NASA TECHNICAL NOTE

NASA TN D-3835

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NASA TN D-3835

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Lewis Research Center
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NATIONAL AERONAUTICS AND SPACE ADMINISTRATION • WASHINGTON, D. C. • FEBRUARY 1967

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SUMMARY

An experimental investigation of a 4.59-inch-tip-diameter radial-inflow turbine was conducted to determine the effect of a change in Reynolds number on the performance of this size and type of turbine. The investigation was conducted with cold argon over a range of inlet pressures from 4.4 to 24.0 pounds per square inch absolute corresponding to a range of Reynolds number from 64 000 to 352 000, at equivalent design speed and pressure ratio. Reynolds number as used herein is defined as weight flow divided by the product of viscosity and rotor tip radius. At each value of turbine inlet pressure, data were taken at equivalent design speed over a range of pressure ratios.

There was a slight increase in equivalent weight flow with an increase in Reynolds number. At equivalent design speed and pressure ratio, the maximum deviation was less than 2.0 percent of the design value of 0.616 pound per second. The total efficiency at equivalent design pressure ratio increased from about 0.85 to 0.88 with an increase in Reynolds number from 64 000 to 352 000. This increase in efficiency was attributed entirely to a decrease in viscous losses since rotor reaction appeared to be independent of Reynolds number. For the same increase in Reynolds number (64 000 to 225 000 at equivalent design speed and pressure ratio), the increase in efficiency was approximately the same for the subject 4.59-inch turbine as for a 6.02-inch similar turbine.

The variation of loss with a change in Reynolds number agrees well with an empirical variation wherein 60 percent of the turbine loss is attributed to viscous losses, and the remaining 40 percent is attributed to other losses, such as tip clearance loss, which are independent of Reynolds number.

INTRODUCTION

NASA is currently studying Brayton cycle space power systems with an output range of a few to a few hundred kilowatts and is experimentally investigating components of systems at the 10-kilowatt power level. The lower power requirement results in turbomachinery of comparatively low weight flow and, thus, relatively low values of Reynolds number. Since the efficiency of the turbine is of utmost importance in a space power system (ref. 1), the variation of turbine performance with Reynolds number should be known.

A number of investigations (refs. 2 and 3) with axial flow turbines have shown a deterioration in performance with decreasing Reynolds number. Little information was available, however, on the effect of a change in Reynolds number on the performance of radial-inflow turbines. Since the design Reynolds number of the turbines under consideration are in the range where performance may deteriorate with a decrease in Reynolds number, a program was initiated at Lewis Research Center wherein a number of radial-inflow turbines of interest are being experimentally investigated over a range of Reynolds number including the design values. Reference 4 presents the results of such an investigation of a 6.02-inch-tip-diameter radial-inflow turbine. The total efficiency of this turbine increased from 0.85 to 0.90 as the Reynolds number changed from 20 000 to 225 000 at equivalent design speed and pressure ratio.

To determine whether there are any size effects associated with the Reynolds number effects, a geometrically similar turbine of 4.59-inch-tip-diameter was investigated over a comparable range of Reynolds number. Tests were made with argon as the working fluid at a nominal inlet temperature of 610° R and at five values of turbine inlet pressure ranging from 4.4 to 24.0 pounds per square inch absolute. At equivalent design speed and pressure ratio, this range of inlet pressures corresponds to a range of Reynolds number from 64 000 to 352 000. (Results of the tests at the design Reynolds number of 82 200 are presented in ref. 5 and are repeated herein.) At each inlet pressure, data were obtained at equivalent design speed over a range of turbine pressure ratios. The results of the investigation in terms of equivalent weight flow, efficiency, and a loss parameter are presented and then compared with those obtained from an investigation of the 6.02-inch turbine.

