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EXPERIMENTAL EVALUATION OF A 3.50-INCH RADIAL TURBINE DESIGNED FOR A 10-KILOWATT SPACE POWER SYSTEM

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Cleveland, Ohio

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16. Abstract

Tests were conducted in argon at the design Reynolds number of 109 000 and over a range of Reynolds number from 68 900 to 219 900. Performance results at design Reynolds number are presented in terms of equivalent specific work, torque, mass flow rate, and efficiency for a range of speeds and pressure ratios. A comparison of turbine efficiency is made with those of 6.02-in. (15.29 cm) and 4.59-in. (11.66 cm) radial-inflow turbines, which were geometrically similar to the 3.50-in. (8.89-cm) diameter turbine.

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SUMMARY

An experimental investigation of a 3.50-inch (8.89-cm) diameter radial-inflow turbine was made to determine the performance characteristics of a radial turbine in this small size range. The experimental results are compared with those obtained from two turbines of geometrically similar design but with tip diameters of 6.02 and 4.59 inches (15.29 and 11.66 cm). The investigation was conducted in cold argon at design Reynolds number and over a range of Reynolds number.

Equivalent specific work of 11.96 Btu per pound (27.84 J/g), which was approximately the design value, was obtained at design Reynolds number and design-point operation. At this point of operation the equivalent mass flow rate was 0.341 pound per second (0.155 kg/sec), which is 4.7 percent lower than design and is attributable to flow areas less than design. Total and static efficiencies of 0.88 and 0.83, respectively, agree closely with the design values. Total efficiency increased from 0.87 to 0.91 as the Reynolds number increased from 68 900 to 219 900.

Comparison of the performance results between the subject turbine and the two reference turbines indicated that, within experimental accuracy, there were no significant changes in turbine performance when turbine size was reduced from a tip diameter of 6.02 inches (15.29 cm) to 3.50 inches (8.89 cm).

INTRODUCTION

The space-power technology program being conducted at NASA Lewis has included the experimental study of several turbines for potential Brayton cycle systems. The initial studies involved turbines that were designed to match the requirements of a two-shaft 10-kilowatt (shaft) power system. The experimental study of a 6.02-inch (15.29-cm) radial-inflow compressor-drive turbine designed for this application (ref. 1) showed that total efficiencies near 0.90 could be obtained for turbines of this size.

It was of interest to determine the effect of reduction in size on turbine performance while maintaining the 10-kilowatt power level requirement. Use of a smaller turbine without a significant penalty in performance would permit a higher system pressure level for the same power output. This would benefit the heat-transfer components with respect to size and weight. Therefore, a 0.76 scale version of the turbine of reference 1 was designed, fabricated and tested (ref. 2). The hot design conditions were the same for both turbines except that the inlet total pressure was increased from 13.20 to 22.70 psia (9.10 to 15.65 N/cm^2). The performance level of this 0.76 scale version (4.59-in. (11.66-cm) tip diameter), was about 2 points lower than that obtained for the original turbine of reference 1. The effects of the differences in shroud clearances between the two turbines as well as inaccuracy of data measurements could have accounted for the difference in efficiency.

An interest in determining whether a further reduction in turbine size would have any appreciable effect on performance resulted in the fabrication and experimental investigation of a third turbine. The tip diameter for this turbine was reduced to 3.50 inches (8.89 cm). The hot design conditions were the same as the turbines of references 1 and 2 with the exception that the inlet total pressure was increased to 39.02 psia (26.90 N/cm^2 abs). Tests were made with argon as the working fluid at an inlet temperature of 578° R (321 K). Performance was first determined at an inlet total pressure of 9.0 psia (6.2 N/cm^2), which corresponds to design Reynolds number at equivalent design speed and pressure ratio. The effect of Reynolds number on performance was then determined by operating the turbine over a range of inlet total pressures from about 6.0 to 19.0 psia (4.1 to 13.1 N/cm^2) with corresponding Reynolds number from 68 900 to 219 900. At each inlet total pressure, data were obtained over a range of blade-jet speed ratio by varying speed and/or pressure ratio.

This report presents the performance of this scaled turbine and compares it with that obtained for the two reference turbines. The results for the subject turbine are presented in terms of equivalent specific work, equivalent mass flow rate, equivalent torque, and efficiency. Included are the results of a radial survey of rotor-exit flow angle, total temperature, and total pressure at design-point operation.

SYMBOLS

A	flow area, in.^{2}; cm^2
g	dimensional constant, 32.174 ft/sec^2; 9.807 m/sec^2
H	isentropic specific work (based on total pressure ratio), ft-lb/lb; J/g
Δh	specific work, Btu/lb; J/g

N	turbine speed, rpm
N_s	specific speed, $NQ^{1/2}/H^{3/4}$, (rpm)(ft)$^{3/4}$/(sec)$^{1/2}$; (rad)(m)$^{3/2}$(kg)$^{3/4}$/(sec)$^{3/2}$(J)$^{3/4}$
p	pressure, psia; N/cm^2
Q	volume flow (based on exit conditions), ft^3/sec; m^3/sec
R	gas constant, (ft-lb)/(lb)($^{\circ}$R); (N)(m)/(kg)(K)
Re	turbine Reynolds number, $w/\mu r_t$
r	radius, ft; m
T	absolute temperature, $^{\circ}$R; K
U	blade velocity, ft/sec; m/sec
V	absolute gas velocity, ft/sec; m/sec
V_j	ideal jet speed corresponding to total- to static-pressure ratio across turbine, ft/sec; m/sec
W	relative gas velocity, ft/sec; m/sec
w	mass flow rate, lb/sec; kg/sec
α	absolute gas flow angle measured from axial direction, deg
γ	ratio of specific heats
δ	ratio of inlet total pressure to U.S. standard sea-level pressure, p'/p^*
ϵ	function of γ used in relating parameters to those using air inlet conditions at U.S. standard sea-level conditions, $\dfrac{0.740}{\gamma}\left(\dfrac{\gamma+1}{2}\right)^{\gamma/\gamma-1}$
η	turbine efficiency
η_s	static efficiency (based on inlet-total- to exit-static-pressure ratio)
η_t	total efficiency (based on inlet-total- to exit-total pressure ratio)
θ_{cr}	squared ratio of critical velocity at turbine inlet to critical velocity at U.S. standard sea-level temperature, $(V_{cr}/V^*_{cr})^2$
μ	gas viscosity, lb/(ft)(sec); kg/(m)(sec)
ν	blade-jet speed ratio (based on rotor-inlet tip speed), U_t/V_j
τ	torque, in.-lb; N-m

