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COLD PERFORMANCE EVALUATION OF A 6.02-INCH RADIAL INFLOW TURBINE DESIGNED FOR A 10-KILOWATT SHAFT OUTPUT BRAYTON CYCLE SPACE POWER GENERATION SYSTEM

by Milton G. Kofskey and Donald E. Holeski

Lewis Research Center

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SUMMARY

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A 6.02-inch tip diameter radial inflow turbine was designed and fabricated under contract for a space power application with argon as the working fluid. A cold argon and air performance evaluation was made over a range of pressure ratios and speeds at the NASA Lewis Research Center. The experimental investigation consisted of two phases, tests in argon and air at a high Reynolds number of 225 000 (approximately four times design) and tests in argon at a design Reynolds number of 63 700.

Results of the initial investigation, at the Reynolds number of 225 000, indicated that the equivalent turbine performance was essentially the same using cold argon or air as the working fluid. The total and static efficiencies obtained at equivalent design speed and design blade- to jet-speed ratio (0.697) were 0.90 and 0.84, respectively. These are higher than the design point values of 0.880 and 0.824, respectively. Equivalent weight flow at equivalent design speed and pressure ratio was 1.050 pounds per second, which is 1.2 percent lower than the design value of 1.063 pounds per second.

Final tests were made in argon at the design Reynolds number of 63 700 in order to establish the efficiency level at this design Reynolds number value. Results at design blade- to jet-speed ratio indicated total and static efficiencies of 0.88 and 0.83, respectively. These values are in good agreement with the design point values of 0.880 and 0.824, respectively. Equivalent weight flow at design equivalent speed and pressure ratio was 1.040 pounds per second, which is approximately 2.2 percent lower than the design value of 1.063. This 2.2 percent reduction in equivalent weight flow results from viscous losses being higher than those assumed in the design of the turbine.

Comparison of results obtained at high and design Reynolds numbers, with argon as the working fluid, indicated a 2.0 point decrease in total efficiency and a 1.0 percent reduction in equivalent weight flow at design Reynolds number. Comparison of stator exit static pressure for the two Reynolds numbers investigated indicated that the stator effective area (as affected by losses) was decreasing faster than the rotor effective area when the Reynolds number was reduced to the design value of 63 700.

Author

*The report has been revised to reflect pressure corrections to equivalent weight flow for design Reynolds number operation (fig. 16).

INTRODUCTION

The Lewis Research Center is currently interested in the Brayton cycle for space power systems because it has many features that make it a potentially reliable system. These features include use of an inert gas such as argon as the working fluid and single-phase flow through the system.

Analytical studies (refs. 1 and 2) indicate that the potential of Brayton cycle space power systems is very dependent on turbomachinery efficiency. For example, reference 2 shows the effect of turbomachinery performance on system size and cycle efficiency. This reference shows, that on the average, a 5 point decrease in turbine efficiency results in a 30 to 35 percent increase in radiator area and a corresponding 5 to 10 percent decrease in cycle efficiency. Thus in order to minimize radiator area and obtain high cycle efficiency, the turbine should be designed for maximum efficiency.

In view of this potential turbine performance problem, Lewis initiated a program to determine experimentally the performance levels of turbines for Brayton cycle space power systems. A 10-kilowatt shaft output system using argon as the working fluid was selected as the reference system. System design conditions and assumptions are reported in reference 3.

One type of turbine that is suitable for this 10-kilowatt power level is the radial inflow turbine. Reference 2 indicated that total efficiencies in the range of 90 percent have been obtained for radial inflow turbines using air as the working fluid. However, factors such as specific heat ratio, Reynolds number, and size can have a large effect on turbine performance. Therefore a radial inflow turbine was designed and fabricated under contract by AiResearch Manufacturing Company of the Garrett Corporation in order to determine the level of turbine efficiency for turbines in the power level being considered. Turbine design conditions of pressure, inlet temperature, and weight flow that are consistent with the 10-kilowatt power level system were specified.

An experimental investigation of the radial inflow turbine was made at Lewis to determine turbine performance at two levels of Reynolds number. Tests were made with air and argon at an inlet temperature of 540° R and an inlet pressure of 9.4 pounds per square inch absolute to determine turbine performance at a Reynolds number of 225 000. The tests in air and argon also provided data for performance comparisons between working fluids having different specific heat ratios. Data were obtained over a range of total- to static-pressure ratios of 1.22 to 2.60 and a range of speeds from 40 to 110 percent of design speed.

In addition, tests were made in argon at inlet conditions of 3.40 pounds per square inch absolute and 610° R, which corresponds to the design Reynolds number of 63 700 at hot design conditions. Data were obtained at design equivalent speed and for a range of total- to static-pressure ratios of 1.40 to 1.82.

This report presents design information and test results in terms of specific work, torque, weight flow, and efficiency in equivalent air values.

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, $NQ^{1/2}/60(gH')^{3/4}$
p	pressure, psia
Q	volume flow (based on exit conditions), ft^3/sec
R	universal 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, $\sqrt{2gJ\,\Delta h_{id}}$, 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'/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, $(V_{cr}/V_{cr}^*)^2$

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

1 station at flow orifice (see figs. 6 and 7)

2 station at turbine inlet (see figs. 6 and 7)

3 station at turbine exit (see figs. 6 and 7)

Superscripts:

' absolute total state

* U.S. standard sea level conditions (temperature equal to 518.67° R, pressure equal to 14.69 psia)

TURBINE DESIGN REQUIREMENTS

As mentioned in the INTRODUCTION the 6.02-inch tip diameter radial inflow turbine was designed for a 10-kilowatt shaft output space power system with argon as the working fluid. The design point values for the turbine are as follows:

Total to total efficiency, η_t	0.880
Total to static efficiency, η_s	0.824
Total- to total-pressure ratio, p_2'/p_3'	1.560
Total- to static-pressure ratio, p_2'/p_3	1.613
Turbine speed, N, rpm	38 500
Specific work, Δh, Btu/lb	34.73
Weight flow, w, lb/sec	0.611
Inlet total temperature, T_2', °R	1950
Inlet total pressure, p_2', psia	13.20
Blade- to jet-speed ratio, ν	0.697
Specific speed, $N_s = NQ^{1/2}/60(gH')^{3/4}$	0.118
Reynolds number, $Re = w/\mu r_t$	63 700

The turbine specific speed of 0.118 was selected from the requirement of high turbine efficiency and from consideration of specific speed requirements of the compressor for high efficiency. The turbine design rotative speed of 38 500 rpm is then obtained from the value of specific speed, volume flow based on exit conditions, and ideal specific work based on total conditions.

The following air equivalent (U.S. standard sea level) design values were computed:

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

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. Rotor inlet tip blade speed was used for the calculation of blade- to jet-speed ratio.

Design Velocity Diagrams

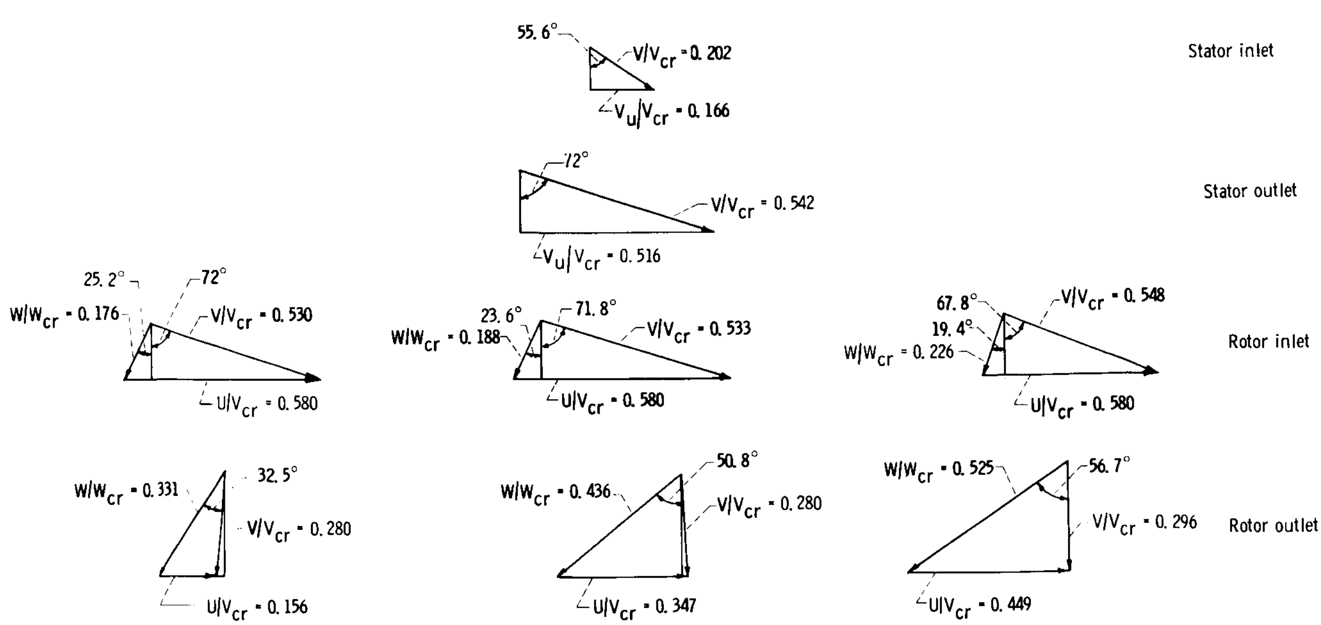
The major turbine rotor dimensions, namely, tip diameter, exducer hub- to tip-diameter ratio, exducer tip diameter and tip width were obtained from calculations involving turbine efficiency and correlations based on the contractor's and other radial turbine data. With these major rotor dimensions and design conditions of pressure, temperature, weight flow, and speed, velocity diagrams were calculated to meet the design work requirement. Figure 1 presents the design velocity diagrams for conditions just inside the blade row. The figure shows the rotor relative inlet angles that result in minimum rotor incidence losses. The diagrams show that the flow was subsonic throughout the turbine. Turning through the stator was 16.4°, and turning through the rotor at the 50 percent streamline was 27.2°. The figure also shows that the turbine was designed with approximately zero exit whirl.

