NASA TECHNICAL NOTE

NASA TN D-3260

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NATIONAL AERONAUTICS AND SPACE ADMINISTRATION • WASHINGTON, D. C. • FEBRUARY 1966

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SUMMARY

An experimental investigation of a 4.59-inch-diameter radial-inflow turbine was conducted to establish the performance characteristics of radial turbines in this size range. The experimental results are compared with those obtained from a reference turbine of geometrically similar design but with a tip diameter of 6.02 inches. The experimental investigation was conducted in cold argon at the design Reynolds number of 82 200 in order to simulate design hot operation. Reynolds number is defined as weight flow divided by the product of viscosity and rotor tip radius. Tests were made at speeds ranging from 0 to 110 percent of equivalent design speed and pressure ratios from 1.30 to 2.35.

Equivalent specific work and weight flow were 11.7 Btu per pound and 0.605 pound per second at equivalent design speed and pressure ratio. These values are 1.7 and 1.8 percent lower, respectively, than design values. Total and static efficiencies of 0.86 and 0.81, respectively, are approximately 2 points lower than the design values. A radial survey of exit total pressure and flow angle at equivalent design operation indicated a nonuniform work distribution from hub to shroud and amounted to a variation of 10 points in local total efficiency.

Comparison of performance results between the subject and reference turbines indicated that there was a small penalty in turbine performance when turbine size was reduced from a tip diameter of 6.02 to 4.59 inches. Comparisons of total and static efficiencies obtained for the two turbines at their respective design Reynolds numbers showed that efficiencies of the subject turbine were approximately 2 points lower than those of the 6.02-inch reference turbine. The effect of blade-shroud clearances between the two turbines and the accuracy of data measurements may have been contributing factors in the 2 point difference in efficiencies.

INTRODUCTION

NASA is currently studying Brayton-cycle space power systems in the 10-kilowatt power range. Results of analytical studies (refs. 1 and 2) have indicated that the potential of the system with respect to minimum size and weight depends to a great extent upon attaining high turbomachinery efficiency. The studies of reference 1 also indicated that the heat-transfer components of the system benefit from relatively high system pressures. However, reference 2 indicated that, for the 10-kilowatt power level, relatively low system pressure levels are required to obtain turbomachinery sizes large enough to be consistent with the attainment of high efficiency. Therefore, the selection of a system pressure level must be made on the basis of a compromise.

A compromise of this type was used in the selection of the 6.02-inch-tip-diameter compressor-drive turbine of reference 3. The corresponding turbine inlet total conditions were 13.20 pounds per square inch absolute and 1950° R. Results of the experimental investigation, as reported in reference 3, showed that total efficiencies near 0.90 can be obtained for turbines in this size range.

With a turbine performance level established for this size range, it was of interest to determine the effect of size on turbine performance for the 10-kilowatt power level. Use of a smaller size turbine without a significant penalty in performance would permit a higher system pressure level. This would benefit the heat-transfer components with respect to size and weight.

Therefore, a smaller size turbine was designed, fabricated, and tested. Design conditions were the same as for the turbine of reference 3 with the exception of turbine inlet total pressure, which was increased from 13.20 to 22.70 pounds per square inch absolute. With these conditions, the design resulted in a 4.59-inch-tip-diameter radial-inflow turbine which is a 0.762 scale version of the turbine of reference 3.

Tests were made at the design Reynolds number of 82 200 with argon as the working fluid. The investigation covered a range of pressure ratios from 1.30 to 2.35, and for each pressure ratio the speed was varied from 0 to 110 percent of equivalent design speed.

This report presents the results of the investigation of this scaled turbine and compares them with those obtained for the reference turbine. The results are presented in terms of equivalent specific work, equivalent weight flow, equivalent torque, and efficiency. A radial survey of exit flow angle and exit total pressure is also presented at design equivalent pressure ratio and speed.

SYMBOLS

A	flow area, sq in.
g	gravitational constant, 32.174 ft/sec
H'	isentropic specific work based on total-pressure ratio, ft-lb/lb
Δh	specific work, Btu/lb
J	mechanical equivalent of heat, 778.029 ft-lb/Btu
N	turbine speed, rpm
N_s	specific speed, $NQ^{1/2}/60(gH')^{3/4}$
p	pressure, psia
Q	volume flow (based on exit conditions), cu ft/sec
R	universal gas constant, ft-lb/(lb)(^oR)
Re	Reynolds number, $w/\mu r_t$
r	radius, ft
T	absolute temperature, ^oR
U	blade velocity, ft/sec
V	absolute gas velocity, ft/sec
V_J	ideal jet speed corresponding to total- to static-pressure ratio across turbine, $\sqrt{2gJ\,\Delta h_{id}}$, ft/sec
W	relative gas velocity, ft/sec
w	weight flow, lb/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, p_2'/p^*
ϵ	function of γ used in relating parameters to that using air inlet conditions at U. S. standard sea-level conditions, $\frac{\gamma^*}{\gamma}\left[\frac{\left(\frac{\gamma+1}{2}\right)^{\gamma/(\gamma-1)}}{\left(\frac{\gamma^*+1}{2}\right)^{\gamma^*/(\gamma^*-1)}}\right]$
η_s	static efficiency (based on total- to static-pressure ratio across turbine)

η_t total efficiency (based on total- to total-pressure ratio across turbine)

θ_{cr} squared ratio of critical velocity at turbine inlet to critical velocity at U. S. standard sea-level temperature, $\left(V_{cr}/V_{cr}^*\right)^2$

μ gas viscosity, lb/(ft)(sec)

ν blade- to jet-speed ratio (based on rotor inlet tip speed), U_t/V_J

τ torque, in. -lb

Subscripts:

cr condition corresponding to Mach number of unity

id ideal

s shroud

t tip

u tangential component

2 station at turbine inlet

3 station at turbine exit

Superscripts:

' absolute total state

* U.S. standard sea-level conditions (temperature equal to 518.67° R and pressure equal to 14.696 psia)