Subscripts:

cr condition corresponding to Mach number of unity

eq equivalent

u tangential component

1 station at turbine inlet (fig. 5)

2 station at stator exit

3 station at rotor exit

Superscripts:

' absolute total state

* U.S. standard sea-level conditions (temperature equal to 518.67° R (288.15 K), pressure equal to 14.70 psia (10.13 N/cm^2)

TURBINE DESCRIPTION

Design Requirements

One of the purposes of this investigation was to compare the performance of the scaled turbine with that obtained for the two reference turbines. This required that the subject turbine be geometrically similar to the reference turbines and have the same mass flow rate and blade speed. All design-point values except inlet total pressure and rotative speed were therefore the same as for the two reference turbines. The turbine-inlet pressure was increased from 22.70 psia (15.65 N/cm^2) (turbine of ref. 2) to 39.04 psia (26.92 N/cm^2) in order to satisfy the design requirement of equal values of mass flow rate. The smaller rotor tip diameter and the design requirement of equal blade speeds resulted in a design rotative speed (corresponding to hot operation) of 66 200 rpm for the subject turbine as compared with 50 500 rpm for the turbine of reference 2.

The design-point values for all three turbines are shown in table I. Rotor inlet tip blade speed was used for the calculation of blade-jet speed ratio. The equivalent pressure ratios were calculated from equivalent specific work in air and the assumption of no change in efficiency when argon or air is used as the working fluid.

Velocity diagrams were calculated to meet the design work requirement and are presented in figure 1. These diagrams are identical to those for the turbines of references 1 and 2. The diagrams indicate a fairly conservative design, with a relatively low level of velocities and very little exit whirl.

TABLE I. - RADIAL-INFLOW TURBINE DESIGN VALUES

	Reference 1 turbine	Reference 2 turbine	Subject turbine
Argon			
Tip diameter, in.; cm	6.02; 15.29	4.59; 11.66	3.50; 8.8
Total to total efficiency, η_t	0.880	0.880	0.880
Total to static efficiency, η_s	0.824	0.824	0.824
Total- to total-pressure ratio, p_1'/p_3'	1.560	1.560	1.560
Total- to static-pressure ratio, p_1'/p_3	1.613	1.613	1.613
Inlet total temperature, T_1', $^{\circ}$R; K	1950; 1083	1950; 1083	1950; 1083
Inlet total pressure, p_1', psia; N/cm^2	13.20; 9.10	22.70; 15.65	39.04; 26.92
Mass flow rate, w, lb/sec; kg/sec	0.611; 0.277	0.611; 0.277	0.611; 0.277
Specific work, Δh, Btu/lb; J/g	34.73; 80.78	34.73; 80.78	34.73; 80.78
Turbine speed, N, rpm	38 500	50 500	66 200
Blade-jet speed ratio, ν	0.697	0.697	0.697
Specific speed, N_s, $\mathrm{ft}^{3/4}/(\mathrm{min})(\mathrm{sec}^{1/2})$; $(\mathrm{rad})(\mathrm{m})^{3/2}(\mathrm{kg})^{3/4}/(\mathrm{sec})^{3/2}(\mathrm{J})^{3/4}$	95.6; 0.74	95.6; 0.74	95.6; 0.74
Reynolds number, Re	63 700	82 200	109 000
Air equivalent (U.S. standard sea-level)			
Equivalent mass flow rate, $\epsilon w\sqrt{\theta_{cr}}/\delta$, lb/sec; kg/sec	1.063; 0.482	0.616; 0.279	0.358; 0.162
Equivalent specific work, $\Delta h/\theta_{cr}$, Btu/lb; J/g	11.90; 27.68	11.90; 27.68	11.90; 27.68
Equivalent speed, $N/\sqrt{\theta_{cr}}$, rpm	22 527	29 550	38 741
Equivalent total- to total-pressure ratio, $(p_1'/p_3')_{eq}$	1.496	1.496	1.496
Equivalent total- to static-pressure ratio, $(p_1'/p_3)_{eq}$	1.540	1.540	1.540
Blade-jet speed ratio, ν	0.697	0.697	0.697
Specific speed, N_s, $\mathrm{ft}^{3/4}/(\mathrm{min})(\mathrm{sec}^{1/2})$; $(\mathrm{rad})(\mathrm{m})^{3/2}(\mathrm{kg})^{3/4}/(\mathrm{sec})^{3/2}(\mathrm{J})^{3/4}$	95.6; 0.74	95.6; 0.74	95.6; 0.74
Equivalent torque, $\epsilon\tau/\delta$, in.-lb; N-m	50.05; 5.65	22.12; 2.50	9.80; 1.11