SYMBOLS

A	flow area, sq in.
g	gravitational constant, 32.174 ft/sec^2
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, $NQ^{1/2}/(H')^{3/4}$, $ft^{3/4}/(min)(sec^{1/2})$

p pressure, psia

Q volume flow (based on exit conditions), cu ft/sec

R gas constant, ft-lb/(lb)($^{\circ}$R)

Re Reynolds number, $w/\mu r_t$

r radius, ft

T absolute temperature, $^{\circ}$R

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, $(2gJ\ \Delta h_{id})^{1/2}$, ft/sec

W relative gas velocity, ft/sec

w weight flow, lb/sec

α absolute rotor exit gas flow angle measured from axial direction, deg

γ ratio of specific heats

δ ratio of inlet total pressure to U.S. standard sea-level pressure, p_1'/p^*

ϵ function of γ used in relating parameters to that using air inlet conditions at U.S. standard sea-level conditions, $\gamma^*/\gamma \left[\left(\frac{\gamma + 1}{2}\right)^{\gamma/(\gamma-1)} \middle/ \left(\frac{\gamma^* + 1}{2}\right)^{\gamma^*/(\gamma^*-1)}\right]$

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

Subscripts:

cr condition corresponding to Mach of 1.0

eq air equivalent (U.S. standard sea level) id ideal

ref reference value (Reynolds number, 225 000)

s shroud

t tip

u tangential component

1 station at turbine inlet

2 station at stator exit

3 station at turbine exit

Superscripts:

' absolute total state

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

TURBINE DESCRIPTION

The turbine used in the subject investigation was the same as that described in reference 5. For convenience, all design-point values with argon as the working fluid and the air-equivalent (U.S. standard sea level) design values are repeated herein (table I) for both the subject and a 6.02-inch geometrically similar turbine (ref. 4), which will hereinafter be referred to as the reference turbine. All design-point values except tip diameter, inlet total pressure, rotative speed, and Reynolds number are the same for both turbines. To obtain equal design power output for the two turbines, an inlet pressure of 22.70 pounds per square inch was required for the subject turbine compared with 13.2 pounds per square inch for the reference turbine. 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 compared with 38 500 rpm for the reference turbine. The smaller tip diameter resulted in a design Reynolds number of 82 200 (corresponding to hot operation) for the subject turbine. This is about 30 percent larger than the design value of 63 700 for the reference turbine.

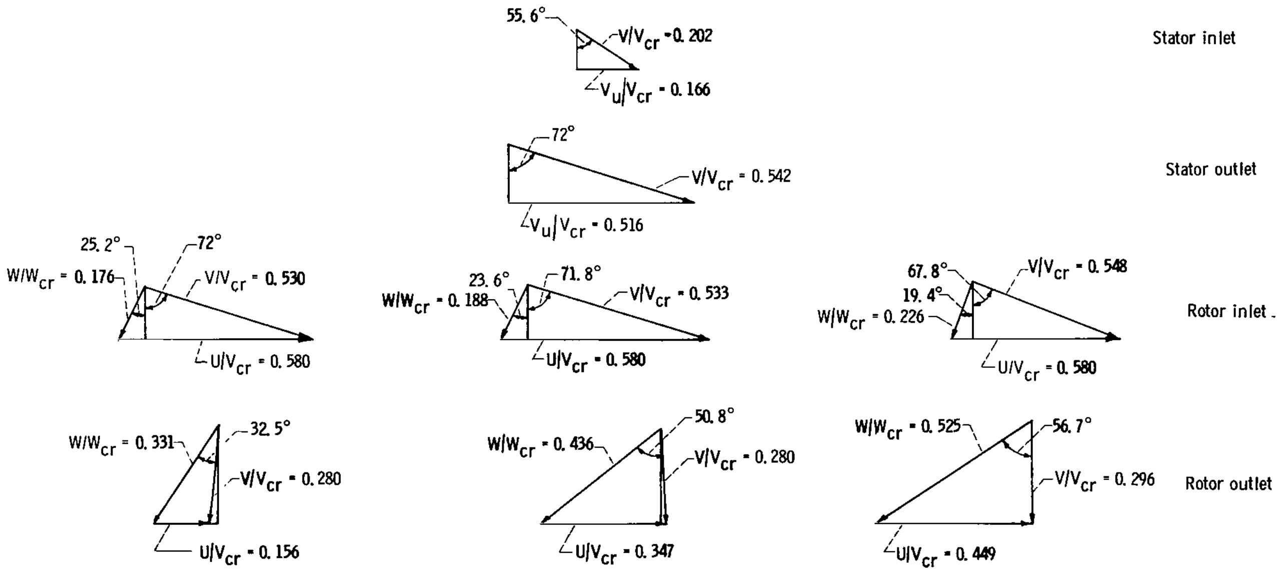
Design velocity diagrams are identical to those for the reference turbine and are presented in figure 1. They indicate a fairly conservative unit that has a relatively low level of velocities and very little exit whirl.