TURBINE DESCRIPTION

Stator

In the design of the stator, the blade shape and the number of blades were optimized for minimum friction and mixing losses by use of boundary layer techniques.

The design resulted in 14 stator blades equally spaced of which 13 are identical to each other and one blade is similar except that the leading edge portion is extended. This one elongated blade blocks the flow from entering the small area end of the inlet scroll.



(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.

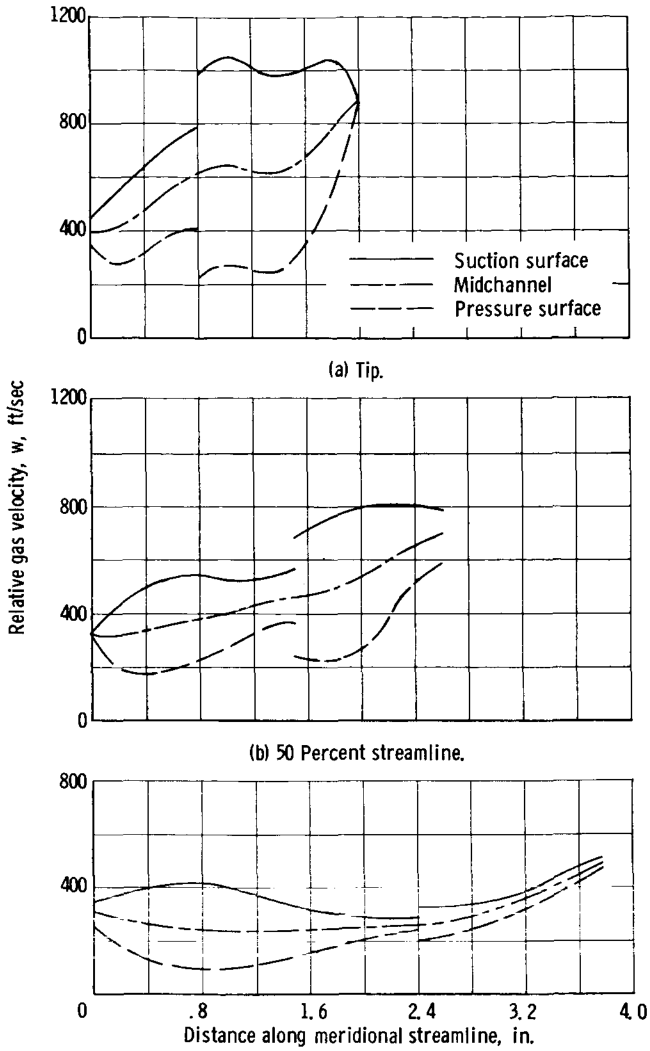


Figure 2. - Design rotor blade midchannel and blade surface velocity distributions.

Rotor

In the design of the rotor, the blade shape and the number of blades were optimized for low losses. Several combinations of blade angle distributions and number of blades were used until the blade loading diagrams were free of excessive velocity decelerations and low or negative velocities. Splitter vanes are used over the initial third of the rotor. The resultant decrease in loading is required at the hub to prevent low blade velocities. The final design resulted in the rotor having 11 blades and 11 splitter vanes.

Figure 2 shows the design velocity distributions for the rotor. The figure shows that the exducer section was heavily loaded, particularly at the tip. The discontinuity in velocities at the exit of the rotor section containing the splitter vanes results from the design method not being able to adjust for flow patterns in this region. In actual operation, however, the sharp discontinuity in velocities does not exist.

A photograph of the turbine rotor and scroll-stator assembly is shown in figure 3. Figure 4 presents a sketch of the turbine giving the major dimensions and a sketch of the stator and rotor blades.

APPARATUS, INSTRUMENTATION, AND PROCEDURE

The apparatus used in the evaluation of the performance of this turbine consisted of a suitable inlet and exhaust system that included flow controls, appropriate instrumentation, and an airbrake, which is similar to the airbrake described in reference 4. Figure 5 shows the experimental turbine installation.

The arrangement of the apparatus is shown schematically in figure 6. Either high-pressure dry air from the laboratory air system or high-pressure argon from a trailer supply can be selected as the driving fluid for the turbine by use of the proper isolation shutoff valves. The fluid was heated by an electric heater and was filtered. The fluid was then passed through a weight flow

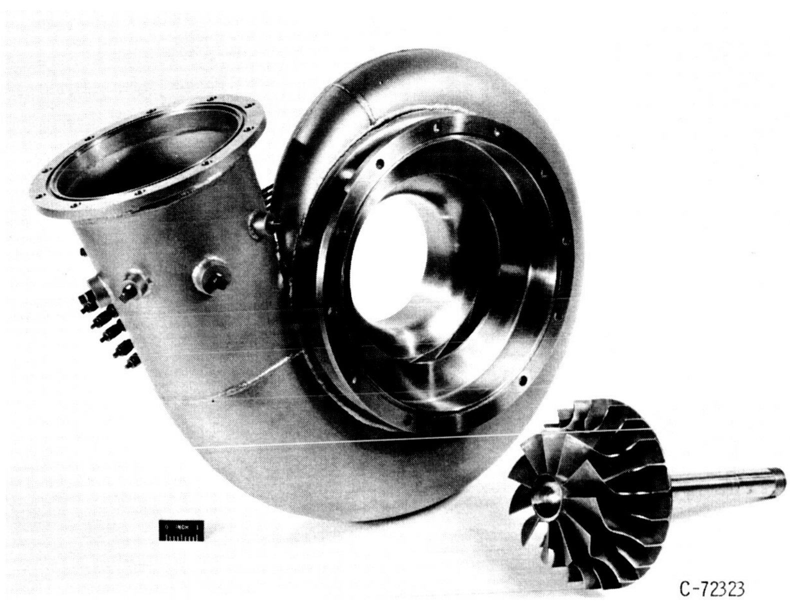


Figure 3. - Turbine rotor and scroll-stator assembly.

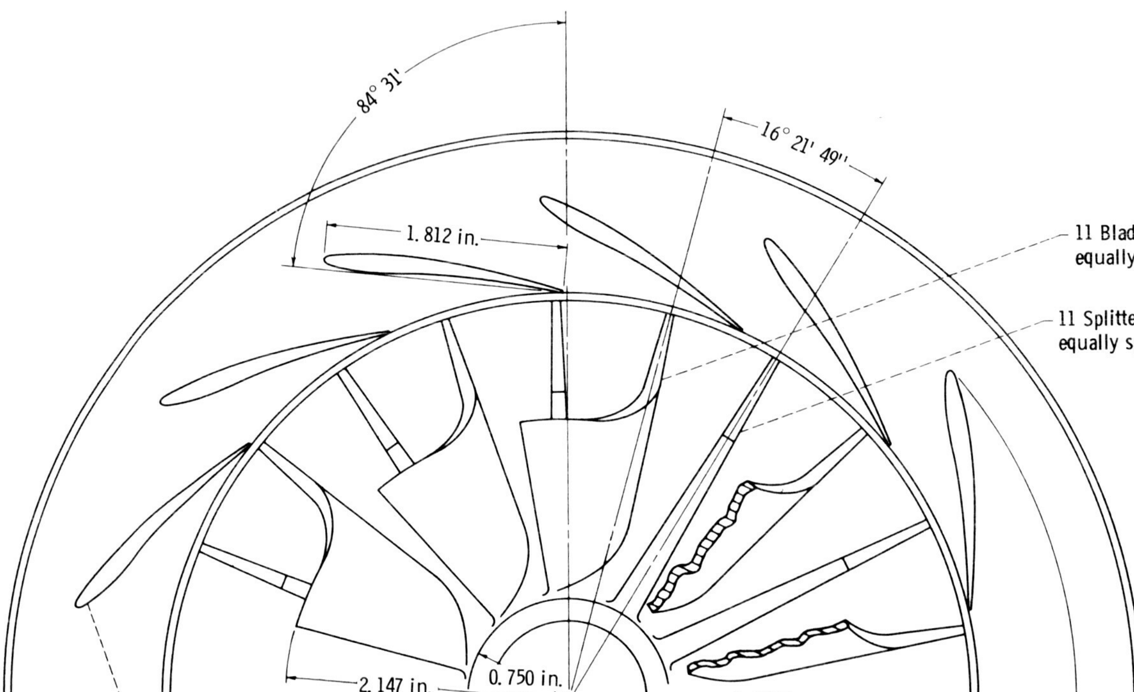


Figure 4. - Turbine stator and rotor.