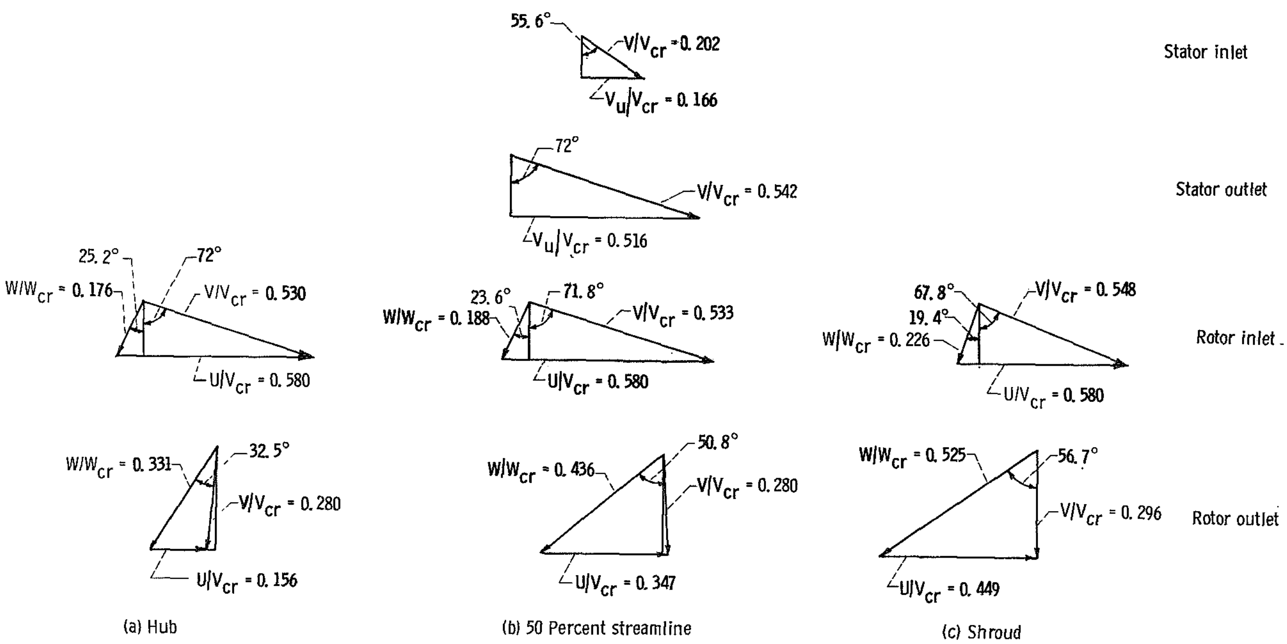

Figure 1. - Design velocity diagrams.

Turbine Geometry

Figure 2 shows the turbine rotor and stator-scroll 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. The figure also shows that the rotor assembly includes 11 blades and 11 splitter blades. These blades extend over approximately one-third the length of the blade near the leading edge and thereby decrease the blade loading in that region.

Although it was intended that the subject turbine be geometrically similar to the reference turbines, some differences existed in the test units. One of these differences was in the shroud clearance values, which have an effect on performance. The rotor axial shroud clearance was 0.008 inch (0.020 cm), and the radial shroud clearance of the exducer section was 0.006 inch (0.015 cm). These values, when expressed as a percent of blade height are 1.8 and 0.7 for the axial and radial clearances, respectively. The turbine of reference 1 had axial and radial clearances of 2.5 and 0.7 percent of the blade height while the turbine of reference 2 had axial and radial clearances of 1.9 and 1.4 percent of blade height. The significance of the difference in clearances will be discussed in the RESULTS AND DISCUSSION section dealing with the performance comparison of the three turbines.

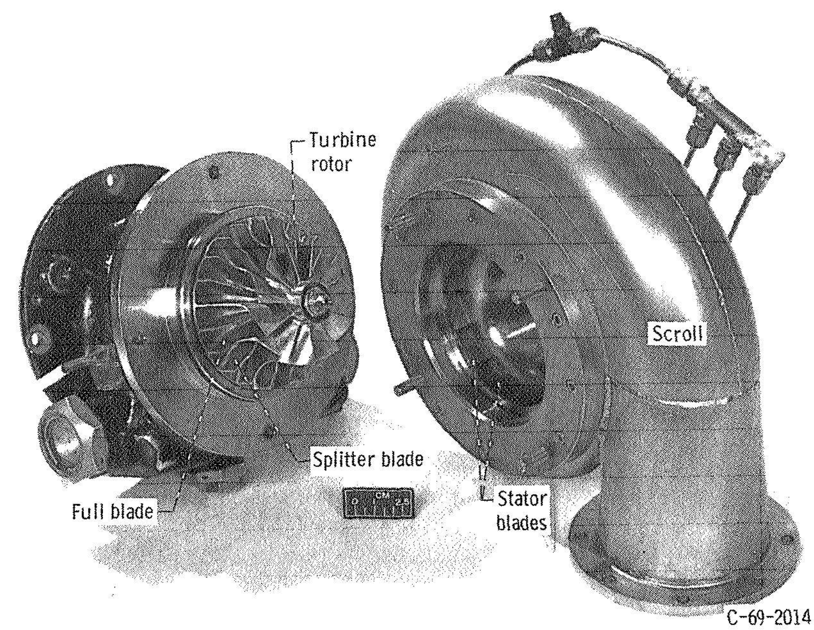

Figure 2. - Rotor and scroll-stator assembly.

APPARATUS, INSTRUMENTATION, AND TEST PROCEDURE

Apparatus

The apparatus used for this investigation consisted of the turbine, an airbrake dynamometer to absorb and measure the net power output of the turbine, and an inlet and exhaust piping system with flow controls. The arrangement of the apparatus is shown schematically in figure 3. Argon was used as the driving fluid for the turbine. The argon was piped to the turbine through an electric heater, a filter, a mass flow measuring station consisting of a calibrated thin-plate orifice, and a remotely controlled pressure regulator. The gas, after passing through the turbine, was exhausted through a system of piping and a remotely operated valve into the laboratory low-pressure exhaust