The turbine stator and rotor assembly is shown in figure 2. There are 14 stator blades, one of which has an extended leading edge portion to prevent the flow from entering the small end of the inlet scroll. The rotor has a tip diameter of 4.59 inches and

TABLE I. - RADIAL-INFLOW TURBINE DESIGN VALUES

	Subject turbine	Reference turbine
Argon		
Tip diameter, in.	4.59	6.02
Total to total efficiency, η_t	0.880	0.880
Total to static efficiency, η_s	0.824	0.824
Total- to total-pressure ratio, p_1'/p_3'	1.560	1.560
Total- to static-pressure ratio, p_1'/p_3	1.613	1.613
Inlet total temperature, T_1', $^{\circ}R$	1950	1950
Inlet total pressure, p_1', psia	22.7	13.2
Weight flow, w, lb/sec	0.611	0.611
Specific work, Δh, Btu/lb	34.73	34.73
Turbine speed, N, rpm	50 500	38 500
Blade-jet speed ratio, ν	0.697	0.697
Specific speed, N_s, $ft^{3/4}/(min)(sec^{1/2})$	95.6	95.6
Reynolds number, Re	82 200	63 700
Air equivalent (U.S. standard sea-level)[a]		
Equivalent weight flow, $\epsilon w\sqrt{\theta_{cr}}/\delta$, 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 total- to total-pressure ratio, $p_1'/p_3'{}_{eq}$	1.496	1.496
Equivalent total- to static-pressure ratio, $p_1'/p_3{}_{eq}$	1.540	1.540
Blade-jet speed ratio, ν	0.697	0.697
Specific speed, N_s, $ft^{3/4}/(min)(sec^{1/2})$	95.6	95.6

[a]Correction from design conditions in argon to air-equivalent conditions were made by the method described in ref. 9.



(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.

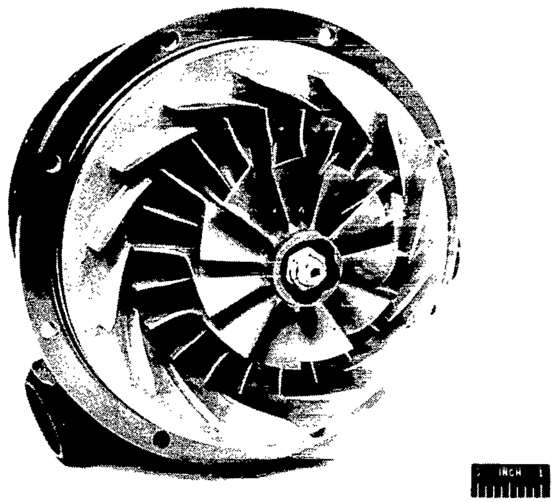


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Figure 2. - Turbine stator and rotor assembly.

includes 11 full blades and 11 splitter blades. The splitter blades extend over approximately one-third the length of the passage near the leading edge, and thereby increase solidity and reduce blade loading in that region.

For purposes of comparison, it was intended that the subject and reference turbines be geometrically similar. However, a slight dissimilarity between the two units was reflected in the shroud clearance values. The subject turbine had axial and radial clearances which were 1.9 and 1.4 percent of the blade height, respectively. For the reference turbine these same clearance values were 2.5 and 0.7 percent of the blade height.

APPARATUS, INSTRUMENTATION, AND TEST PROCEDURE

The apparatus consisted of the turbine, an airbrake dynamometer to absorb and measure the power output of the turbine, and an inlet and exhaust piping system with flow controls. A schematic drawing of the experimental equipment is shown in figure 3.

Pressurized argon was passed through an electric heater, a filter, a weight-flow measuring station, and a pressure control valve before entering the turbine. The argon passed through the turbine and was then piped through an exhaust control valve and into the laboratory low-pressure exhaust system. The turbine test facility is shown in figure 4. Appropriate instrumentation was provided for obtaining the overall performance and the pressure-distribution through the turbine.

