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RADIAL-INFLOW TURBINE PERFORMANCE WITH EXIT DIFFUSERS DESIGNED FOR LINEAR STATIC-PRESSURE VARIATION

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16. Abstract

Two alternative turbine exit diffusers were designed with an area ratio identical to that of the original but with a significantly different axial static-pressure variation. These two diffusers had the same flow area variation; they differed only in the contours of the inner and outer walls. Each diffuser was tested as part of the turbine with cold argon at design Reynolds number. Overall total-to-total efficiency for operation at the design point with either of the two alternative diffusers was about 0.906 as compared to 0.894 for operation with the original. The diffuser loss was reduced from 0.019 to 0.007 in terms of overall total efficiency.

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SUMMARY

Two turbine exit diffusers were designed for the turbine of a 2- to 10-kilowatt Brayton cycle rotating unit in an effort to obtain an improvement in performance over that of the diffuser originally designed for the unit. The two alternative diffusers were designed with the same area ratio as the original but with slightly greater length. The flow area was varied to give a linear variation of static pressure with axial distance through both diffusers. They differed only in the contours of the inner and outer walls.

The diffusers were tested successively as parts of the turbine assembly. Tests were made with cold argon at inlet total conditions of 4.8 newtons per square centimeter absolute (7.0 psi) and 339 K (610° R). These conditions correspond to design Reynolds number at equivalent design speed and pressure ratio. Overall turbine efficiency is presented for each diffuser over a range of turbine equivalent total- to static-pressure ratios from 1.45 to 2.10 and a speed range from 30 to 100 percent of equivalent design. Results are compared with those obtained for operation with the original diffuser.

Operation of the turbine at the equivalent design point with either of the alternative diffusers resulted in an overall total efficiency of about 0.906, which is 1.2 points larger than that obtained with the original diffuser and represents a diffusion penalty of only 0.007 in total efficiency. This improvement indicates a potential increase of about 3.0 percent in system electrical net power output.

The static-pressure recovery in either alternative diffuser was larger than that for the original diffuser. This improvement is reflected in a $1\frac{1}{2}$-point increase in overall static efficiency to a value of 0.901.

INTRODUCTION

The Brayton cycle space power technology program being conducted at the NASA

Lewis Research Center includes an investigation of a single-shaft system wherein a 12.62-centimeter (4.97-in.) radial-inflow turbine drives a compressor and alternator. The system is designed for a shaft rotative speed of 36 000 rpm with a power range of 2 to 10 kilowatts (electrical).

Components are being investigated experimentally, both as isolated units and in combination with other components. The turbine efficiency is of great importance in determining the performance of the complete Brayton power system. Reference 1 states that a point in turbine efficiency is equivalent to about $2\frac{1}{2}$ percent in net system power output. As part of the program, a research model of the turbine was fabricated as one component of a complete package procurement and delivered to the Lewis Research Center for an aerodynamic evaluation.

A description of the mechanical and aerodynamic design of the turbine is given in reference 2. The turbine assembly included an exit diffuser with a cylindrical inner body and a conical outer wall. The turbine was tested with and without the diffuser in cold argon. This investigation and design information are reported in reference 3. Results indicated that the diffuser loss was 0.02 in overall total-to-total efficiency. This value is about 1.5 times the design loss.

Design flow characteristics in the diffuser were then examined with a one-dimensional calculation of velocity and static pressure as functions of flow area and total-pressure loss. These calculations showed a large rate of deceleration of the flow near the inlet of the diffuser. This is generally considered the best design because of the thinner boundary layer and higher Reynolds number here. The boundary-layer separation characteristics, however, may be influenced by the unsteadiness of the flow due to the rotor blade wakes and loss accumulations. Accordingly, an alternate diffuser (2) was designed with the same design area ratio but with a smaller rate of deceleration near the inlet. This diffuser was designed for a linear increase in static pressure with distance from the inlet.

The experimental investigation of the turbine with diffuser 2 is reported in reference 1. The results showed a diffuser loss of less than 1 point in total-to-total efficiency, which is an improvement over the original diffuser. However, the results indicated flow separation from the inner body of the diffuser with the associated total-pressure loss.