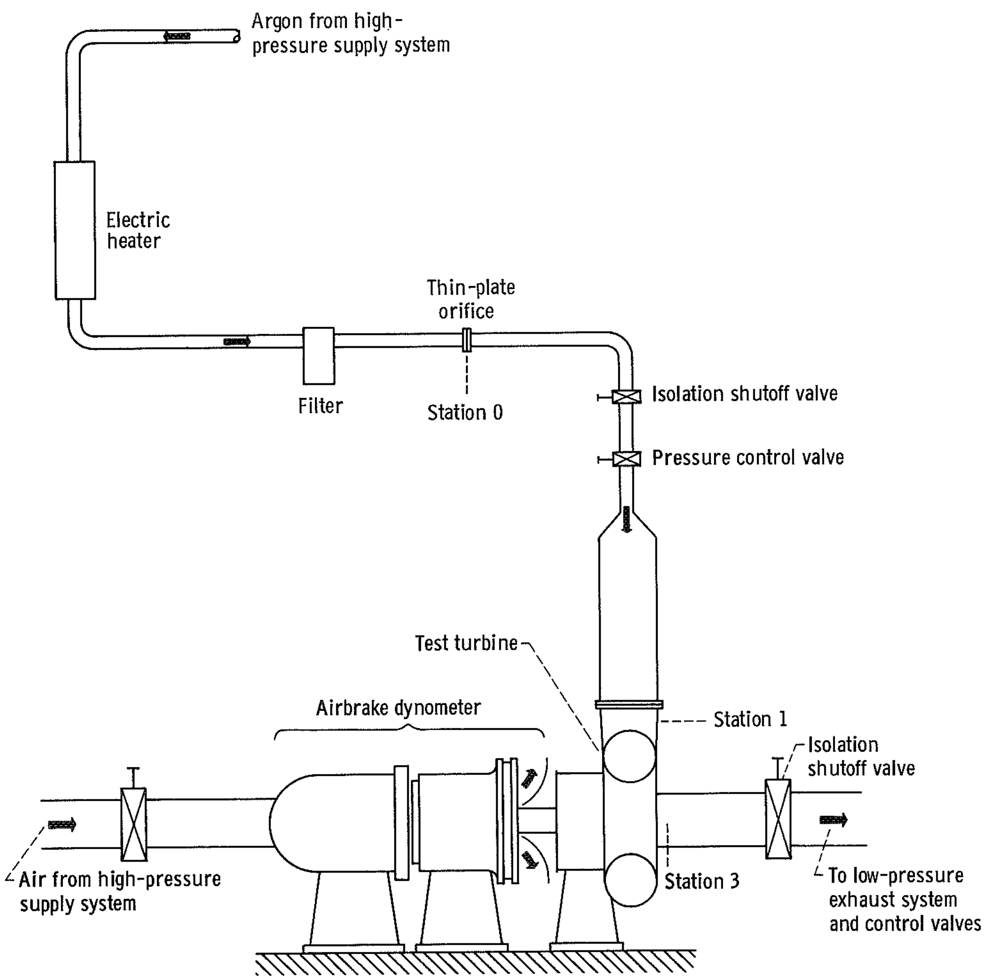

Figure 3. - Experimental equipment.

system. With a fixed inlet pressure, the remotely operated valve in the exhaust line was used to obtain the desired pressure ratio across the turbine.

The airbrake dynamometer, which was cradle-mounted on air bearings for torque measurement, absorbed the power output of the turbine and, at the same time, controlled the speed. The force on the torque arm was measured with a commercial strain-gage load cell. The rotational speed was measured with an electronic counter in conjunction with a magnetic pickup and a shaft-mounted gear. The turbine test facility is shown in figure 4.

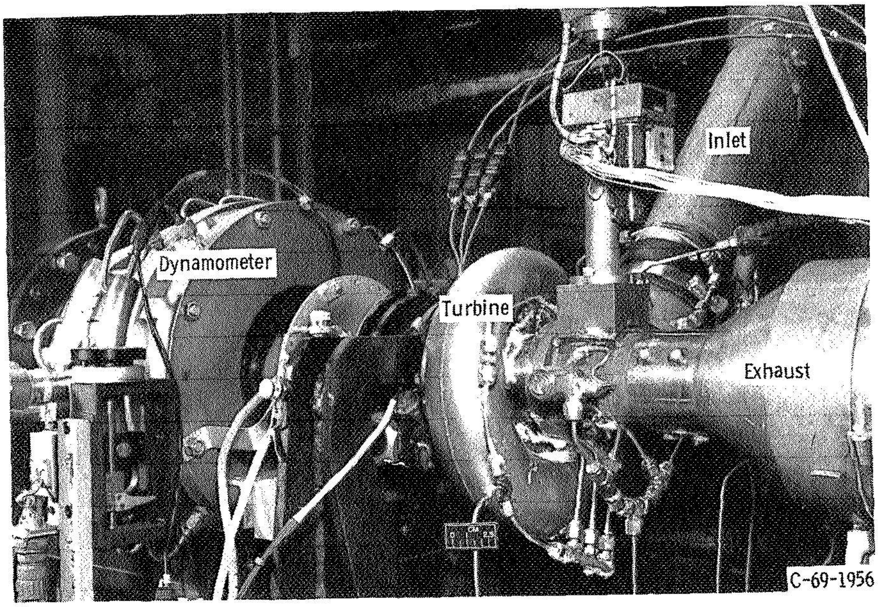

Figure 4. - Experimental turbine test setup.

Instrumentation

The instrumentation stations are shown in figure 5. Turbine performance was determined by measurements taken at stations 1 and 3. The following instrumentation was located at the turbine inlet (station 1): four static-pressure taps and two total-temperature rakes each containing three thermocouples. At station 3, downstream of the rotor exit and in the annular exit section, the instrumentation consisted of six static-pressure taps (three each at the inner and outer walls) and a self-alining probe for flow angle, total-temperature, and total-pressure measurements. In the calculation of pressure ratio across the turbine, the pressure at station 3 was determined from the average of the static pressures at the inner and outer walls. Two interstage pressure taps were in-

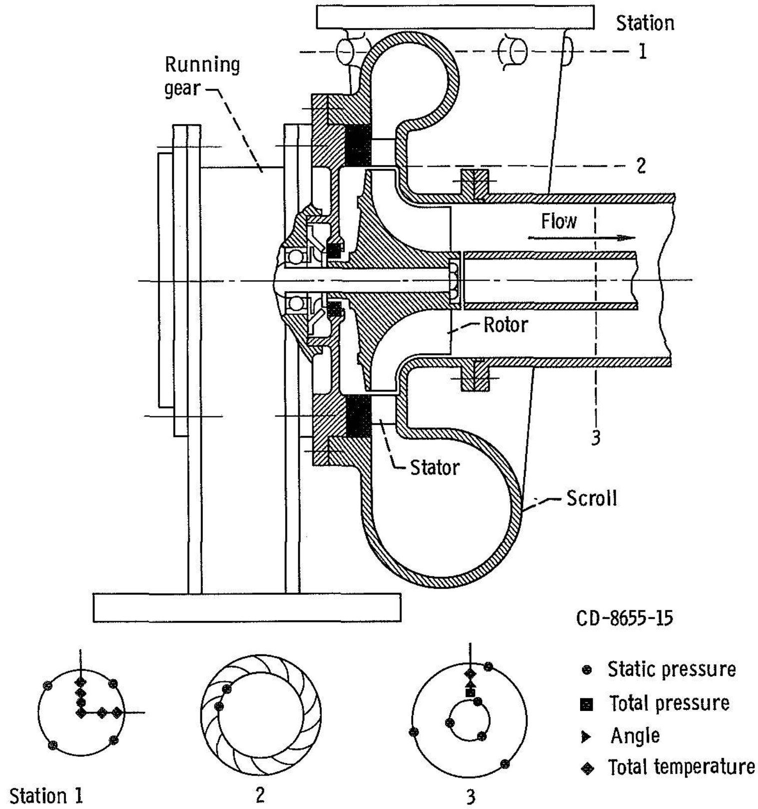

Figure 5. - Turbine test section and instrumentation.

stalled at station 2 (stator exit), thereby making it possible to determine the variation of static pressure through the turbine.