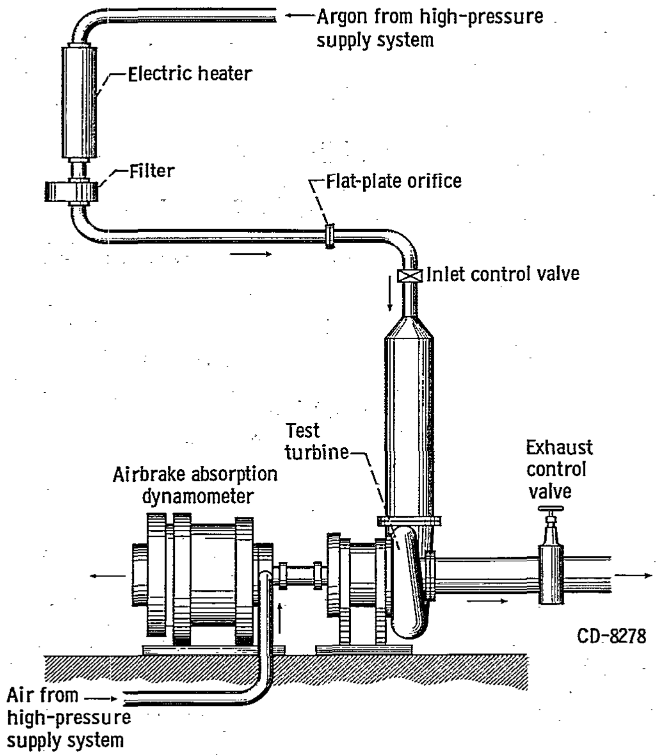


Figure 3. - Experimental equipment.

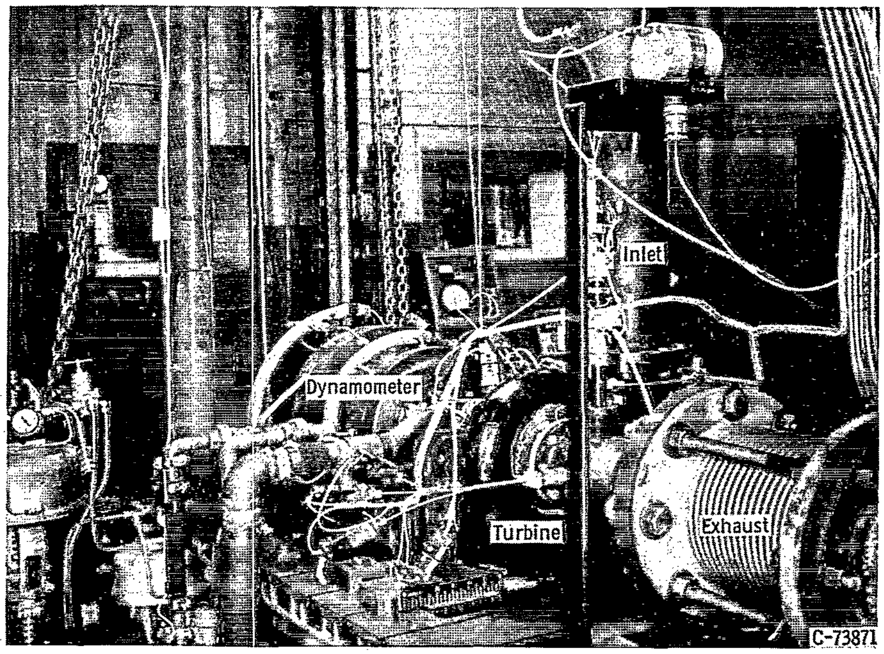


Figure 4. - Experimental turbine test setup.