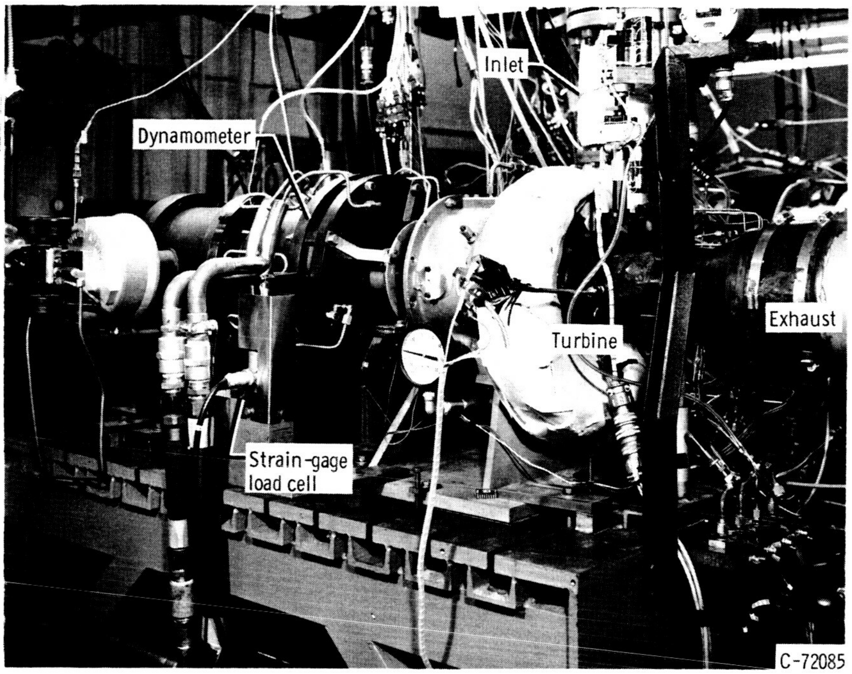


Figure 5. - Experimental turbine and apparatus.

measuring station that consisted of a calibrated flat-plate orifice. A pressure control valve upstream of the turbine regulated the turbine inlet pressure. The fluid after passing through the turbine was exhausted into the laboratory low-pressure exhaust system. With a fixed inlet pressure, a remotely controlled valve in the low-pressure exhaust line was used to maintain the desired pressure ratio across the turbine.

The rotor axial tip clearance was 0.019 inch, and the radial tip clearance of the exducer section was 0.010 inch.

The power output of the turbine was absorbed and the speed controlled by an airbrake dynamometer that was cradle mounted on air bearings for torque measurement. The torque-force measurement was made with a calibrated commercial strain-gage load cell.

The rotational speed of the turbine was measured with an electronic counter in conjunction with a magnetic pickup and a shaft-mounted gear. Figure 7 shows the instrument measuring stations. Station 2, turbine inlet, contained the following instrumentation: four static pressure taps, one total pressure probe, and one total temperature rake containing three thermocouples. The total pressure probe and one thermocouple were used only for setting and monitoring the turbine inlet conditions. Station 3, turbine exit, contained the following instrumentation: three static pressure taps each at the inner and outer walls, a two-tube probe for flow angle measurement, and a total temperature rake containing three thermocouples. One of the three thermocouples was used for monitoring the turbine exit temperature.

All data were recorded by an automatic digital potentiometer and were

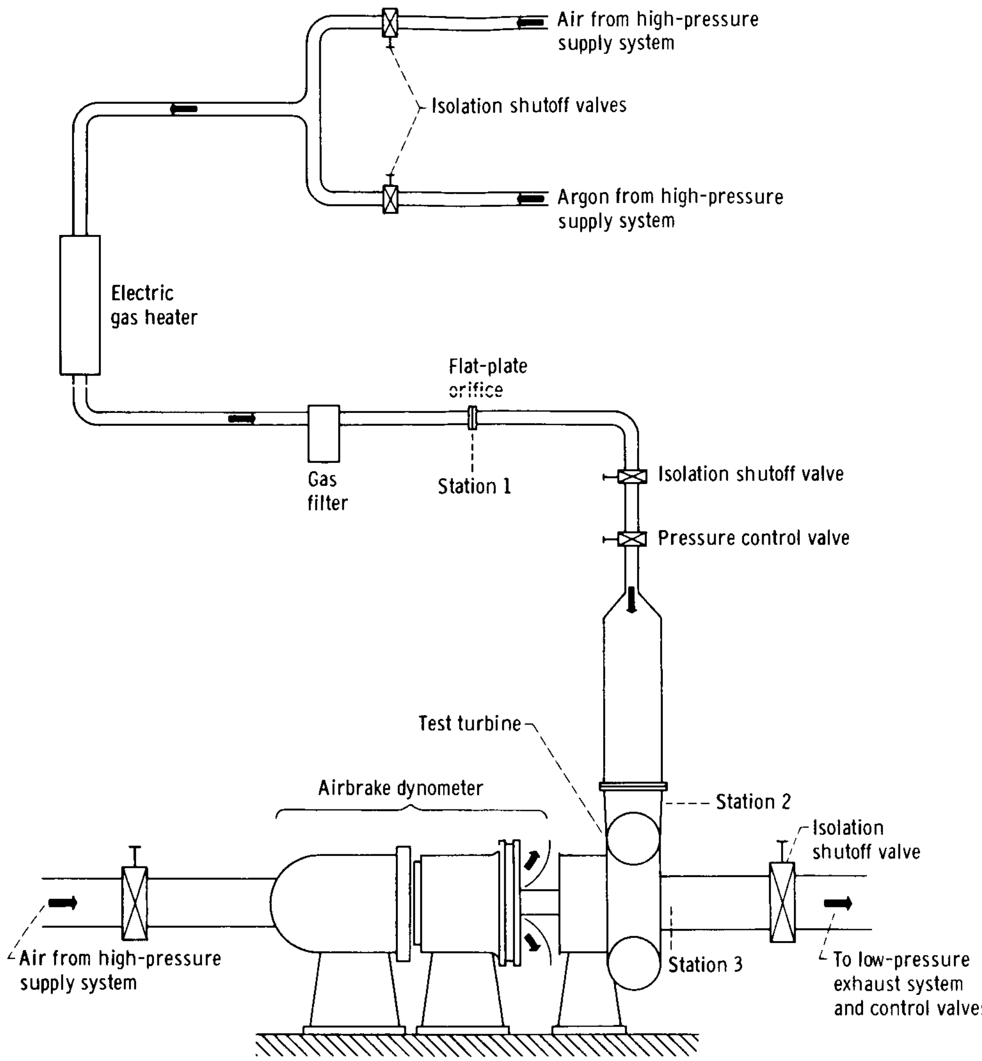


Figure 6. - Experimental equipment.

processed through an electronic digital computer. Experimental data, for the Reynolds number of 225 000 were taken over a range of inlet-total- to exit-static-pressure ratios from approximately 1.22 to 2.60. At each pressure ratio the turbine rotative speed was varied from 40 to 110 percent of design equivalent speed in 10 percent increments. The turbine total temperature was approximately 540° R and the inlet total pressure was approximately 9.4 pounds per square inch absolute. Friction torque of the bearings and seals was obtained by motoring the shaft and rotor over the range of speeds covered in this investigation. The turbine was evacuated to minimize windage losses on the rotor, and care was taken to simulate the operating bearing temperatures for this study. For the high Reynolds number investigation, a friction torque value of approximately 2.8 inch-pounds was obtained for design equivalent speed. Since friction torque values of 2.8 inch-pounds would represent approximately 22 percent of turbine torque obtained at design equivalent speed and pressure ratio for the design Reynolds number investigation, modifications were made to the seals and lubrication system. The modification consisted of removing the carbon face seal