The values of the absolute pressures at the various stations were measured by electrical transducers. Measurements were made on an integrating digital voltmeter and recorded on paper tape for computer processing.

Procedure

Two series of tests were made, with the first made at design Reynolds number. The second series was made over a range of inlet pressures to determine the effect of Reynolds number on turbine performance.

Performance data at design Reynolds number were taken at nominal inlet total conditions of 578° R (321 K) and 9.0 psia (6.2 N/cm^2). These values of temperature and pressure correspond to a Reynolds number of about 109 000 at design operation. Data were obtained over a range of equivalent total- to static-pressure ratios from 1.32 to 2.30 and a speed range from 0 to 110 percent of equivalent design. For the Reynolds number tests, the range of inlet pressures was from 6.0 to 19.0 psia (4.1 to 13.1 N/cm^2). Data were obtained over a range of equivalent pressure ratios of 1.32 to 1.94 at equivalent design speed.

Friction torque of the bearings and seals was obtained by measuring the amount of torque required to rotate the shaft and rotor over the range of speeds covered in the investigation. In measuring the friction torque, windage losses were minimized by evacuating the air from the turbine to a pressure of about 0.001 psia (6.666×10^{-4} N/cm^2). A friction torque value of approximately 0.73 inch-pound (0.08 N-m) was obtained at equivalent design speed. This value corresponds to 12.6 percent of the turbine torque obtained at equivalent design-point operation. Friction torque was added to the dynamometer torque for turbine efficiency calculation.

The turbine was rated on the basis of both total and static efficiency. The total pressures were calculated from mass flow rate, 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 obtained from the investigation of the subject turbine are first presented for operation at design Reynolds number. A discussion of the effect of a change in Reynolds number on the performance of the turbine and a discussion of the efficiency comparison of the subject turbine with those of the other two geometrically similar reference turbines are also presented. All results with the exception of the radial surveys, are shown in terms of air equivalent values.

Performance at Design Reynolds Number

The equivalent specific work output $\Delta h/\theta_{cr}$ is shown in figure 6 as a function of equivalent pressure ratio $(p_1'/p_3)_{eq}$ for lines of constant speed. The curves show trends similar to those of other radial turbines that have been investigated. For the range covered in this investigation, any increase in pressure ratio results in an increase in spe-

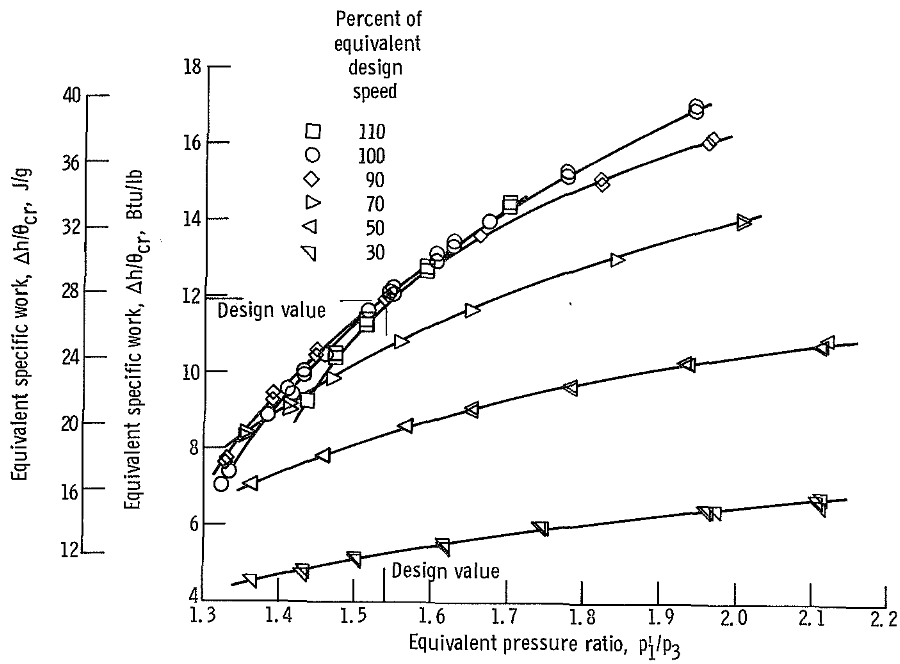

Figure 6. - Variation of specific work with pressure ratio and speed.

cific work at all speeds. This indicates that the turbine has not reached the point of limiting loading. A value of 11.96 Btu per pound (27.84 J/g) was obtained at equivalent design speed and pressure ratio. This value agrees very closely with the design value of 11.90 Btu per pound (27.70 J/g).

Mass flow characteristics are shown in figure 7. Mass flow rate $(\epsilon w\sqrt{\theta_{cr}})/\delta$ is plotted as a function of pressure ratio $(p_1'/p_3)_{eq}$ for each of the seven speeds. The resulting curves are typical of subsonic turbines in that there is an increase in mass flow rate with an increase in pressure ratio at all speeds. At equivalent design speed and pressure ratio, the mass flow rate was 0.341 pound per second (0.155 kg/sec), which is 4.7 percent lower than the design value of 0.358 pound per second (0.162 kg/sec). Dimensions of the stator throats indicated that the stator throat area was about 3 percent lower than the design value. The smaller-than-design rotor throat area as well as accuracy of measurements could account for the remainder of deficiency in mass flow rate.

The torque $\epsilon\tau/\delta$ is shown in figure 8 as a function of speed and pressure ratio. The values of torque were obtained from faired curves because data were taken at constant blade speeds and not at constant pressure ratio. The torque at equivalent design speed and pressure ratio was 9.39 inch-pound (1.06 N-m). This value is about 4.1 percent smaller than the design value of 9.80 inch-pounds (1.11 N-m) and about 55 percent of that obtained at zero speed and design pressure ratio. The curves are typical of those for radial inflow turbines in that they deviate appreciably from straight lines.