A third diffuser was then designed with an axial variation in flow area and static pressure identical to that of diffuser 2 but with different contouring of the flow passage. Diffuser 3 was built and tested under conditions similar to those used in tests of the other two diffusers. They were tested as a part of the 12.62-centimeter (4.97-in.) radial-inflow turbine. Tests were made with argon as the working fluid at an inlet temperature of 339 K (610° R) and an inlet pressure of 4.8 newtons per square centimeter absolute (7.0 psi). These values of total temperature and total pressure correspond to design Reynolds number at equivalent design speed and pressure ratio. Data were obtained over a range of blade-jet speed ratios by varying speed and pressure ratio.

This report presents turbine design information including the design characteristics of the original, second, and third diffusers. Overall turbine performance is presented for the turbine operating with the two alternate diffusers. The same information for the original diffuser is included for purposes of completeness and comparing results.

SYMBOLS

A	flow area, cm^2; $in.^2$
H	isentropic specific work (based on total-pressure ratio), J/g; ft-lb/lb
Δh	specific work, J/g; Btu/lb
N	turbine speed, rpm
N_s	specific speed, $NQ^{1/2}/H^{3/4}$, dimensionless for SI units; rpm $(ft)^{3/4}/sec^{1/2}$
p	pressure, N/cm^2abs; psia
Q	volume flow (based on exit conditions), m^3/sec; ft^3/sec
Re	Reynolds number, $w/\mu r_t$
r	radius, m; ft
T	absolute temperature, K; ^oR
U	blade velocity, m/sec; ft/sec
V	absolute gas velocity, m/sec; ft/sec
V_j	ideal jet speed corresponding to total- to static-pressure ratio across turbine, m/sec; ft/sec
W	relative gas velocity, m/sec; ft/sec
w	mass flow, kg/sec; lb/sec
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 those using air inlet conditions at U.S. standard sea-level conditions, $(0.740/\gamma)\left[(\gamma + 1)/2\right]^{\gamma/(\gamma-1)}$
η_s	turbine static efficiency (based on inlet-total- to exit-static-pressure ratio)

η_t turbine 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, kg/(m)(sec); lb/(ft)(sec)

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

Subscripts:

cr condition corresponding to Mach number of unity

eq equivalent

t tip

u tangential component

1 station at turbine inlet (fig. 10)

2 station at rotor exit

3 station at diffuser exit

Superscripts:

' absolute total state

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

TURBINE DESCRIPTION

Turbine Design

The 12.62-centimeter- (4.97-in.-) tip-diameter radial-inflow turbine was designed for a 6.0-kilowatt net electrical output with a xenon-helium mixture as the working fluid. A detailed description of the turbine can be obtained from references 2 and 3. The design-point values for the turbine are as follows:

Inlet total temperature, T'_1, K; °R 1144; 2060

Inlet total pressure, p'_1, N/cm^2 abs; psia 17.24; 25

Mass flow, w, kg/sec; lb/sec 0.3395; 0.7484

Turbine rotative speed, N, rpm 36 000

Total- to total-pressure ratio

Overall, p_1'/p_3' 1.749

Rotor exit, p_1'/p_2' 1.740

Total- to static-pressure ratio

Overall, p_1'/p_3 . . . . 1.763

Rotor exit, p_1'/p_2 . . . . 1.800

Blade-jet speed ratio, ν 0.690

Total-to-total efficiency

Overall, $\eta_{t,1 \text{ to } 3}$ 0.886

Rotor exit, $\eta_{t,1 \text{ to } 2}$ 0.894

Total-to-static efficiency

Overall, $\eta_{s,1 \text{ to } 3}$ 0.875

Rotor exit, $\eta_{s,1 \text{ to } 2}$ 0.848

Specific work, Δh, J/g; Btu/lb 50.44; 21.67

Reynolds number, $Re = w/\mu r$ 76 200

Specific speed, $N_s = NQ^{1/2}/H^{3/4}$, dimensionless for SI units; rpm $(ft^{3/4})/sec^{1/2}$ 0.59; 76