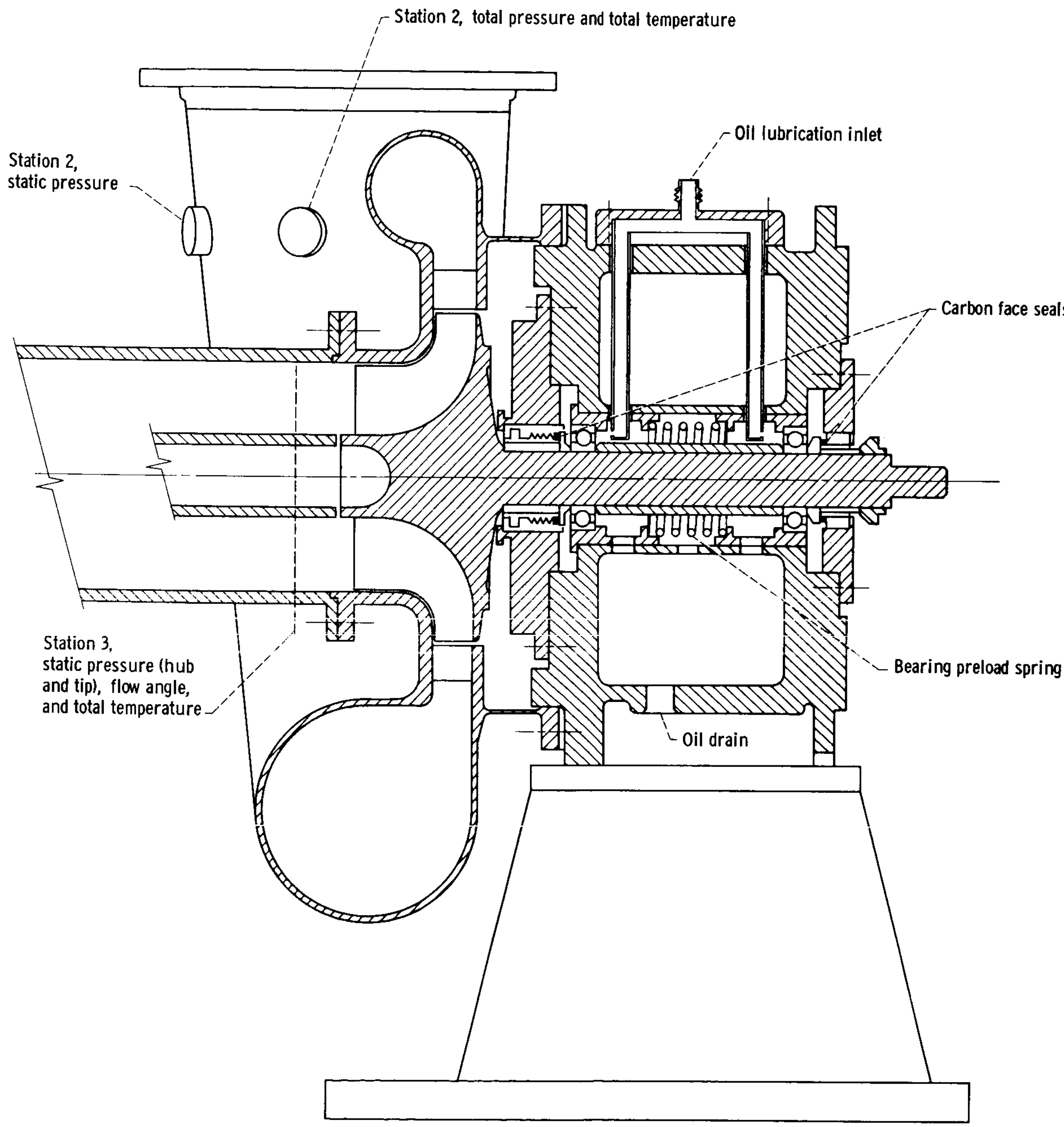


Figure 7. - Turbine test section.

on the side opposite the rotor (see fig. 7) and replacing this seal with a labyrinth seal. An oil mist lubrication was used in place of the oil jet system to minimize excess oil to the bearing. Friction torque was reduced from the 2.8 to approximately 0.70 inch-pound. This amounts to approximately 5.6 percent of turbine torque at design equivalent speed and pressure ratio for the design Reynolds number investigation.

The turbine was rated on the basis of total efficiency 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.

The performance of the turbine at the design Reynolds number of 63 700 (corresponding to hot design operation) was obtained in the same manner, except that the inlet total pressure was approximately 3.4 pounds per square inch absolute and the inlet total temperature was approximately 610° R. The turbine was operated with argon as the driving fluid and over a range of total- to static-pressure ratios of 1.40 to 1.82 at design rotative speed.

RESULTS AND DISCUSSION

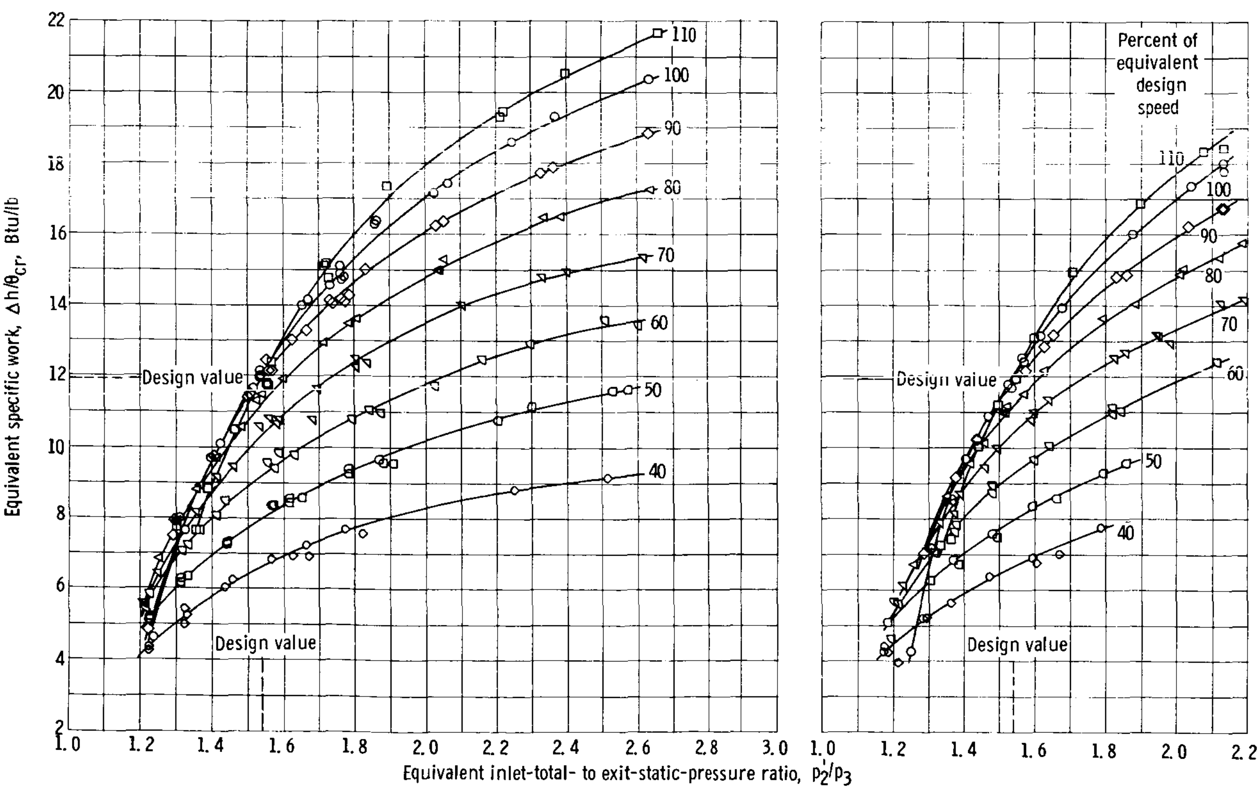
The results of this investigation are presented in two sections. The first section, Performance at Reynolds Number of 225 000, presents the results of the turbine tests with air as well as argon as the working fluid. This section also presents a comparison of the turbine performance with the two different working fluids. The second section, Performance at Design Reynolds Number of 63 700, presents the performance of the turbine at the design Reynolds number (corresponding to design hot operation) and includes a comparison with results obtained from the first section.

Performance at Reynolds Number of 225 000

Performance results are presented in figures 8 to 12 for the tests made in air and argon. Part (a) of the figures shows the results obtained in air, and part (b) shows the results obtained in argon. The argon data has been corrected to air equivalent data to facilitate direct comparison between results for the two fluids. Since the results obtained in argon were essentially the same as obtained in air, discussion of the figures will essentially be limited to the data obtained with air as the working fluid.

Figure 8 presents the variation of turbine equivalent specific work output $\Delta h/\theta_{cr}$ with equivalent total- to static-pressure ratio p_2'/p_3 for lines of constant speed. The turbine equivalent specific work output obtained at design equivalent total- to static-pressure ratio was 12.1 Btu per pound, which is 1.6 percent greater than the design value of 11.9. It may be noted that at the equivalent design pressure ratio (1.54), turbine work varied by less than 2 percent as the speed was varied from 90 to 110 percent. This indicates that, at this pressure ratio, the turbine is operating in the region where peak efficiency occurs. The figure also shows that limiting loading was not reached over the pressure ratio range investigated.