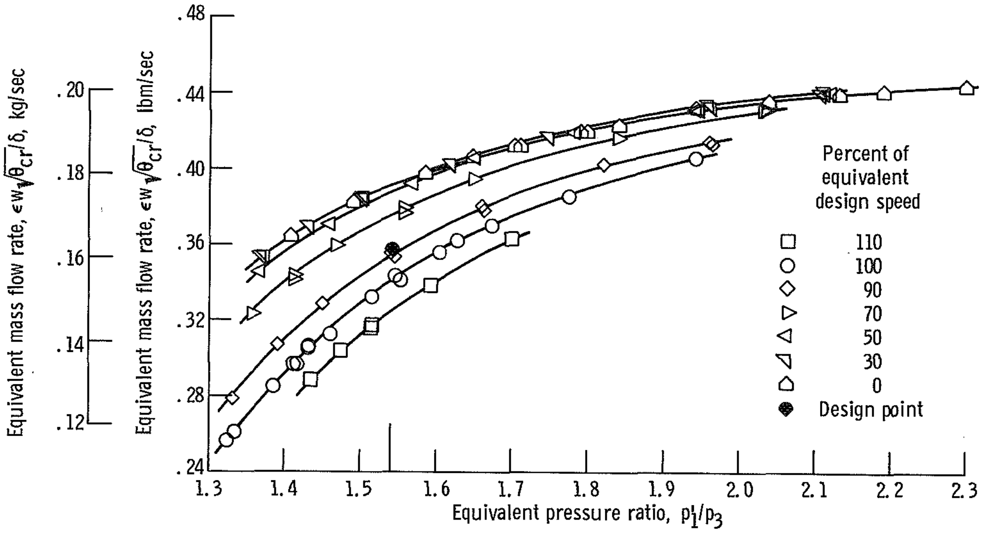

Figure 7. - Variation of mass flow with pressure ratio and speed.

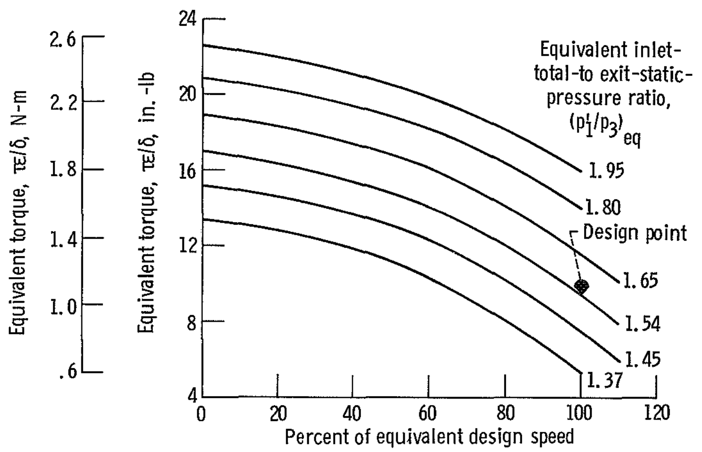

Figure 8. - Variation of torque with speed and pressure ratio.

Figure 9 shows the performance of the turbine in terms of total and static efficiencies. Efficiency is plotted as a function of blade-jet speed ratio for the size values of speed covered in the investigation. Values of total efficiency of 0.88 (fig. 9(a)) and static efficiency of 0.83 (fig. 9(b)) were obtained at equivalent design-point operation. These values agree closely with the design values of 0.880 and 0.824, respectively.

Turbine internal flow characteristics can be determined from the measured static pressures in the turbine together with the results of a radial survey of turbine-exit total pressure and flow angle.

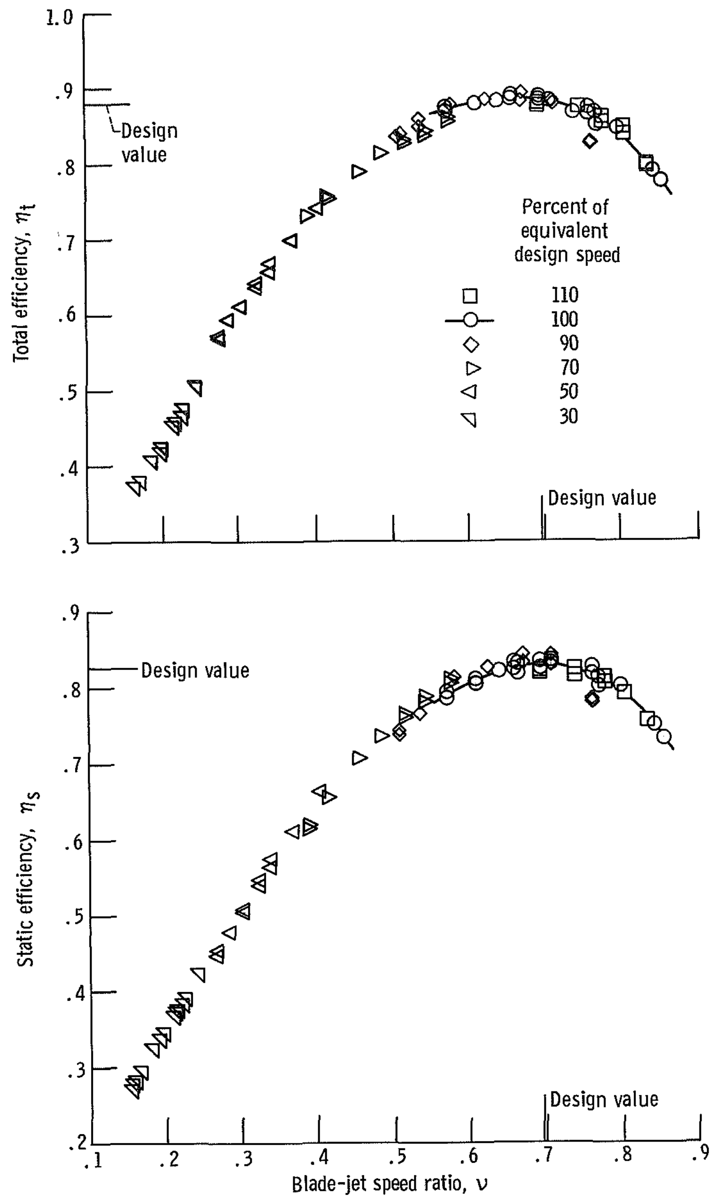

Figure 9. - Variation of efficiency with blade-jet speed ratio.

The experimental value of stator-exit static pressure at equivalent design speed and pressure ratio was very nearly equal to the design value. This indicates that near design free-stream gas velocities were obtained. The measured static pressure also indicated that the rotor was operating near the design value of reaction. Since rotor reaction was near design and the mass flow rate was about 4.7 percent lower than design, this would indicate that the rotor throat area was smaller than design value.