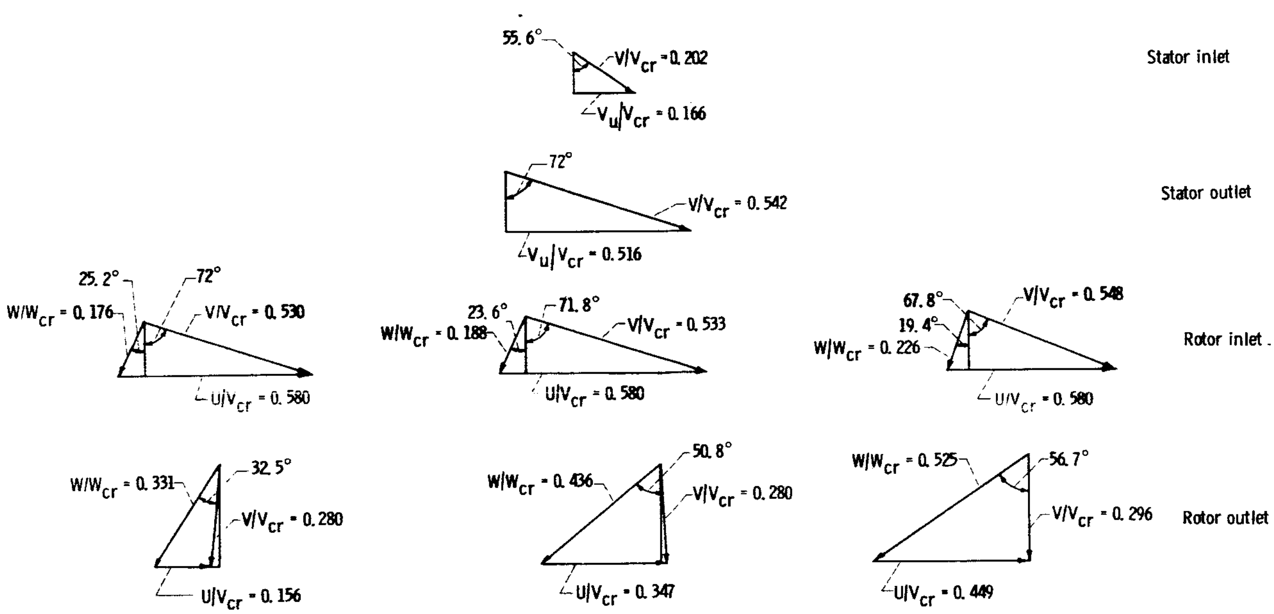
TURBINE DESCRIPTION

Design Requirements

Since the purpose of this investigation was to compare the performance of the 0.762 scale turbine with that obtained for the reference turbine, it was required that the subject turbine be geometrically similar to the reference turbine. All design-point values except inlet total pressure and rotative speed were the same as for the 6.02-inch reference turbine. The turbine inlet pressure was increased from 13.20 (reference turbine) to 22.70 pounds per square inch absolute in order to fulfill the design requirement of equal values of weight flow for the two turbines. The smaller rotor tip diameter and the design requirement of equal blade speeds resulted in a design rotative speed (corresponding to hot operation) of 50 500 rpm for the subject turbine as compared to 38 500 rpm for the reference turbine.

TABLE I. - AIR EQUIVALENT DESIGN VALUES

	Subject turbine	Reference turbine
Equivalent weight flow, $w\sqrt{\theta_{cr}}\,\epsilon/\delta$, lb/sec	0.616	1.063
Equivalent specific work, $\Delta h/\theta_{cr}$, Btu/lb	11.9	11.9
Equivalent speed, $N/\sqrt{\theta_{cr}}$, rpm	29 550	22 527
Equivalent torque, $\tau\epsilon/\delta$, in.-lb	22.12	50.05
Equivalent total- to total-pressure ratio, p_2'/p_3'	1.496	1.496
Equivalent total- to static-pressure ratio, p_2'/p_3	1.540	1.540
Blade- to jet-speed ratio, ν	0.697	0.697
Specific speed, N_s	0.118	0.118



(a) Hub; $r/r_s = 0.347$. (b) 50 Percent streamline; $r/r_s = 0.772$. (c) Shroud; $r/r_s = 1.000$.

Figure 1. - Design velocity diagrams.

Air equivalent (U.S. standard sea level) design values were computed as in reference 3 and are presented in table I together with those for the reference turbine. Rotor inlet tip blade speed was used for the calculation of blade- to jet-speed ratio.

Velocity diagrams were calculated to meet the design work requirement and are presented in figure 1. These diagrams are identical to those for the reference turbine. They indicate a fairly conservative unit, having a relatively low level of velocities and very little exit whirl.

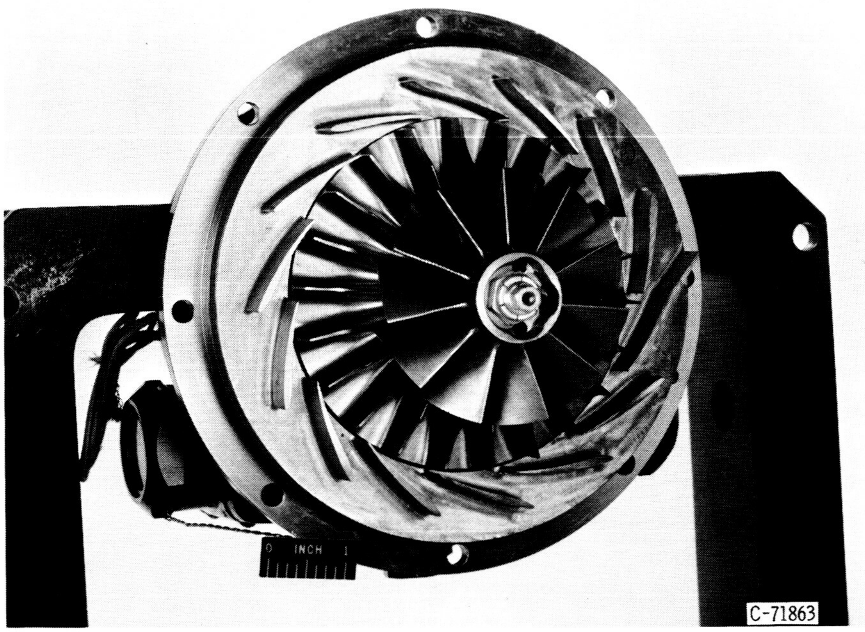


Figure 2. - Turbine stator and rotor assembly.