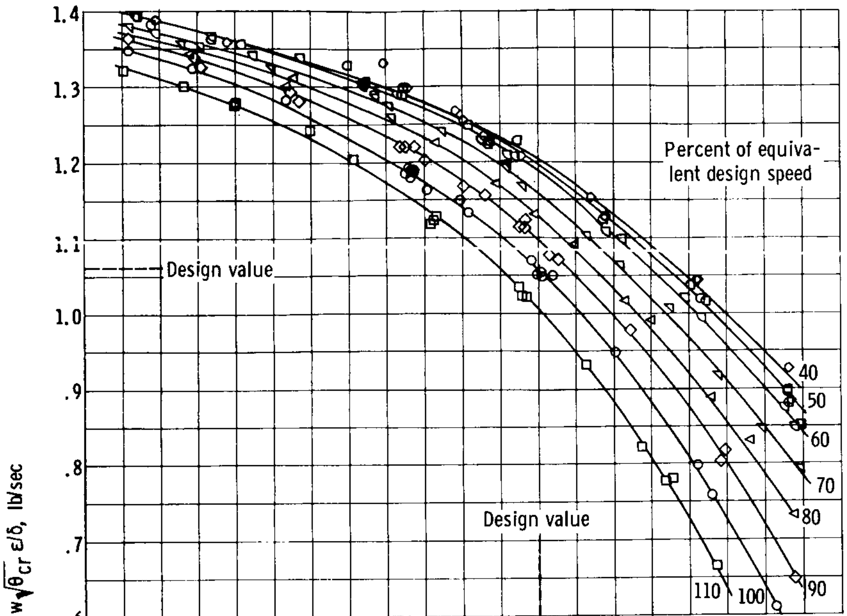
Figure 9 presents the variation of turbine equivalent weight flow $w\sqrt{\theta_{cr}}\,\epsilon/\delta$ with exit-static- to inlet-total-pressure ratio p_3/p_2'. The reciprocal of pressure ratio of figure 8 was used to spread the weight flow-speed curves further apart for comparison purposes. The figure shows that at design static- to total-pressure ratio (0.649), the equivalent weight flow was 1.060 pounds per second, which is within a quarter of 1 percent of the design value of 1.063. One of the characteristics of radial inflow turbines is that, at exit- to inlet-pressure ratios near 1.0, weight flow decreases more rapidly with speed than it does for axial flow turbines because of the pressure gradient produced by centrifugal effects in the rotor.



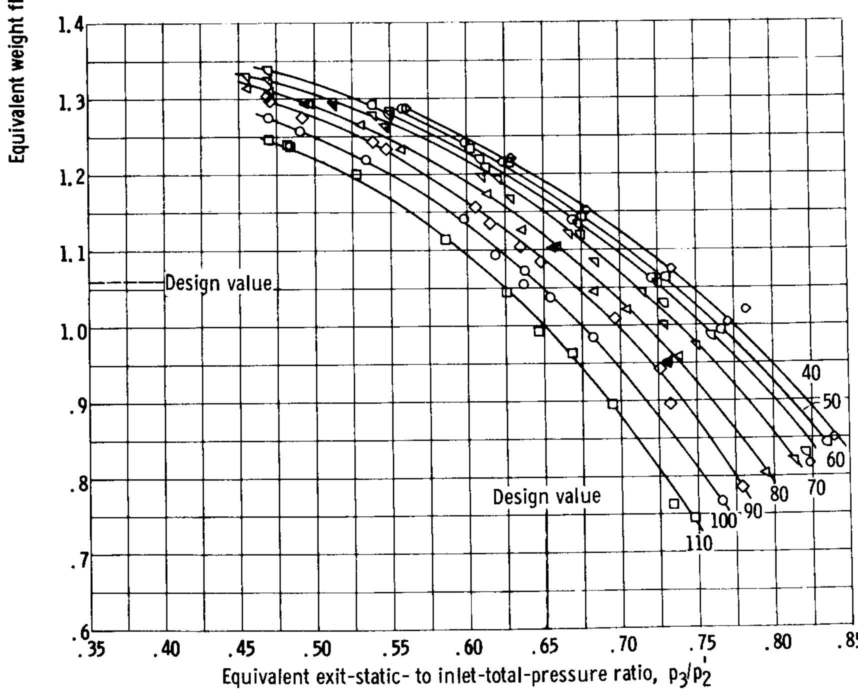
(a) Using cold air as driving fluid.

(b) Using cold argon as driving fluid. Data corrected to equivalent air conditions.

Figure 8. - Variation of specific work with pressure ratio and speed.

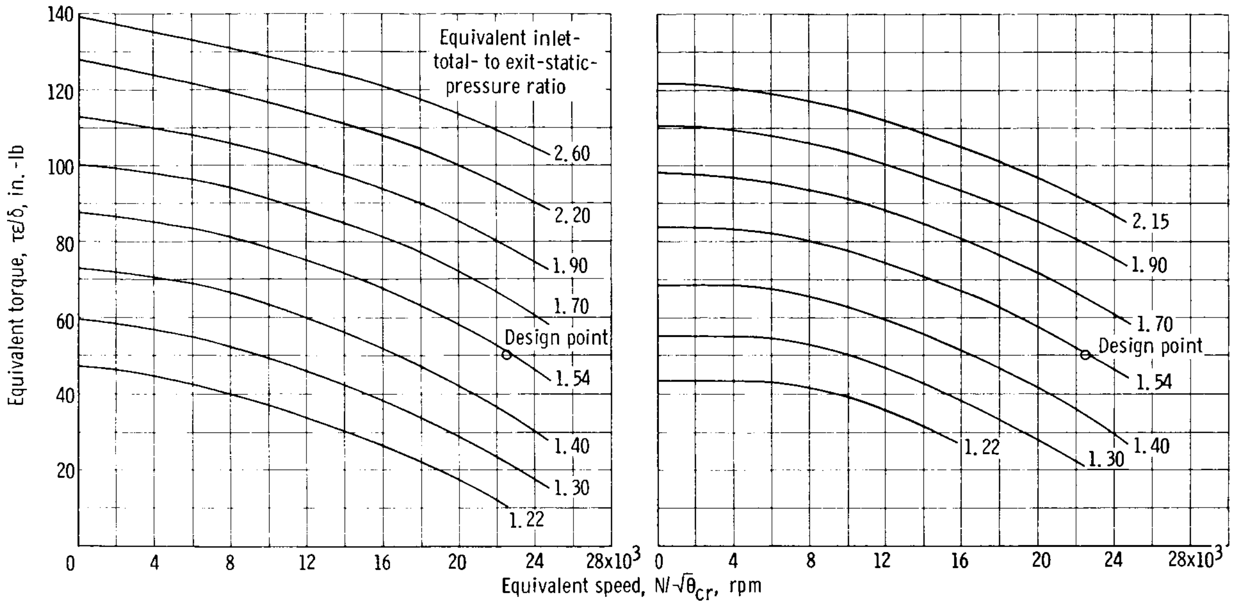


(a) Using cold air as driving fluid.



(b) Using argon as driving fluid. Data corrected to equivalent air conditions.

Figure 9. - Variation of equivalent weight flow with pressure ratio for lines of constant speed.



(a) Using cold air as driving fluid.

(b) Using cold argon as driving fluid. Data corrected to equivalent air conditions.

Figure 10. - Variation of equivalent torque with equivalent speed and pressure ratio.

