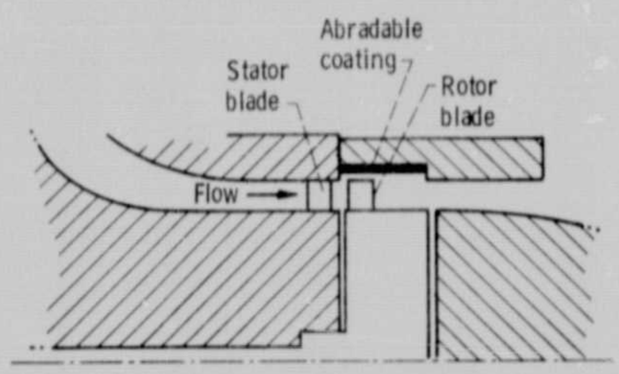
Figure 9. - Variation of rotor exit flow angle with radius ratio. Data at design equivalent speed and total pressure ratio.



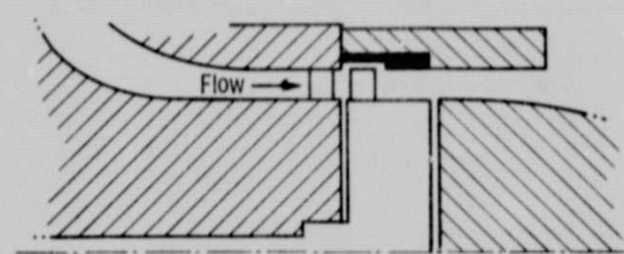
(a) Recessed casing (optimum).

(b) Reduced blade height.

Figure 10. - Comparison of static pressure variation through turbine for various rotor tip clearances at design total pressure ratio.



(a) Long axial recessed clearance.



(b) Short axial recessed clearance.

Figure 11. - Schematic of two different axial recessed clearance configurations tested.

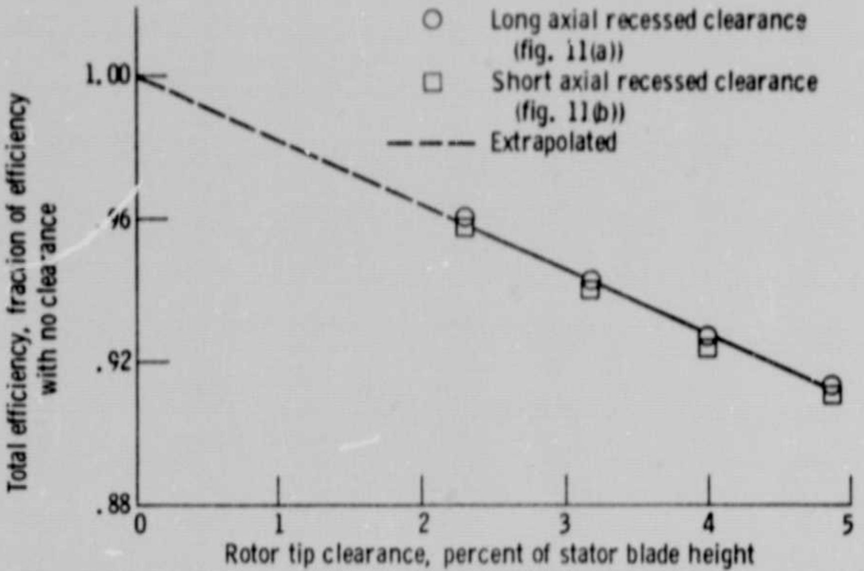


Figure 12. - Comparison of change in total efficiency with rotor tip clearance for two different axial recessed clearance configurations. Data obtained at design equivalent speed and design total pressure ratio.

	Source	Turbine	Type of configuration	R_x	$\frac{(1 - (\eta/\eta_0))\times 100}{\%\ \text{tip clearance}}$
A	Ref. 4	Impulse	S	-0.241	0.25
B	Ref. 1	Impulse	S	0	.3
C	Ref. 1	Impulse	RC	0	.9
D	Subject	Reaction	RC	.890	1.5
E	Ref. 1	Impulse	RBH	0	1.7
F	Ref. 2	Reaction	RBH	.805	1.8
G	Subject	Reaction	RBH	.890	2.0
H	Ref. 3	Reaction	RBH	.834	2.9

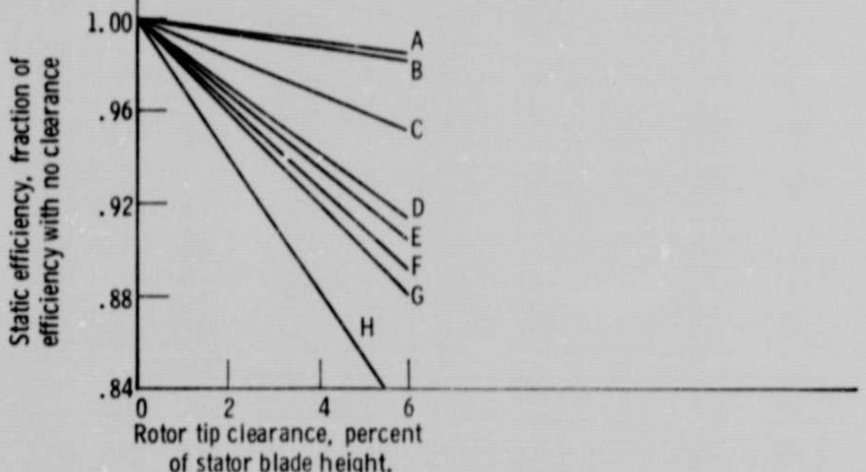


Figure 13. - Effect of rotor tip clearance on performance for various turbines.