The results of a radial survey of turbine-exit total pressure, total temperature, and flow angle obtained at equivalent design-point conditions are shown in figure 10. The var-

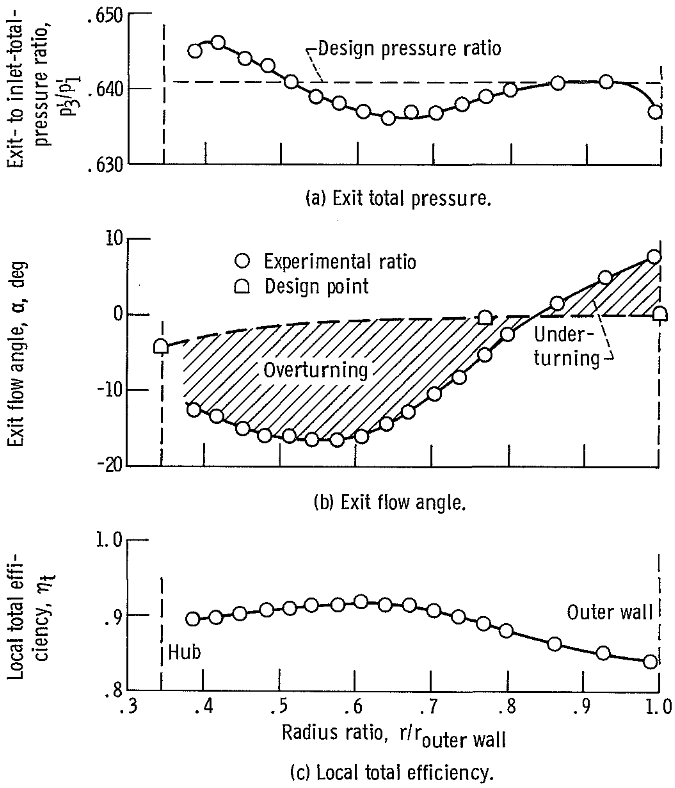

Figure 10. - Variation of exit total pressure, flow angle, and local total efficiency with radius ratio at design-point operation.

iation of exit total pressure with radius ratio is presented in figure 10(a). As can be seen from the figure, there were minor variations in total pressure across the passage with small deviations from the design value. These deviations were less than 1 percent of the design value.

Rotor exit flow angle is shown in figure 10(b) as a function of radius ratio. Positive angles indicate a flow velocity with a component in the direction of rotation. It can be seen that there was overturning of the flow from the hub to a radius ratio of about 0.835. The maximum radial variation in specific work, as a result of the total pressure and flow angle distribution, was calculated to be approximately 8.9 percent of overall specific work as determined by torque, mass flow rate, and speed measurements.

Calculations were then made to determine the values of total efficiency at each of the radial positions where data were obtained. These values of local total efficiency were calculated on the basis of the change in tangential momentum through the rotor and the measured values of total pressure and total temperature at the rotor exit. Figure 10(c) shows values of local total efficiency as a function of radius ratio. There is a maximum variation of approximately 10 points in efficiency, and in the region near the shroud the values of efficiency are several points below the design value. This indicates that design work was not obtained from that part of the rotor blade near the shroud.

Performance Over Range of Reynolds Number

Performance results indicated a small effect of Reynolds number on mass flow rate. An increase in Reynolds number from 68 900 to 219 900 resulted in an increase in equivalent mass flow rate of only 0.8 percent at equivalent design speed and pressure ratio. Static-pressure measurements indicated the stator pressure ratio to be independent of Reynolds number. This would suggest that the small increase in mass flow rate with an increase in Reynolds number was due to decreased viscous losses in the stator and rotor, the ratio between the losses in the stator and rotor remaining the same.

The variations of total and static efficiencies with blade-jet speed ratio for the five values of turbine inlet total pressure are shown in figure 11. The total efficiency shows similar trends for all inlet pressures. The curves indicate a maximum value of efficiency at a value of blade-jet speed ratio that is lower than the design value of 0.697. At the higher values of blade-jet speed ratio, there is a noticeable decrease in efficiency. As shown in reference 3, the large decrease in efficiency at the higher values of blade-jet speed ratio is typical of radial-inflow turbines. The maximum value of total efficiency

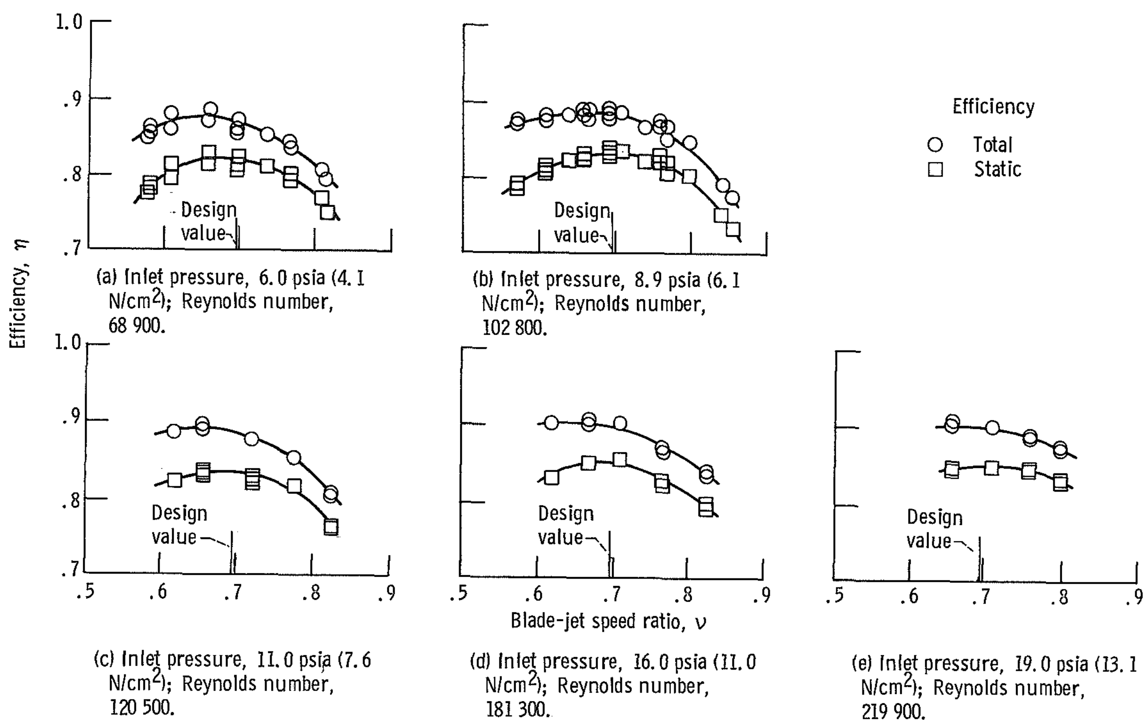

Figure 11. - Performance characteristics over range of turbine inlet pressures.

increased from 0.87 at the lowest Reynolds number to 0.91 for the highest Reynolds number.