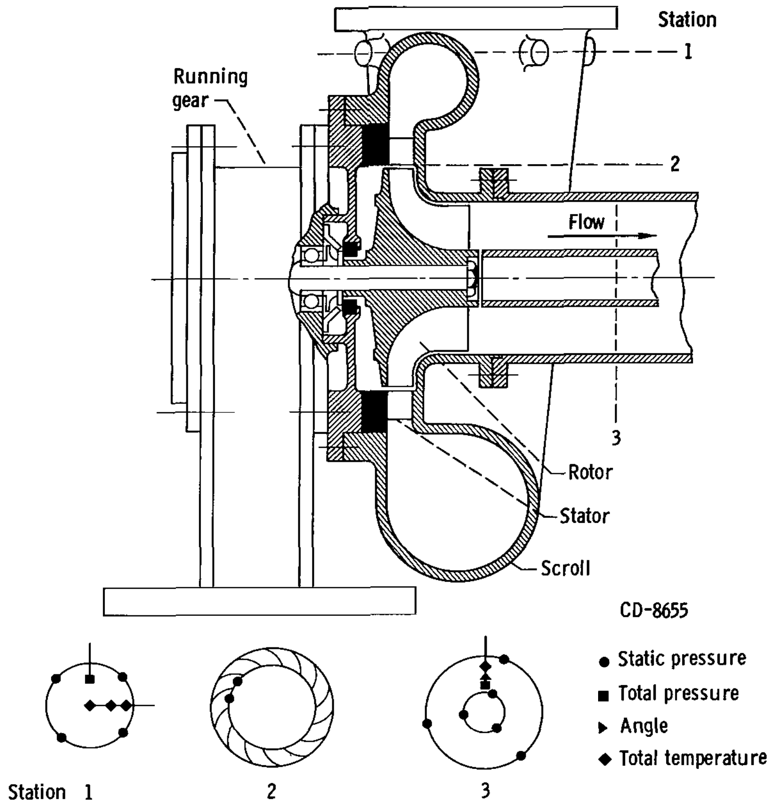


Figure 5. - Turbine test section and instrumentation.

A cross-sectional drawing of the turbine test section showing the instrumentation stations is presented in figure 5. A detailed description of the instrumentation is given in reference 5.

The turbine was operated at a nominal inlet total temperature of 610° R and at five values of inlet total pressure ranging from about 4.4 to 24.0 pounds per square inch absolute. This variation in inlet pressure resulted in a range of Reynolds number from 64 000 to 352 000 at equivalent design speed and pressure ratio. Reynolds number is defined herein as $Re = w/\mu r_t$. At each inlet pressure, data were obtained at equivalent design speed and over a range of total- to static-pressure ratio.

A friction torque of approximately 0.5 inch-pound was measured by the method described in reference 5. This friction torque corresponds to 7.8 and 1.3 percent of the turbine work at turbine inlet absolute pressures of 4.4 and 24.0 pounds per square inch, respectively. The friction torque was added to the shaft torque to obtain the turbine output.

Turbine efficiency was computed as the ratio of actual turbine output to ideal turbine output. The turbine was rated on the basis of both total and static efficiency. Turbine inlet and outlet total pressures were calculated from values of weight flow, static pressure, total temperature, and flow angle by the following equation, which is a rearranged form of equation (6) of reference 6:

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

In the calculation of turbine inlet total pressure the flow was assumed to be normal to the plane defined by station 1.

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

RESULTS AND DISCUSSION

The subject turbine was investigated over a range of inlet pressures from 4.4 to 24.0 pounds per square inch absolute at equivalent design speed and various pressure ratios. For operation at equivalent design pressure ratio, this range of inlet pressures corresponds to a range of Reynolds number from 64 000 to 352 000. Performance results at the design Reynolds number of 82 200 were obtained from reference 5. Results are presented in two sections. In the first section the performance of the subject 4.59-inch turbine is discussed. In the second section the performance of the subject turbine is compared with that of the 6.02-inch reference turbine.

Overall Performance

Figure 6 (p. 10) shows the variation of equivalent weight flow $\epsilon w\sqrt{\theta_{cr}}/\delta$, with equivalent turbine exit-static- to inlet-total-pressure ratio, for operation at equivalent design speed and at five values of turbine inlet pressure. The general trend of weight-flow variation with pressure ratio is typical of subsonic turbines. At any pressure ratio there is a trend of a small increase in equivalent weight flow with an increase in turbine inlet pressure. This variation in weight flow is represented by the band between the two curves in the figure. The equivalent design weight flow of 0.616 pound per second was

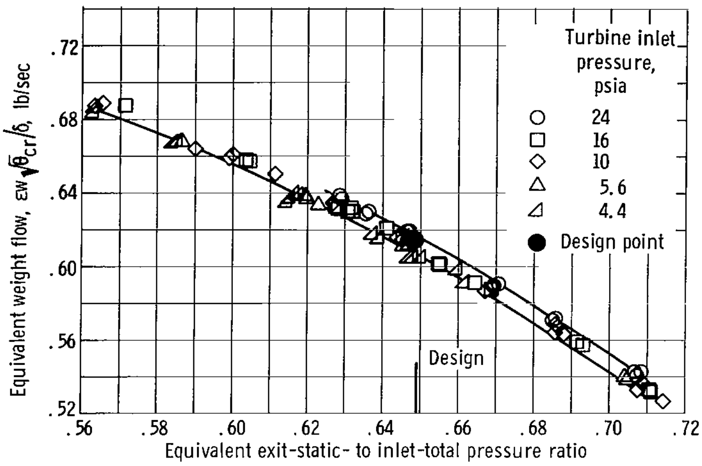


Figure 6. - Variation of equivalent weight flow with pressure ratio and turbine inlet pressure at equivalent design speed; data corrected to equivalent-air conditions.