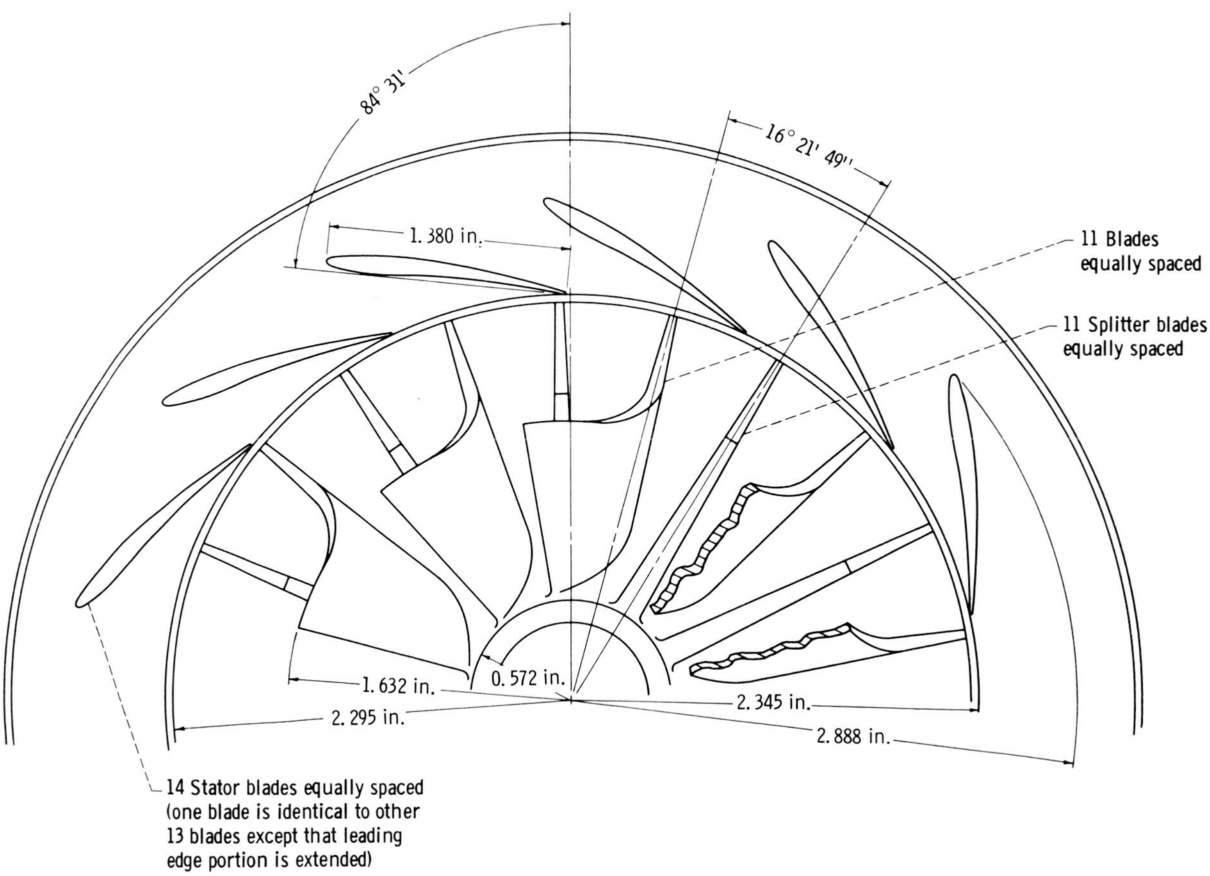


Figure 3. - Turbine stator and rotor.

Turbine Geometry

Figure 2 shows the turbine stator and rotor assembly. There are 14 stator blades, one of which has an extended leading edge portion to block the flow from entering the small end of the inlet scroll. As can be seen from figure 2, the rotor assembly includes 11 blades and 11 splitter vanes. These vanes extend over approximately one-third the length of the blade near the leading edge and thereby decrease the blade loading in that region and prevent low-pressure surface velocities at the hub. Major dimensions of the stator and rotor assemblies are given in figure 3, together with a view of the blading.

Although it was intended that the subject turbine be geometrically similar to the reference turbine, some differences existed in the test units. One of these differences was the shroud clearance values, which could have an effect on performance. The rotor axial shroud clearance was 0.011 inch, and the radial shroud clearance of the exducer section was 0.015. These values, when expressed as a percent of blade height are 1.9 and 1.4 for the axial and radial clearances, respectively. The reference turbine had rotor axial and radial shroud clearances of 0.019 and 0.010 inch, respectively. These values are 2.5 and 0.7 percent of the blade height for the axial and radial clearances, respectively.

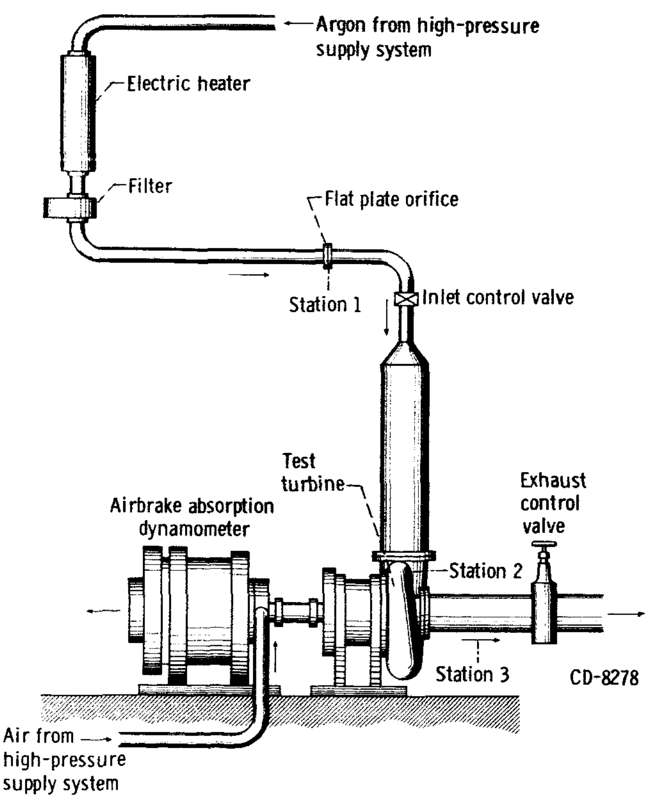


Figure 4. - Experimental equipment.

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 is given in figure 4.

Pressurized argon from a trailer supply was passed through an electric heater, filters, a weight flow measuring station, and a pressure control valve before entering the turbine. The argon, after passing through the turbine, was exhausted into the labo-

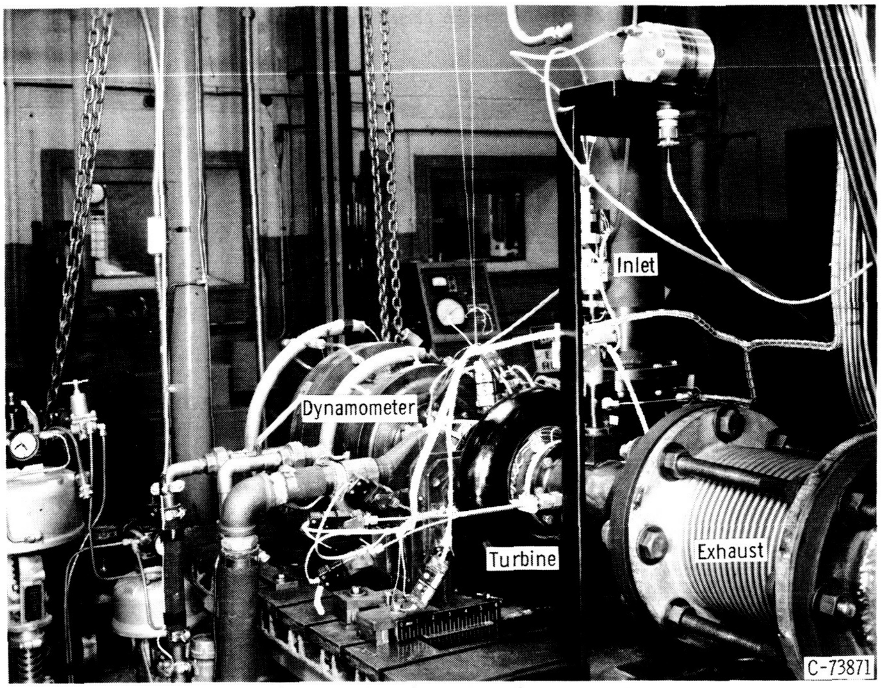


Figure 5. - Experimental turbine test setup.

ratory low-pressure exhaust system. With a fixed inlet pressure, a remotely operated valve in the exhaust line was used to obtain the desired pressure ratio across the turbine. The instrument measuring stations are shown in figure 4. Weight flow was measured at station 1 with a sharp-edged orifice that was calibrated and installed according to the ASME Power Test Code. Four static-pressure taps, a total-pressure probe, and a total-temperature rake consisting of three thermocouples were located at station 2, the turbine inlet. One thermocouple from the total-temperature rake and the total-pressure probe was used for setting and monitoring the turbine inlet conditions. At the turbine exit, station 3, the instrumentation consisted of six static pressure taps, three each at the inner wall and the outer wall, a total temperature rake consisting of three thermocouples, and a three-tube probe. The center tube in the probe was used to record exit total pressure, and the other two were used for angle measurements. One thermocouple from the total-temperature rake and one outer wall static-pressure tap were used for monitoring the turbine exit conditions. In addition to the instrumentation at these measuring stations, two static-pressure taps were located in the plane of the trailing edge at the stator exit, one in each adjacent channel at the midchannel point. Figure 5 is a photograph of the turbine test facility.