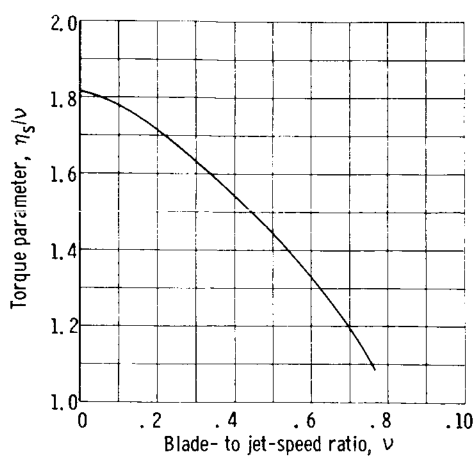
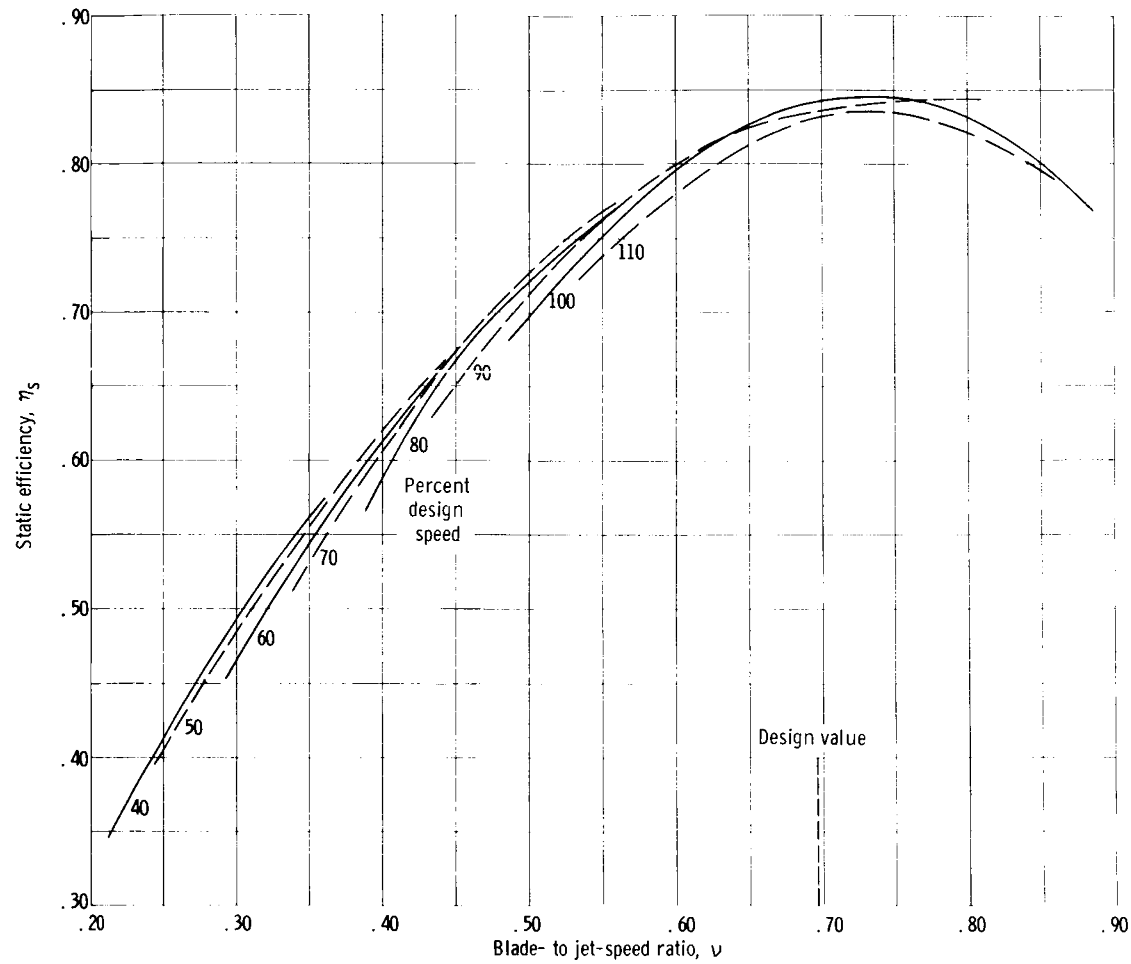


Figure 11. - Torque-speed characteristics using cold air as driving fluid for design pressure ratio.

Figure 10 presents the variation of turbine equivalent torque $\tau\epsilon/\delta$ with equivalent speed $N/\sqrt{\theta_{cr}}$ for lines of constant pressure ratio. Since these results are obtained from faired data, symbols are omitted. This was done because the data were taken at constant blade speed values and not at constant pressure ratio values. Figure 10(a) shows that the equivalent torque at design speed and pressure ratio was 50.8 inch-pounds, which is approximately 1.5 percent greater than the design value of 50.05 inch-pounds. The increased equivalent torque value coupled with a slight decrease in equivalent weight flow resulted in the 1.6 percent increase in turbine equivalent specific work as mentioned previously. The figure shows that zero speed torque is approximately 1.7 times the torque obtained at design speed and pressure ratio. Comparison of zero speed torque curves of figures 10(a) and (b) (air and argon tests, respectively) shows that higher equivalent torque values were obtained with the tests in air. Examination of zero speed equivalent weight flow data at constant equivalent pressure ratio showed that the weight flow was higher for the tests in air as compared to tests in argon. This accounts for the higher zero speed equivalent torque values obtained for the air tests.

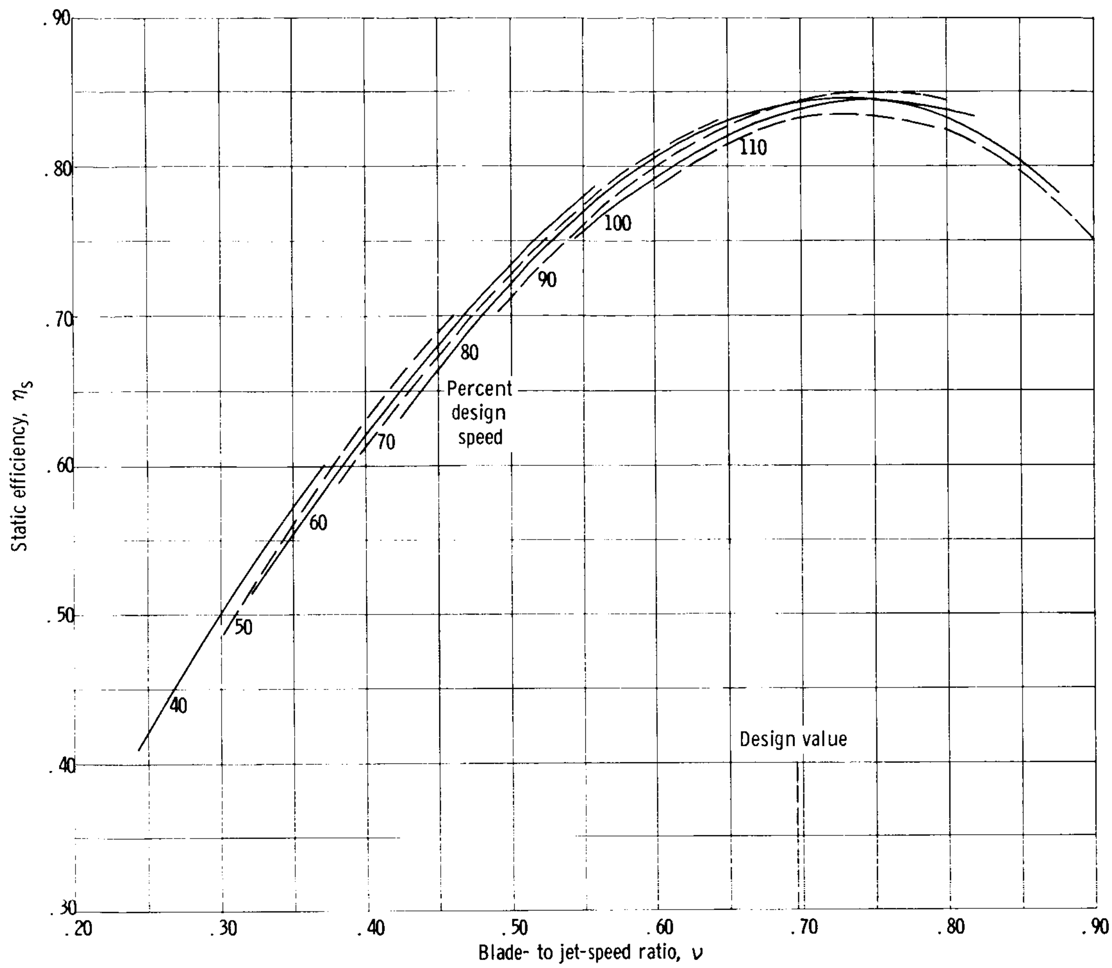


(a) Using cold air as driving fluid.

Figure 12. - Variation of static efficiency with blade- to jet-speed ratio.

The torque-speed characteristic curve is shown in figure 11, where the dimensionless torque parameter η_s/ν is plotted against the blade- to jet-speed ratio. Cold air test data at design pressure ratio were used. The curve shown differs significantly from those of axial flow turbines, which approximate straight lines. The curvature shown by the curve illustrates one of the major differences in performance between axial and radial inflow turbines. Radial turbine velocity diagrams are more sensitive to changes in pressure ratio and speed because of the centrifugal pressure gradient in the rotor. This is discussed in reference 5 where changes in speed in an axial turbine are compared with changes in a radial turbine. Decreasing speed from the maximum efficiency point resulted in greater rotor viscous losses in the radial turbine, while increasing speed resulted in greater rotor incidence losses in the radial turbine. The reference analysis showed that in the radial machine, efficiency decreases more rapidly with deviation from optimum blade- to jet-speed ratio than in the axial machine.