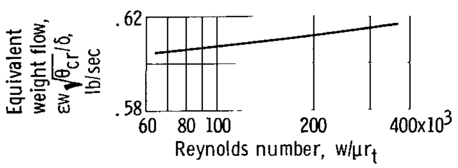
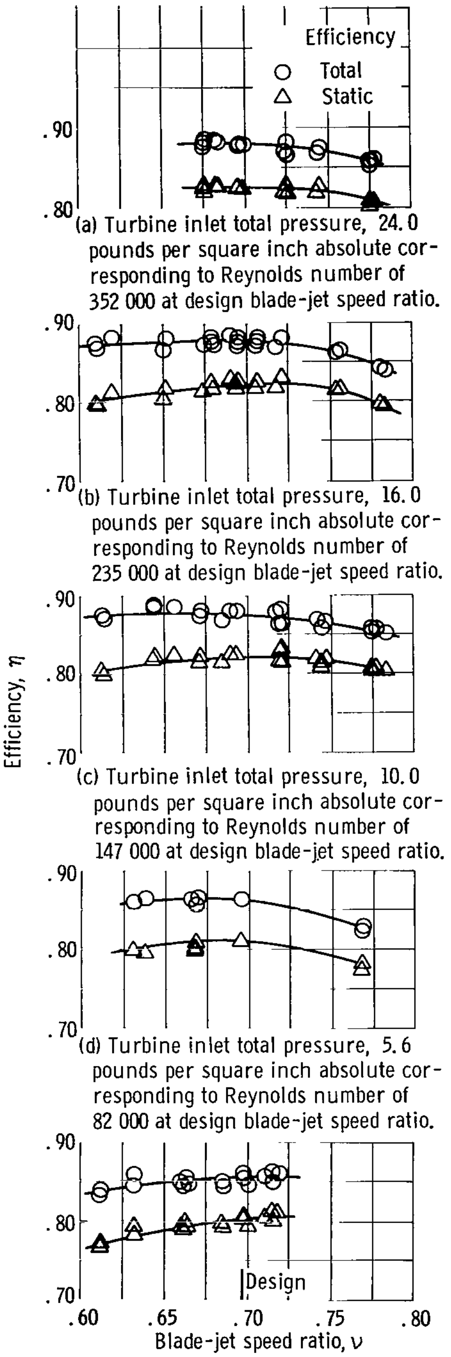


Figure 7. - Variation of equivalent weight flow with Reynolds number at equivalent design speed and pressure ratio.

obtained at an inlet pressure of 24.0 pounds per square inch absolute and at an equivalent design pressure ratio, corresponding to a Reynolds number of 352 000. This weight flow is approximately 1.8 percent larger than that obtained at an inlet pressure of about 5.6 pounds per square inch, corresponding to the design Reynolds number of 82 200. This variation in equivalent weight flow with Reynolds number at equivalent design pressure ratio is shown more clearly in figure 7, which is a cross plot of figure 6 at equivalent design pressure ratio. Static pressure measurements at equivalent design speed and turbine pressure ratio indicated very little change in stator pressure ratio with a change in Reynolds number. Therefore, it is assumed that the pressure distribution through the turbine had a small effect on the change in weight flow. The general trend of decreasing equivalent weight flow with a decrease in Reynolds number is attributed to increased viscous losses in both the stator and the rotor.

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 8. The total efficiency curves show similar trends for all inlet pressures. The curves indicate a maximum value of efficiency at or near the design value of blade-jet speed ratio. At higher values of blade-jet speed ratio, there is a noticeable decrease in efficiency. This trend is not shown by the curve for the lowest inlet pressure (4.4 psia) because limitations of the airbrake dynamometer prevented operation at the higher values of blade-jet speed ratio. As shown in reference 7, the large decrease in efficiency at the higher values of blade-jet speed



(a) Turbine inlet total pressure, 24.0 pounds per square inch absolute corresponding to Reynolds number of 352 000 at design blade-jet speed ratio.