The rotational speed of the turbine was measured with an electronic counter in conjunction with a magnetic pickup and a shaft-mounted gear. An airbrake dynamometer, which is similar to that of reference 3, absorbed the power output of the turbine. A description of the airbrake dynamometer and its operation is given in reference 4. The torque force was measured with a commercial strain-gage load cell.

The subject turbine was operated at a Reynolds number of 82 200, which is near the design Reynolds number of 83 200 corresponding to hot design operation. Reynolds number is defined herein as $Re = w/\mu r_t$. Performance data were taken at nominal inlet total conditions of 610° R and 5.6 pounds per square inch absolute. Data were obtained over a range of total- to static-pressure ratios from 1.30 to 2.35 and a speed range from 0 to 110 percent of equivalent design.

In order to obtain friction torque of the bearings and seals, air was evacuated from the turbine with the inlet and exhaust valves closed. This was done in order to minimize windage losses. The friction torque was then obtained by motoring the rotor and shaft over the range of speeds covered in this investigation. A friction torque value of approximately 0.5 inch-pound, which corresponds to 5.7 percent of the turbine work, was obtained at design-point operation. Friction torque was added to the shaft torque when turbine efficiency was determined.

The turbine was rated on the basis of both total and static efficiency. The total pressures were calculated from weight flow, static pressure, total temperature, and flow angle from the following equation:

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

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

RESULTS AND DISCUSSION

The results of this investigation are presented in three sections. The first section presents the overall performance of the 4.59-inch turbine. The second presents the internal flow characteristics resulting from an exit radial survey and static-pressure variation through the turbine at design operation. The third section compares the performance of the 6.02-inch reference turbine with that of the subject turbine at their respective design Reynolds numbers.

Overall Performance

Performance results in argon for the subject turbine are presented in figures 6 to 11. The argon data have been corrected to equivalent air conditions to facilitate a

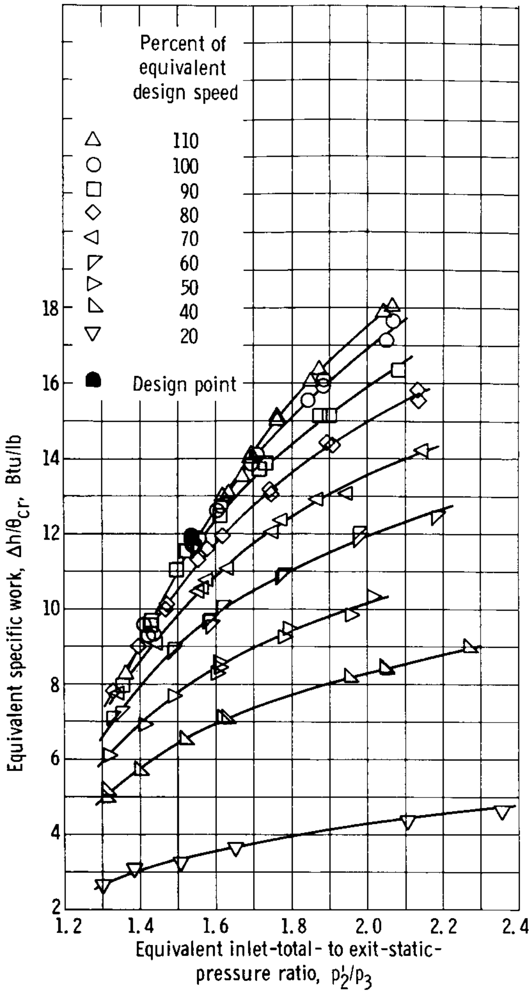


Figure 6. - Variation of specific work with pressure ratio and speed. Data corrected to equivalent air conditions.

direct comparison with the 6.02-inch reference turbine.

Figure 6 presents the variation of equivalent specific work $\Delta h/\theta_{cr}$ with total- to static-pressure ratio for lines of constant speed. The equivalent specific work was approximately 11.7 Btu per pound at equivalent design speed and pressure ratio. This is approximately 1.7 percent lower than the design value. For the speed range 90 to 110 percent, the turbine work was essentially the same, which indicated a constant efficiency region around the design point. For the pressure-ratio range investigated, these curves show that limiting loading was not reached.

In figure 7 equivalent weight flow $w\sqrt{\theta_{cr}}\,\epsilon/\delta$ is plotted against exit-static- to inlet-total-pressure ratio for lines of constant equivalent speed. The reciprocal of the pressure ratio of figure 6 is used in order to separate the speed curves for comparison purposes. At the design pressure ratio of 0.649, the equivalent weight flow was 0.605 pound

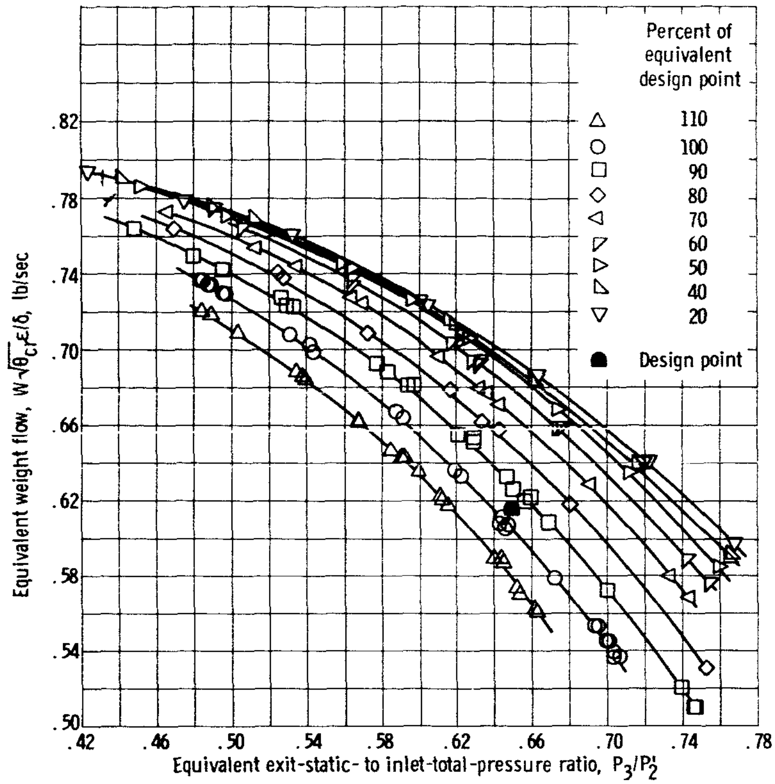


Figure 7. - Variation of weight flow with pressure ratio and speed. Data corrected to equivalent air conditions.

per second, which is approximately 1.8 percent lower than the design value. This difference in weight flow may result from deviation in design values of one or more of the following: stator flow area, rotor flow area, viscous losses, and stator exit static pressure. The trends of the weight flow curves, that is, the variation of weight flow with pressure ratio for lines of constant speed are typical of subsonic turbines.