The static efficiency curves show trends that are similar to those of the total efficiency curves, with a larger dropoff in efficiency at the lower values of blade-jet speed ratio. The larger difference between total and static efficiencies at the lower values of blade-jet ratio reflects the larger values of exit kinetic energy.

Performance Comparison with the 6.02-Inch (15.29-cm) and 4.59-Inch (11.66-cm) Reference Turbines

For ease of comparison, experimental values obtained for the subject turbine as well as the turbines of references 1 and 2 are listed in table II. Included in the table are the tip clearances for each turbine. It should be remembered that these turbines were designed to be geometrically similar and differ only by the scale factor. Table II shows that the same efficiencies were obtained for both the subject and the 6.02-inch (15.29-cm)

TABLE II. - EXPERIMENTAL VALUES

	Reference 1 turbine	Reference 2 turbine	Subject turbine
Tip diameter, in.; cm	6.02; 15.29	4.59; 11.66	3.50; 8.89
Design Reynolds number	63 700	82 200	109 000
Axial clearance, percent of blade height	2.5	1.9	1.8
Radial clearance, percent of blade height	0.7	1.4	0.7
Total efficiency	0.88	0.86	0.88
Static efficiency	0.83	0.81	0.83

turbine at their respective design Reynolds number and at equivalent design speed and pressure ratio. The table also shows that the radial tip clearances were the same but that the axial tip clearance was larger for the turbine of reference 1. If the results of a tip clearance investigation of reference 4 are applied, the difference in axial tip clearance would only amount to about one-tenth of a point in efficiency.

At equivalent design speed and pressure ratio, the efficiencies of the subject turbine are 2 points higher than those of the 4.59-inch (11.66-cm) turbine of reference 2. Comparison of tip clearances (table II), shows that the reference turbine had a larger radial tip clearance and about the same value of axial tip clearance. The tip clearance investigation of reference 4 indicated that the radial tip clearance had the most significant effect on turbine efficiency. Applying the results of this reference, about one of the two points difference in efficiency can be attributed to the difference in radial tip clearance between the two turbines. The remaining one point difference could result from the inaccuracy of measurements. With regard to the accuracy of data, the inaccuracies involved in the measurements of torque, friction torque, and mass flow rate could result in a ±0.5 point error in the experimentally obtained efficiencies. The main conclusion, therefore, can be made that, within the accuracy of measurements, no significant effect of turbine size on performance was obtained over the range of tip diameters investigated.

The variation of total efficiency with Reynolds number at equivalent design speed and pressure ratio is presented in figure 12. Included in this figure are the results obtained for the turbines of references 1 and 2. All curves show similar trends: an increase in efficiency with an increase in Reynolds number. The figure shows that, for the subject turbine, the total efficiency increased from 0.87 to 0.91 as Reynolds number increased from 68 900 to 219 900.

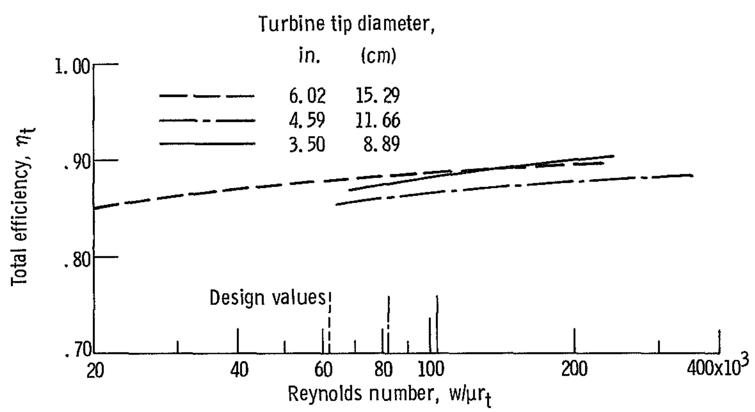

Figure 12. - Comparison of efficiency as function of Reynolds number at equivalent design speed and pressure ratio.

SUMMARY OF RESULTS

An experimental investigation of a 3.50-inch (8.89-cm) tip diameter radial-inflow turbine was conducted to determine its performance at design Reynolds number and over a range of Reynolds number from 68 900 to 219 900. The results were compared with those obtained from geometrically similar turbines with 6.02- and 4.59-inch (15.29- and 11.66-cm) tip diameters which were designed for the same application. The pertinent results of the investigation are as follows:

1. Equivalent specific work of 11.96 Btu per pound (27.84 J/g) was obtained near design Reynolds number (109 000) and at equivalent design speed and inlet-total- to rotor-

exit-static pressure ratio. This value agrees very closely with the design value of 11.90 Btu per pound (27.70 J/g). The associated static and total efficiencies were 0.83 and 0.88, respectively, which compare favorably with design.

2. At this design point, the mass flow rate was 0.341 pound per second (0.155 kg/sec). This value was 4.7 percent lower than design and resulted principally from the flow areas in the blade rows being smaller than design since approximate design reaction was obtained.

3. The value of total efficiency at equivalent design speed and pressure ratio increased from 0.87 to 0.91 as Reynolds number increased from 68 900 to 219 900. The increase in turbine efficiency was attributed entirely to a decrease in viscous losses since rotor reaction appeared to be independent of Reynolds number.

4. There was no significant effect of turbine size on performance for the sizes investigated. The efficiency level of the subject turbine was the same as that of the 6.02-inch (15.29-cm) turbine and 2.0 points higher than that of the 4.59-inch (11.66-cm) turbine. The lower efficiency in the case of the 4.59-inch (11.66-cm) turbine was attributed in part to the excessive radial blade clearance over the exducer. Experimental inaccuracy could account for the remaining difference.

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