(b) Turbine inlet total pressure, 16.0 pounds per square inch absolute corresponding to Reynolds number of 235 000 at design blade-jet speed ratio.

(c) Turbine inlet total pressure, 10.0 pounds per square inch absolute corresponding to Reynolds number of 147 000 at design blade-jet speed ratio.

(d) Turbine inlet total pressure, 5.6 pounds per square inch absolute corresponding to Reynolds number of 82 000 at design blade-jet speed ratio.

(e) Turbine inlet total pressure, 4.4 pounds per square inch absolute corresponding to Reynolds number of 64 000 at design blade-jet speed ratio.

Figure 8. - Performance characteristics over range of turbine inlet pressures.

ratio is typical of radial-flow turbines. In these turbines, the centrifugal effects cause significant changes in weight-flow distribution at the higher values of rotative speed and thereby cause a rapid change in velocity diagrams. The maximum value of total efficiency increased from 0.85 at the lowest Reynolds number to about 0.88 for the highest Reynolds number.

The static efficiency curves show trends similar to those of the total efficiency curves, with a slightly larger dropoff in efficiency at the lower values of blade-jet speed ratio. This larger difference between total and static efficiencies at the lower values of blade-jet speed ratio reflects the larger values of exit kinetic energy which result from the decrease in density at the higher pressure ratios.

Figure 9 (p. 12) presents total and static efficiencies at equivalent design speed and pressure ratio as functions of Reynolds number. Both curves show the trend of increasing efficiency with increasing Reynolds number. Increasing the Reynolds number from 64 000 to 352 000 caused an increase in total efficiency from about 0.85 to 0.88 and a corresponding increase in static efficiency from about 0.80 to 0.82. The efficiency level of a turbine is known to be affected by the amount of rotor reaction and the level of viscous losses. Stator-exit static-pressure measurements indicated little change in stator pressure ratio with a change in Reynolds number at equivalent design speed and pressure ratio. Near-design rotor reaction, therefore, was obtained under these conditions for the range of Reynolds number investigated. Rotor reaction, consequently, was not a factor in the deterioration of turbine performance with a decrease in Reynolds number. The decrease in efficiency, therefore, is attributed to an increase in viscous losses in both the stator and the rotor.

Performance Comparison with 6.02-Inch Reference Turbine

The subject and reference turbines are compared at their respective equivalent designs speeds of 29 550 and 22 527 rpm and at the equivalent design total- to static-pressure ratio of 1.54, which is common to both turbines. Comparison is made in figure 10 for the range of Reynolds number which was covered by both turbine investigations (64 000 to 225 000). It should be noted that the design Reynolds numbers for the subject and reference turbines are 82 200 and 63 700, respectively. A comparison of these turbines at their respective design values of Reynolds number is presented in reference 5. This reference shows about a 2-point difference in both total and static efficiencies at design-point operation. Shroud clearance and accuracy of data measurement were mentioned as two possible causes of this difference.

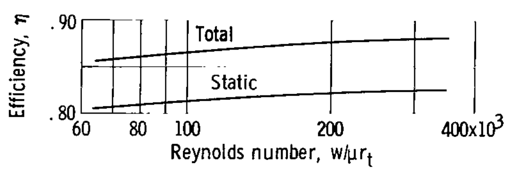


Figure 9. - Effect of Reynolds number on turbine efficiency at equivalent design speed and pressure ratio.

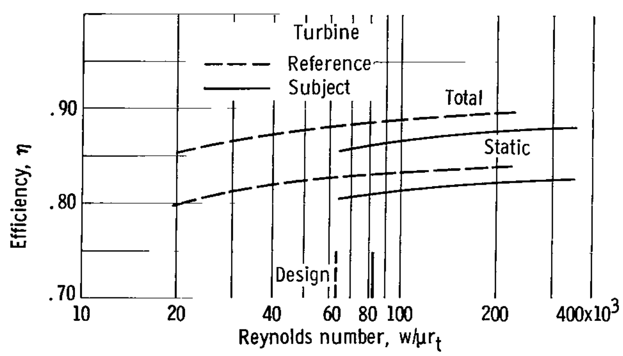


Figure 10. - Comparison of efficiency as function of Reynolds number at equivalent design speed and pressure ratio.