The variation of equivalent torque $\tau\epsilon/\delta$ with equivalent speed $N/\sqrt{\theta_{cr}}$ for lines of constant equivalent pressure ratio is presented in figure 8. These results are obtained from faired data because the data were taken at constant blade speeds and not at constant pressure ratios. The equivalent torque at design speed and pressure ratio was 21.4 inch-pounds. This is approximately 3.3 percent less than the design value of 22.12 inch-pounds and reflects the lower values of equivalent specific work and weight flow noted previously. Figure 8 shows that zero-speed torque is approximately 1.8 times the torque obtained at design speed and pressure ratio.

Figure 9 presents the dimensionless torque parameter η_s/ν as a function of blade- to jet-speed ratio ν. Design pressure ratio data taken over the range of speeds investigated were used for this curve. At the design blade- to jet-speed ratio of 0.697, a torque parameter value of 1.160 was obtained. This is approximately 2 percent lower than the design value of 1.184. The curves shown in figures 8 and 9 differ significantly from those

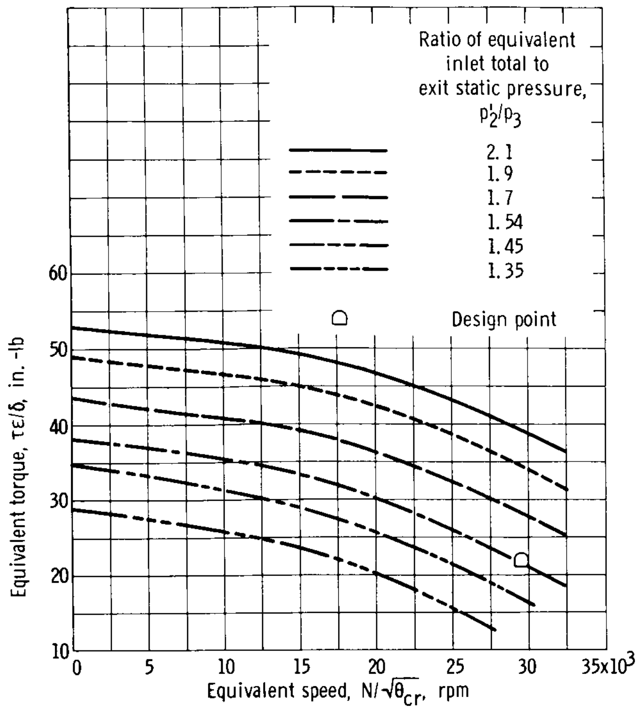


Figure 8. - Variation of torque with speed and pressure ratio. Data corrected to equivalent air conditions.

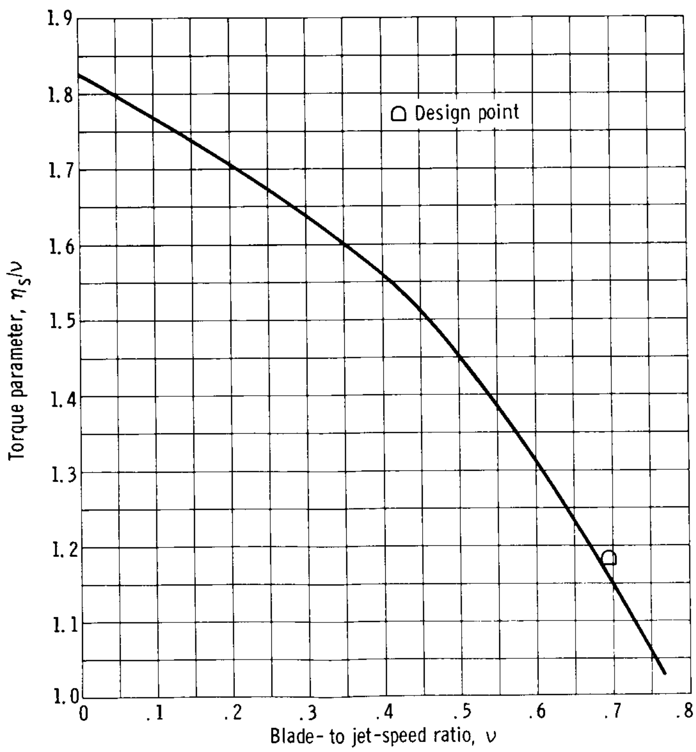


Figure 9. - Torque-speed characteristics for design pressure ratio.