Figures 12 and 13 present the static and total efficiencies of the turbine over the range of turbine blade- to jet-speed ratios investigated for lines of constant blade speed. These figures show the decrease in efficiency as the

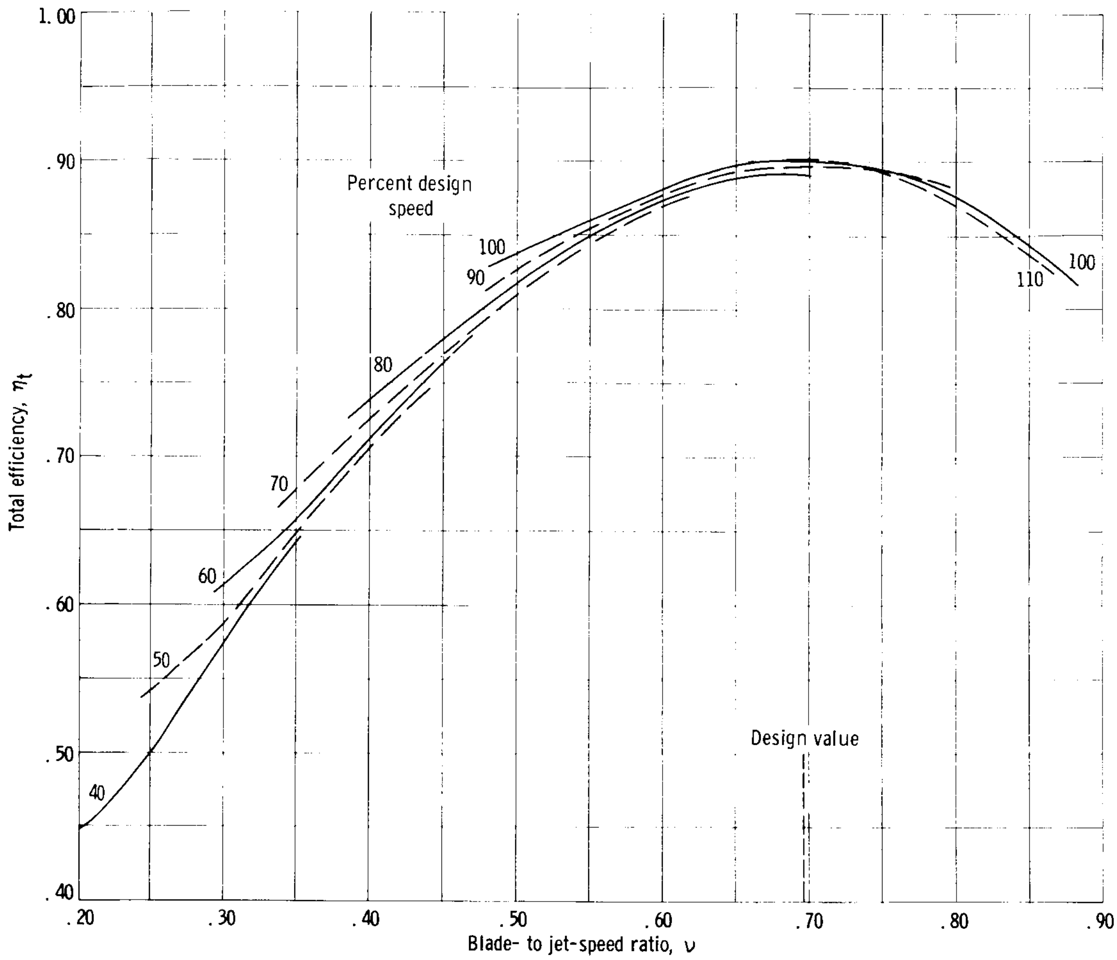


(b) Using cold argon as driving fluid.

Figure 12. - Concluded.

blade- to jet-speed ratio deviates from the design value of 0.697. The figure shows that a static efficiency of 0.84 and total efficiency of 0.90 were obtained at equivalent design point operation. These values are higher than the respective design point values of 0.824 and 0.880.

A radial survey of exit total pressure and flow angle was made at equivalent design pressure ratio and speed for the cold air tests. Figure 14(a) (p. 19) presents the variation of exit total pressure with radius ratio. Exit total pressure has been divided by the inlet total pressure in order to eliminate the effects of slight changes of inlet total pressure with time. This figure shows the exit total pressure to be reasonably constant across the passage with slightly higher values near the walls. Figure 14(b)(p. 19) presents the variation of exit flow angle with radius ratio. Design flow angles are also shown in this figure. It shows that design angle distribution was not obtained and that there was overturning (as indicated by negative exit angles) over most of the passage. The radial variation in specific work, as a result of this angle distribution, was calculated to be approximately 8 percent of the total

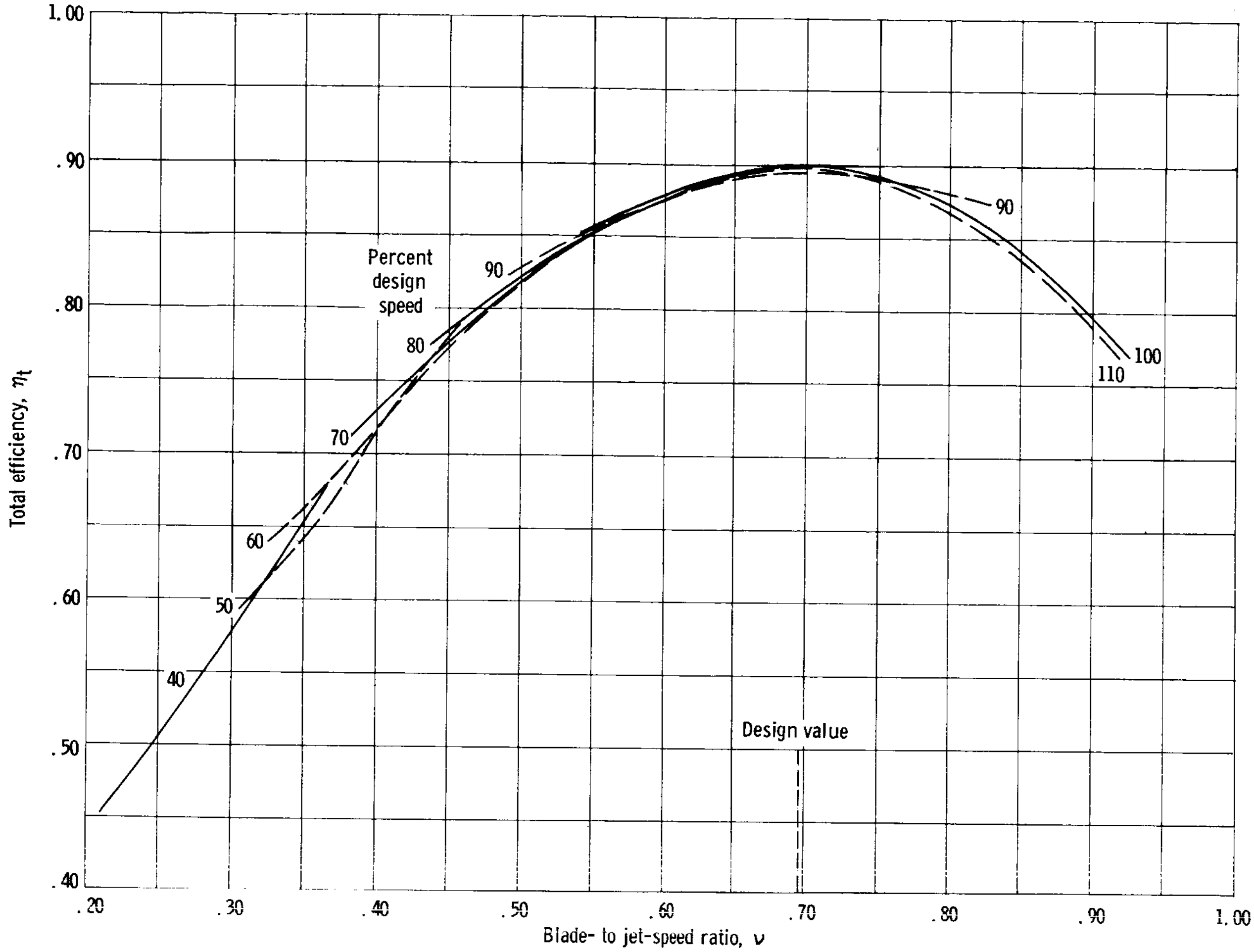


(a) Using cold air as driving fluid.

Figure 13. - Variation of total efficiency with blade- to jet-speed ratio for lines of constant speed.

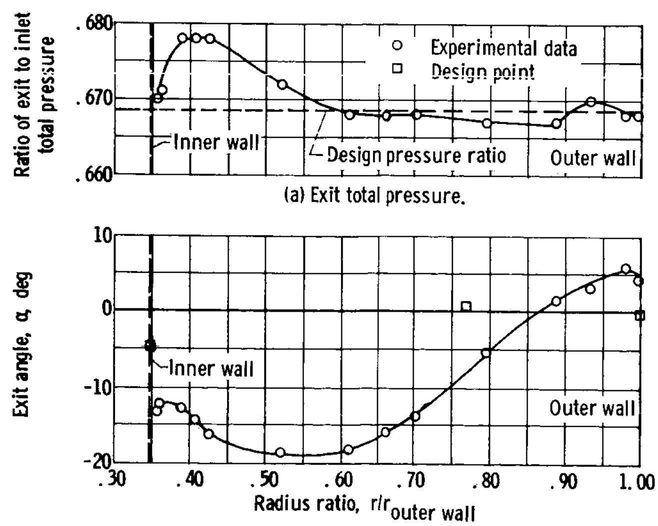
work. Calculations based on the measured total pressure ratio and exit angle distributions, as shown in figure 14(b), indicated that there was an approximate 10 point variation in local total efficiency from hub to shroud.