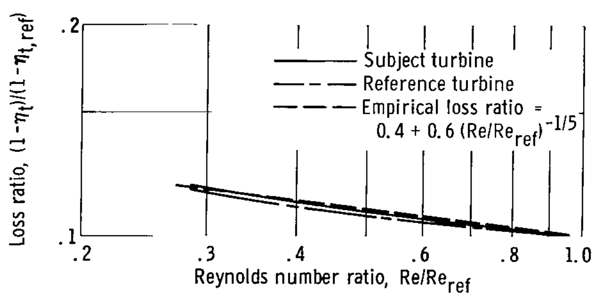


Figure 11. - Comparison of loss ratio as function of Reynolds number at equivalent design speed and pressure ratio.

These values of efficiency are repeated in figure 10 along with those obtained from measurements at other values of Reynolds number. The curves for the reference turbine are reproduced from reference 4; the curves for the subject turbine are taken from figure 9 of this report. Both the total- and static-efficiency curves for the reference turbine are higher than those for the subject turbine at all values of Reynolds number investigated. The curves show the effect of Reynolds number on efficiency to be approximately the same for the two turbines. Limitations of the airbrake dynamometer prevented operation of the subject turbine at the lower values of Reynolds number, which are shown for the reference turbine.

The performance of the turbine at various Reynolds numbers can also be expressed in terms of a loss ratio $(1 - \eta_t)/(1 - \eta_{t,ref})$, where $1 - \eta_{t,ref}$ is calculated at a reference Reynolds number of 225 000. The variation of this loss ratio with the Reynolds number ratio Re/Re_{ref} for both the subject and reference turbines is shown in figure 11. The variation is approximately the same for the two turbines at the range of Reynolds number over which they are compared. Since viscous losses are only one of the factors that contribute to overall turbine loss, the relation between this loss ratio and Reynolds number cannot be expected to follow the one-fifth power law for viscous losses. However, as pointed out in reference 8, this relation can be approximated by a formula of the form

$$\frac{1 - \eta_t}{1 - \eta_{t, ref}} = A + B\left(\frac{R_e}{Re_{ref}}\right)^{-1/5}$$

where the values of the constants A and B depend upon turbine type and design factors such as rotor blade tip clearance, surface diffusion, and loading. Reference 4 shows that experimental data for the reference turbine are in fair agreement with the formula when respective values of 0.3 and 0.7 are assigned to A and B. It was noted in this reference that the agreement was within the experimental error of the test, and since there was only one turbine involved, any attempt to obtain a better agreement by using different values for the constants would be of small merit. However, when data from the reference turbine are supplemented by data from the subject turbine, it is considered valid to select better values for the constants. The resulting values are $A = 0.4$ and $B = 0.6$. An empirical curve using these values is shown in figure 11. Very good agreement is observed for both the subject and reference turbines.

SUMMARY OF RESULTS

An experimental investigation of a 4.59-inch-tip-diameter radial-inflow turbine was conducted. The turbine was investigated over a range of inlet pressures from 4.4 to 24.0 pounds per square inch absolute at equivalent design speed and various pressure ratios. At equivalent design pressure ratio, this range of inlet pressures corresponds to a range of Reynolds number from 64 000 to 352 000. The results were compared with those obtained from a geometrically similar turbine of 6.02-inch tip diameter designed for the same application. The pertinent results of the investigation are given as follows:

1. At equivalent design speed and pressure ratio and for the range of Reynolds number investigated, there was a slight increase in equivalent weight flow with an increase in Reynolds number; the maximum variation was less than 2 percent of the design value of 0.616 pound per second.

2. The value of total efficiency at equivalent design speed and pressure ratio increased from about 0.85 to 0.88 as Reynolds number increased from 64 000 to 352 000.

This increase in turbine efficiency was attributed entirely to a decrease in viscous losses since rotor reaction appeared to be independent of Reynolds number.

3. For the same increase in Reynolds number (64 000 to 225 000), the increase in efficiency of the subject turbine was approximately the same as that of the reference turbine.

4. The variation of loss with a change in Reynolds number agrees very well with an empirical variation wherein 0.6 of the turbine loss is attributed to viscous losses and the remaining 0.4 is attributed to other losses, such as tip-clearance losses, which are independent of Reynolds number.

Lewis Research Center,
National Aeronautics and Space Administration,
Cleveland, Ohio, November 2, 1966,
120-27-03-13-22.

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