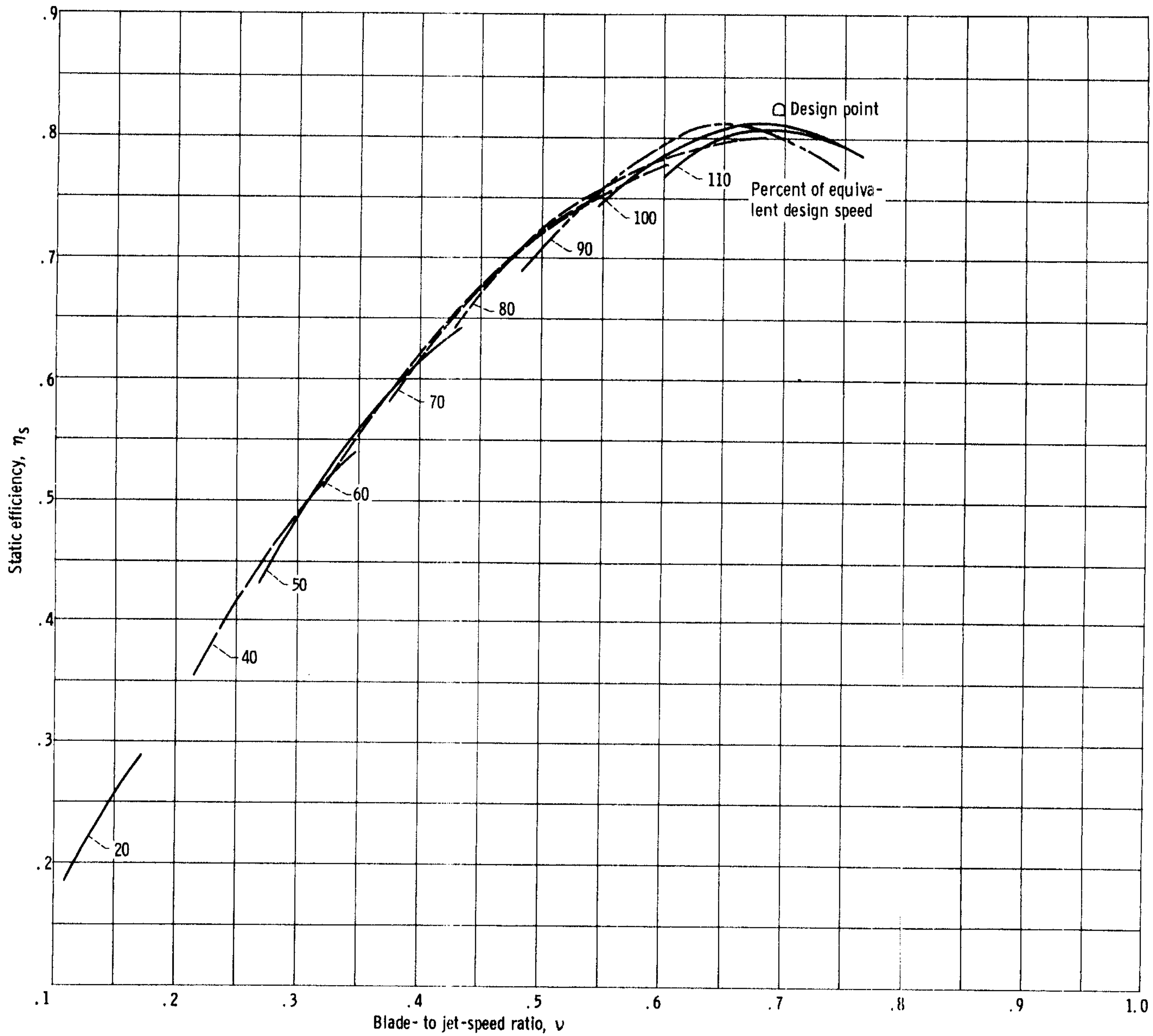


Figure 10. - Variation of static efficiency with blade- to jet-speed ratio.

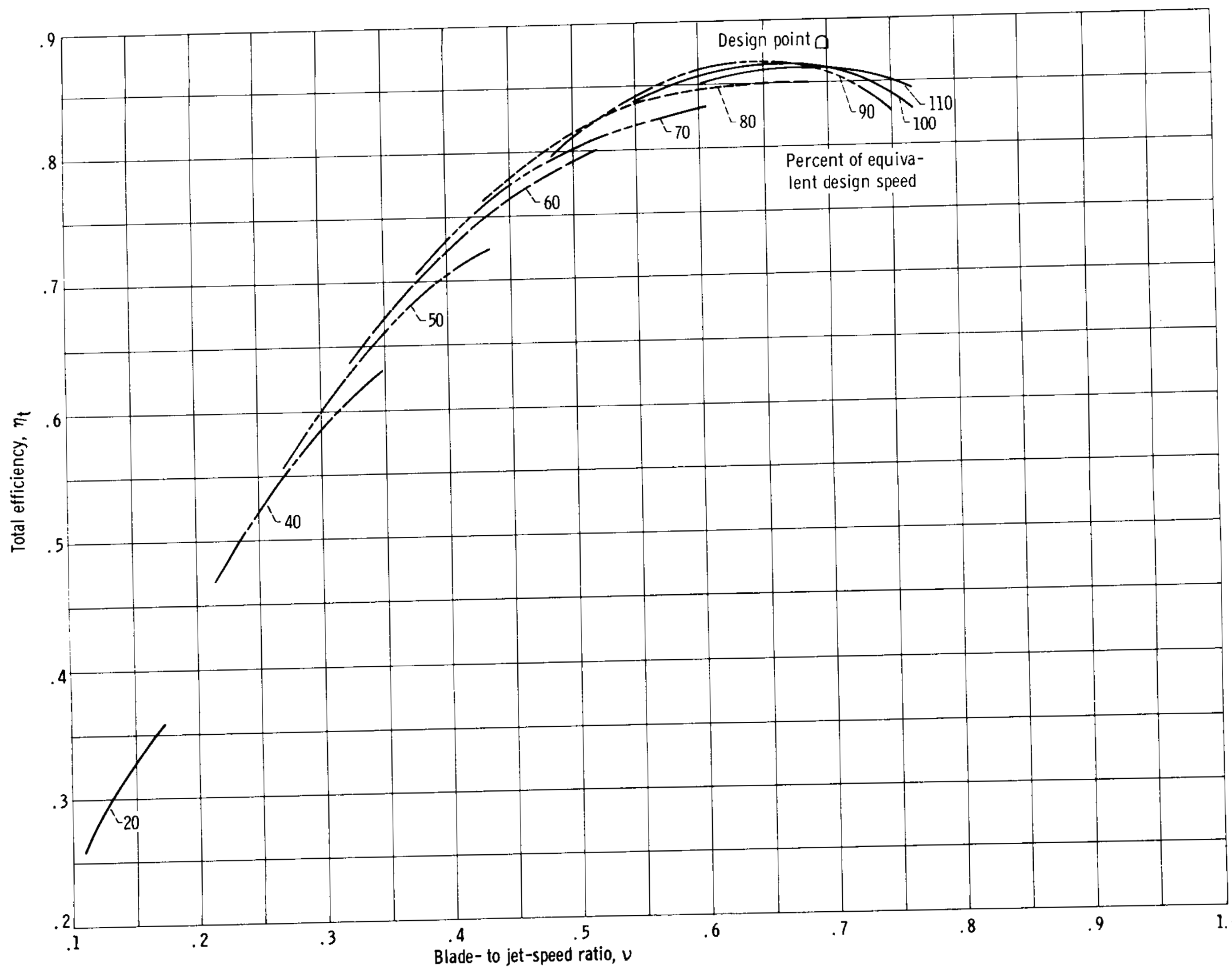


Figure 11. - Variation of total efficiency with blade- to jet-speed ratio.

of axial-flow turbines, which approximate straight lines. The curvature shown in figures 8 and 9 (p. 12) illustrates one of the major differences in performance between axial- and radial-inflow turbines. This is discussed in reference 5, where changes in speed in an axial turbine are compared with corresponding changes in a radial turbine. The analysis in reference 5 showed that, in the radial machine, efficiency decreases more rapidly with deviation from optimum blade- to jet-speed ratio than in the axial machine. This in turn results in the curved torque-speed curves.

Figure 10 (p. 13) presents the variation of static efficiency η_s with blade- to jet-speed ratio. A static efficiency of 0.81 was obtained at design speed and the design blade- to jet-speed ratio of 0.697. This is approximately 1.5 points lower than the design value of 0.824. Figure 10 also shows no appreciable change in efficiency at design blade- to jet-speed ratio as speed was varied from 80 to 110 percent.

Variation of total efficiency with blade- to jet-speed ratio is presented in figure 11. At design blade- to jet-speed ratio the total efficiency was 0.86. This is approximately 2 points lower than the design value of 0.880. There are no appreciable changes in efficiency in the blade- to jet-speed ratio range from 0.60 to 0.72 for speeds of 90, 100, and 110 percent. At design speed it may be noted that the peak efficiency occurs at a slightly lower value of blade- to jet-speed ratio than design. Since the static efficiency was 0.81, 5 points in efficiency were involved in exit kinetic energy. This amounts to approximately 6 percent of the turbine work.