A comparison of the turbine performance with argon and air as the working fluid is shown in figure 15 (p. 20) where total efficiency is plotted against blade- to jet-speed ratio. The data shown are for design rotative speed over a range of pressure ratios. Figure 15 shows that, at design blade- to jet-speed ratio, the total efficiency was 0.90 for the test in argon and in air. This figure also indicates that the turbine performance was essentially the same for both argon and air over the range of pressure ratios investigated. The conclusion can therefore be drawn that there is no significant effect of specific heat ratio on turbine performance. This agrees with the results of the analytical investigation of reference 6, which indicated no significant effect of specific heat ratio on turbine performance over a range of specific heat ratios of 1.2 to $1\frac{2}{3}$.



(b) Using cold argon as driving fluid.

Figure 13. - Concluded.



(a) Exit total pressure.

(b) Exit flow angle.

Figure 14. - Variation of exit total pressure and flow angle with radius ratio at design point operation in air.

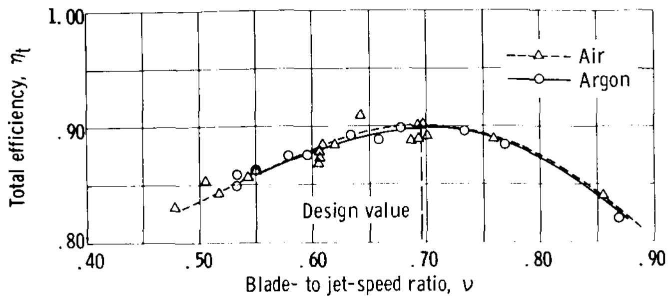
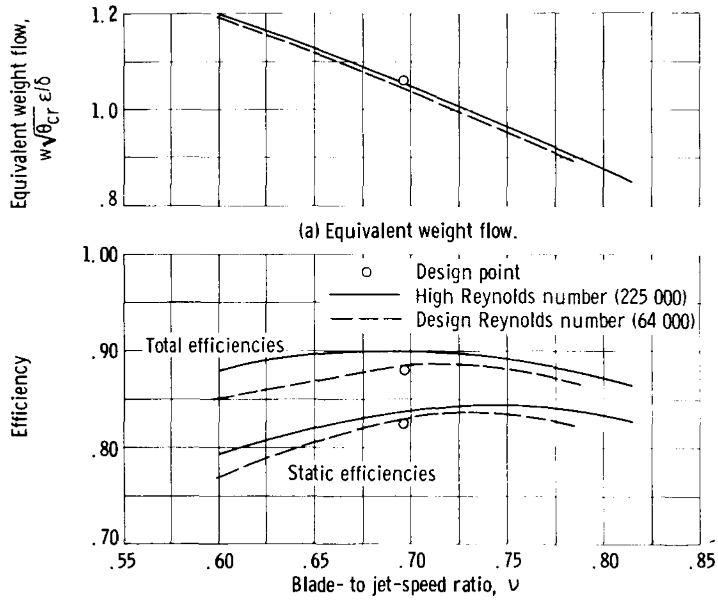


Figure 15. - Comparison of turbine total efficiency at design equivalent speed for two working fluids investigated.



(a) Equivalent weight flow.

(b) Turbine efficiency.

Figure 16. - Effect of Reynolds number on turbine performance using argon as driving fluid.

Performance at Design Reynolds Number of 63 700

The performance results, as discussed in previous sections, were obtained at a Reynolds number of approximately 225 000. Reynolds number is defined herein as $Re = w/\mu r_t$. Since the turbine will be operating at a Reynolds number of 63 700 at hot design operation in argon, tests were made at a reduced inlet pressure of 3.40 pounds per square inch absolute and an inlet temperature of 610° R in order to establish the efficiency level at this low Reynolds number. Operation at this reduced inlet pressure and higher inlet temperature results in a Reynolds number of approximately 63 700 at design pressure ratio and speed. Data were obtained over a series of pressure ratios at design speed with argon as the working fluid. Figure 16 shows the results of these tests as well as the results obtained at the high Reynolds number over the same blade- to jet-speed ratio range.

Figure 16(a) shows equivalent weight flow as a function of blade- to jet-speed ratio for both high and design Reynolds numbers. Blade- to jet-speed ratio was used for this figure to facilitate correlation of efficiency with equivalent weight flow. The figure shows that, at design blade- to jet-speed ratio, there was a 1.0 percent reduction in equivalent weight flow when the Reynolds number was reduced from 225 000 to 63 700. The reduction in weight flow from 1.050 to 1.040 pounds per second results from increased viscous losses, which reduced the effective flow area. Examination of the stator exit static pressure at design speed and design total- to static-pressure ratio indicated that the free stream velocity at the stator exit increased as the inlet pressure was decreased from 9.5 pounds per square inch absolute for the high Reynolds number case, to 3.4 pounds per square inch absolute for the design Reynolds number case. Apparently the stator effective area (as affected by losses) is decreasing faster than the rotor effective area. Comparison of stator exit static pressure and rotor exit static pressure indicated that the reaction across the rotor was higher than design for the high Reynolds number case. The reaction across the rotor decreased with decreasing inlet pressure and was near design value at the design Reynolds number

TABLE I. - PERFORMANCE VALUES

	Design	Air tests	Argon tests	Argon tests
Reynolds number, Re	63 700	225 000	225 000	63 400
Total efficiency, η_t	0.880	0.90	0.90	0.88
Static efficiency, η_s	0.824	0.84	0.84	0.83
Equivalent specific work, $\Delta h/\theta_{cr}$, Btu/lb	11.9	12.13	12.08	11.96
Equivalent weight flow, $w\sqrt{\theta_{cr}}\ \epsilon/\delta$, lb/sec	1.063	1.060	1.050	1.040
Equivalent torque, $\tau\epsilon/\delta$, in.-lb	50.05	50.8	50.9	49.2

case. Equivalent weight flow at design Reynolds number and pressure ratio is 2.2 percent lower than the design value of 1.063 pounds per second. A reduction in weight flow with decreasing Reynolds number was also obtained in the performance evaluation of a single-stage axial flow turbine reported in reference 4.

Figure 16(b) shows turbine efficiency, total and static, as functions of blade- to jet-speed ratio for both Reynolds numbers. At design Reynolds number (63 700), the total efficiency was 0.88 and the static efficiency was 0.83. These efficiency values are in good agreement with the design point values of 0.880 and 0.824.

Comparison of experimentally obtained efficiencies for both Reynolds numbers shows that the total efficiency decreased by 2.0 points and the static efficiency decreased by 1.0 point when the Reynolds number was reduced to the design value. The greater reduction in total efficiency results from the reduction in total efficiency results from the reduction in rotor reaction and velocity level.

In order to obtain an overall evaluation of turbine performance for the range of conditions investigated, design point performance results are given in table I along with the values assumed in the design of the turbine.

SUMMARY OF RESULTS

A 6.02-inch tip diameter radial inflow turbine was designed and fabricated under contract for a 10-kilowatt shaft output space power system with argon as the working fluid. An experimental investigation of the turbine at two values of Reynolds number was conducted with both argon and air as the working fluids.

Results of the investigation are presented as follows for operation at a Reynolds number of 225 000:

1. Cold air performance of the turbine at design speed and blade- to jet-speed ratio showed that the total efficiency was 0.90 and the equivalent specific work was 12.1 Btu per pound. These values show that the turbine performed better than the design values of 0.880 and 11.9 Btu per pound. At this point, a static efficiency of 0.84 was obtained, which is slightly higher than the design value of 0.824.

2. Cold air performance at design speed and pressure ratio showed that the equivalent weight flow was 1.060 pounds per second, which compares closely with the design value of 1.063.

3. Comparison of results obtained with argon and air as the working fluid indicated no significant difference in turbine performance over the range of pressure ratios and speeds investigated. The conclusion can therefore be drawn that there is no significant effect of specific heat ratio on turbine performance.

The following results were obtained when the turbine was operated at design Reynolds number of 63 700 with argon as the working fluid:

1. At design speed and blade- to jet-speed ratio, total and static efficiencies were 0.88 and 0.83. These values are in good agreement with the design values of 0.880 and 0.824.

2. A 2.0-point reduction in total efficiency was obtained at design speed and blade- to jet-speed ratio when the turbine operating Reynolds number was reduced from approximately 225 000 to the design value of 63 700.

3. An equivalent weight flow of 1.040 pounds per second was obtained at equivalent design speed and pressure ratio. This amounts to an approximate 1.0 percent decrease from 1.050 pounds per second obtained at the high Reynolds number. This reduction in weight flow results from increased viscous losses through the turbine when the Reynolds number was decreased. Comparison of stator exit pressures for the two Reynolds numbers indicated that the stator effective area (as affected by losses) was decreasing faster than the rotor effective area since the stator exit free stream velocity increased with decreasing Reynolds number.

Lewis Research Center,
National Aeronautics and Space Administration,
Cleveland, Ohio, May 19, 1965.

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