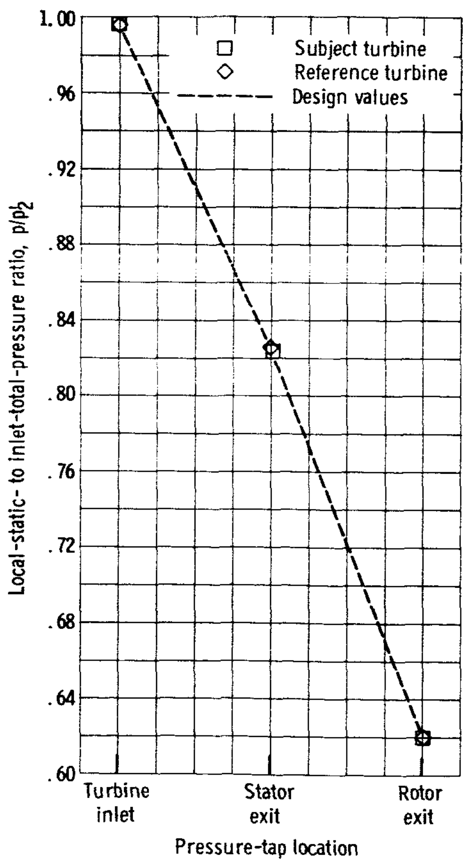


Figure 12. - Variation of static pressure through turbine at equivalent design speed and pressure ratio.

Internal Flow Characteristics

The determination of turbine internal flow characteristics was based on the measured static-pressure distribution through the turbine together with the results of a radial survey of turbine exit total pressure and flow angle.

Figure 12 shows the variation in static pressure through the turbine at equivalent design speed and pressure ratio. The experimental value of exit static pressure is very nearly equal to the design value, which indicates that near-design free-stream gas velocities were obtained. The measured static pressures also indicated that the rotor was operating near the design value of reaction.

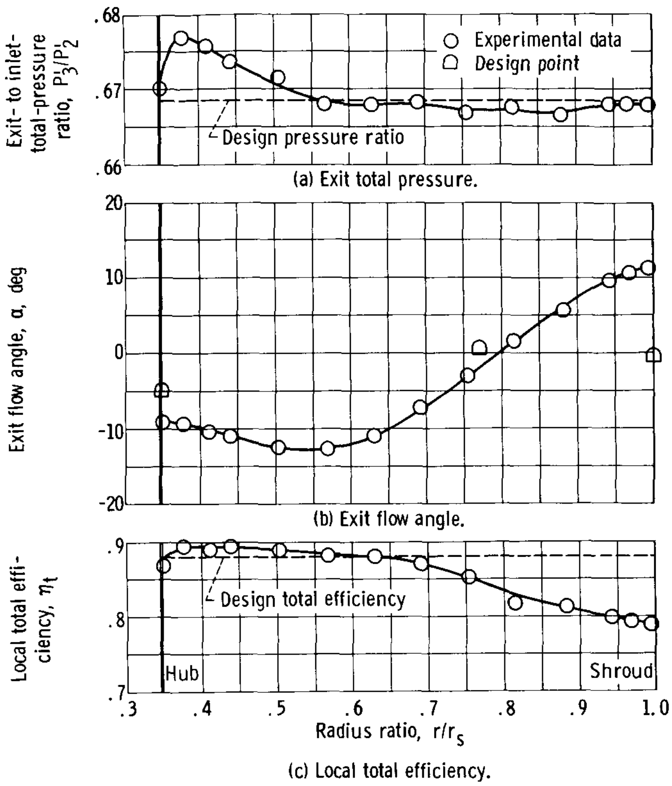


Figure 13. - Variation of exit total pressure, flow angle, and local total efficiency with radius ratio at design-point operation.

The results of a radial survey of turbine exit total pressure and flow angle taken at equivalent design speed and pressure ratio are presented in figure 13. Figure 13(a) presents the variation of exit total pressure with radius ratio. As can be seen from the figure, there were minor variations in total pressure across the passage with small deviations from the design value. Figure 13(b) shows the variation in turbine exit flow angle with radius ratio. Over-turning, as evidenced by the negative angles, was obtained over approximately two-thirds of the passage, and under-turning over the remaining part of the passage near the shroud. Since over-turning was obtained over a large percentage of the passage height and specific work was lower than design, as discussed previously, it followed that the rotor inlet tangential momentum was lower than design. This difference in tangential momentum was determined in the following manner.

The mass-averaged exit tangential momentum was determined from calculations involving the rotor exit survey data and exit static pressures. With this mass-averaged exit tangential momentum value, the rotor inlet tangential momentum was calculated to satisfy measured specific work. The results of these calculations showed that the rotor inlet tangential momentum was approximately 2.7 percent lower than the design value.

The variation in exit flow angle (fig. 13(b)) suggested a nonuniform work distribution over the blade height. Local values of total efficiency were calculated on the basis of the change in tangential momentum through the rotor and the radial distribution of total pressure at the rotor exit. These values of efficiency are shown as a function of radius ratio in figure 13(c). There is a maximum variation of approximately 10 percent in efficiency, with values near design in the hub region. In the region near the shroud, which passes approximately one-half of the turbine weight flow, the values of efficiency are several points below the design value. It was indicated, therefore, that design work was not obtained from that part of the rotor blade near the shroud.

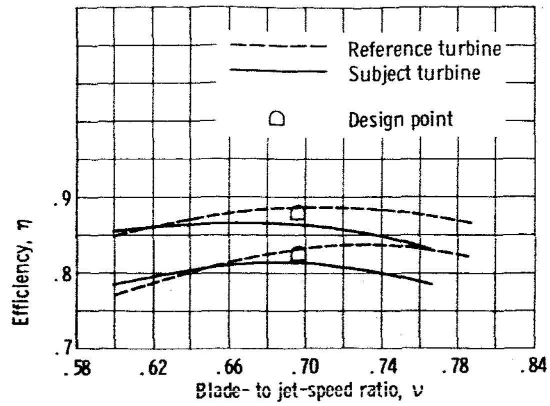


Figure 14. - Comparison of efficiency as function of blade- to jet-speed ratio for design equivalent speed and design Reynolds number.

Performance Comparison With 6.02-Inch Reference Turbine

The following comparison between the subject and reference turbines is made at their respective design Reynolds numbers. The design Reynolds numbers were 82 200 for the subject turbine and 63 400 for the reference turbine.

Figure 14 shows the efficiencies, total and static, for both turbines. The curves represent results obtained at the respective equivalent design speeds for a range of total- to static-pressure ratios. At the design blade- to jet-speed ratio of 0.697, the efficiencies of the subject turbine are approximately 2 points lower than those of the reference turbine. It will be noted that, from a comparison of the two total efficiency curves, the reference turbine was operating at its optimum efficiency point near the design blade- to jet-speed ratio, while the optimum efficiency point of the subject turbine was at a value of blade- to jet-speed ratio which was slightly lower than the design value of 0.697.

Since there were only 2 points difference in efficiency level between the two turbines at design operation, turbine size can be reduced from a tip diameter of 6.02 to 4.59 inches without a large penalty in turbine performance.

The 2 point lower efficiency level of the subject turbine cannot be attributed to the difference in design Reynolds numbers between the two turbines since the subject turbine was operating at a slightly higher Reynolds number. In fact, if the familiar Prandtl 1/5 power law for turbulent boundary layer was used for estimating the variation of viscous loss with Reynolds number, the efficiency of the subject turbine would have been slightly higher at its design Reynolds number.

The static-pressure variation through the turbine of reference 3 is superimposed on that of the subject turbine in figure 12 (p. 15) for comparison purposes. As can be noted, design stator exit static pressure and, therefore, design reaction across the rotor were obtained for both turbines. Hence, the difference in performance cannot be attributed to different degrees of reaction.

At design operation, equivalent weight flows were 2.2 and 1.8 percent lower than the equivalent design values for the reference and subject turbines, respectively. Hence, the flow rates were comparable at their respective design Reynolds numbers. This would indicate that the viscous losses were comparable for both turbines.

Two possible factors which may have contributed to the 2 point difference in performance levels between the two turbines are shroud or tip clearance and the accuracy of

TABLE II. - PERFORMANCE VALUES

	Subject turbine		Reference turbine	
	Design	Argon tests	Design	Argon tests
Reynolds number, Re	83 200	82 200	63 700	63 400
Static efficiency, η_s	0.824	0.81	0.824	0.83
Total efficiency, η_t	0.880	0.86	0.880	0.88
Equivalent specific work, $\Delta h/\theta_{cr}$, Btu/lb	11.9	11.7	11.9	12.0
Equivalent weight flow, $\left(w\sqrt{\theta_{cr}}\,\epsilon\right)/\delta$, lb/sec	0.616	0.605	1.063	1.040
Equivalent torque, $\tau\epsilon/\delta$, in.-lb	22.12	21.4	50.05	49.2

data measurements. As noted in the section Turbine Geometry, shroud clearances when expressed as a percentage of blade height were not comparable between both turbines. The radial shroud clearance for the subject turbine was twice as large as that for the reference turbine. Hence, the difference in shroud clearance may have been a factor affecting the difference in performance levels between the two turbines.

With regard to the accuracy of data for both turbines, the inaccuracies involved in the measurements of torque, friction torque, and weight flow could result in a ±0.5 point error in the experimentally obtained efficiencies for the subject and reference turbines.

Design-point performance results for both turbines are given in table II along with the values assumed in the design of the turbines.

SUMMARY OF RESULTS

An experimental investigation of a 4.59-inch-tip-diameter radial-inflow turbine is presented. This turbine was a 0.762 scale version of the 6.02-inch turbine described in NASA TN D-2987 and was designed for the same application. Performance characteristics were obtained at the design Reynolds number of 82 200, and these results are compared with those of the reference turbine in order to determine the effect of this size reduction on performance.

From this investigation the following results were obtained:

1. At equivalent design speed and pressure ratio, equivalent specific work and weight flow were 11.7 Btu per pound and 0.605 pound per second, respectively. These values are 1.7 and 1.8 percent lower than the design values. The corresponding equivalent torque was 21.4 inch-pounds, which is 3.3 percent lower than the design value. Zero-speed torque was 1.8 times the value obtained at design operation.

2. Total and static efficiencies were 0.86 and 0.81, respectively, at equivalent design speed and blade- to jet-speed ratio. These values are approximately 2 points lower than the design values of 0.880 and 0.824.

3. A radial survey of rotor exit total pressure and flow angle at design equivalent speed and pressure ratio indicated a nonuniform work distribution from hub to shroud. This nonuniform work distribution resulted in a 10 point variation in local total efficiency from hub to shroud. Results based on the calculation of rotor exit mass-averaged whirl momentum and the measured specific work indicated that the whirl momentum at the rotor inlet was approximately 2.7 percent lower than the design value.

4. The effect of the turbine size on performance, for the size range investigated, was not large. The performance level of the subject turbine was approximately 2 points lower than that obtained for the 6.02-inch reference turbine. The effects of the differences in shroud clearances between the two turbines and the accuracy of data measurements may have been the contributing factors in the 2 point difference in efficiency between the two turbines.

Lewis Research Center,
National Aeronautics and Space Administration,
Cleveland, Ohio, November 30, 1965.

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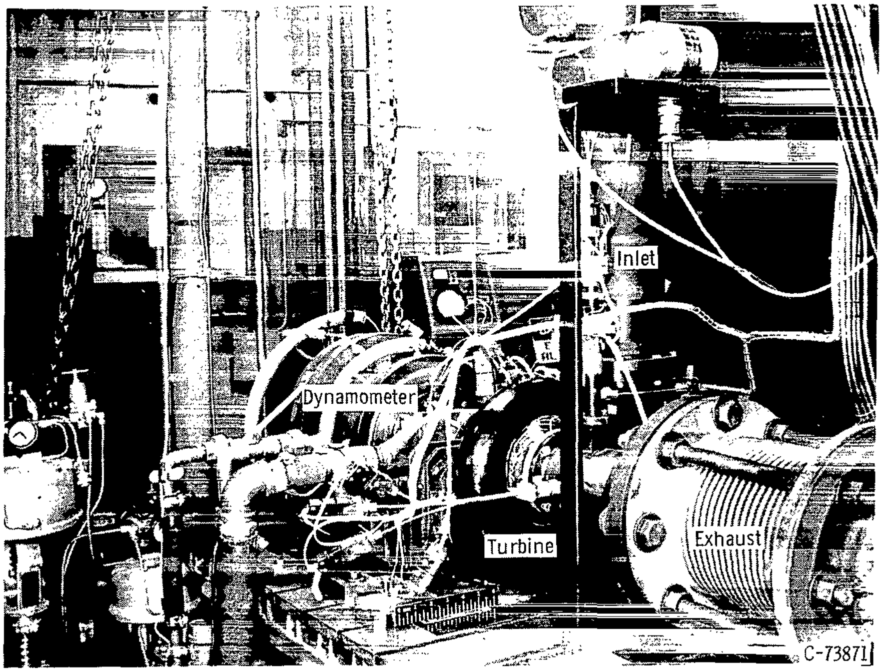

Figure 5. - Experimental turbine test setup.

ratory low-pressure exhaust system. With a fixed inlet pressure, a remotely operated valve in the exhaust line was used to obtain the desired pressure ratio across the turbine. The instrument measuring stations are shown in figure 4. Weight flow was measured at station 1 with a sharp-edged orifice that was calibrated and installed according to the ASME Power Test Code. Four static-pressure taps, a total-pressure probe, and a total-temperature rake consisting of three thermocouples were located at station 2, the turbine inlet. One thermocouple from the total-temperature rake and the total-pressure probe was used for setting and monitoring the turbine inlet conditions. At the turbine exit, station 3, the instrumentation consisted of six static pressure taps, three each at the inner wall and the outer wall, a total temperature rake consisting of three thermocouples, and a three-tube probe. The center tube in the probe was used to record exit total pressure, and the other two were used for angle measurements. One thermocouple from the total-temperature rake and one outer wall static-pressure tap were used for monitoring the turbine exit conditions. In addition to the instrumentation at these measuring stations, two static-pressure taps were located in the plane of the trailing edge at the stator exit, one in each adjacent channel at the midchannel point. Figure 5 is a photograph of the turbine test facility.

The rotational speed of the turbine was measured with an electronic counter in conjunction with a magnetic pickup and a shaft-mounted gear. An airbrake dynamometer, which is similar to that of reference 3, absorbed the power output of the turbine. A description of the airbrake dynamometer and its operation is given in reference 4. The

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