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

Mass flow, $\epsilon w\sqrt{\theta_{cr}}/\delta$, kg/sec; lb/sec 0.2204; 0.4860

Specific work, $\Delta h/\theta_{cr}$, J/g; Btu/lb 34.29; 14.73

Rotative speed, $N/\sqrt{\theta_{cr}}$, rpm 29 687

Total- to total-pressure ratio

Overall, $(p_1'/p_3')_{eq}$ 1.658

Rotor exit, $(p_1'/p_2')_{eq}$ 1.645

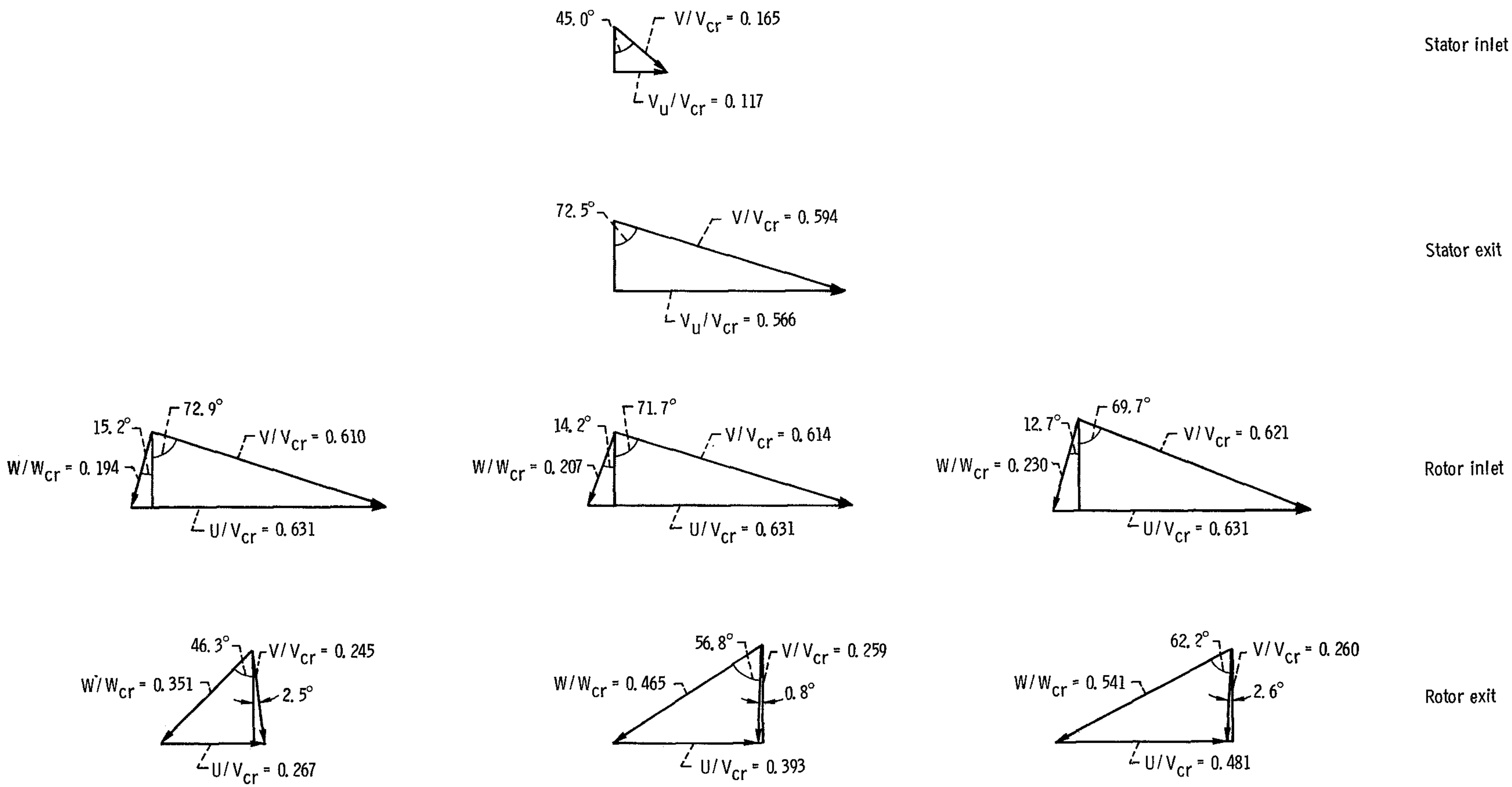
Total- to static-pressure ratio

Overall, $(p_1'/p_3)_{eq}$ 1.669

Rotor exit, $(p_1'/p_2)_{eq}$ 1.695

Blade-jet speed ratio, ν 0.690

Design velocity diagrams (inside the blade rows) corresponding to these conditions and to the selected turbine geometry are shown in figure 1. Figure 2 shows a section through the research turbine with the original exit diffuser. The turbine tip diameter, as noted previously, is 12.62 centimeters (4.97 in.). Exit shroud and hub diameters are 8.839 and 4.628 centimeters (3.480 and 1.822 in.), respectively.



(a) Hub.

(b) 50-Percent streamline.

(c) Shroud.

Figure 1. - Design velocity diagrams (inside blade row).

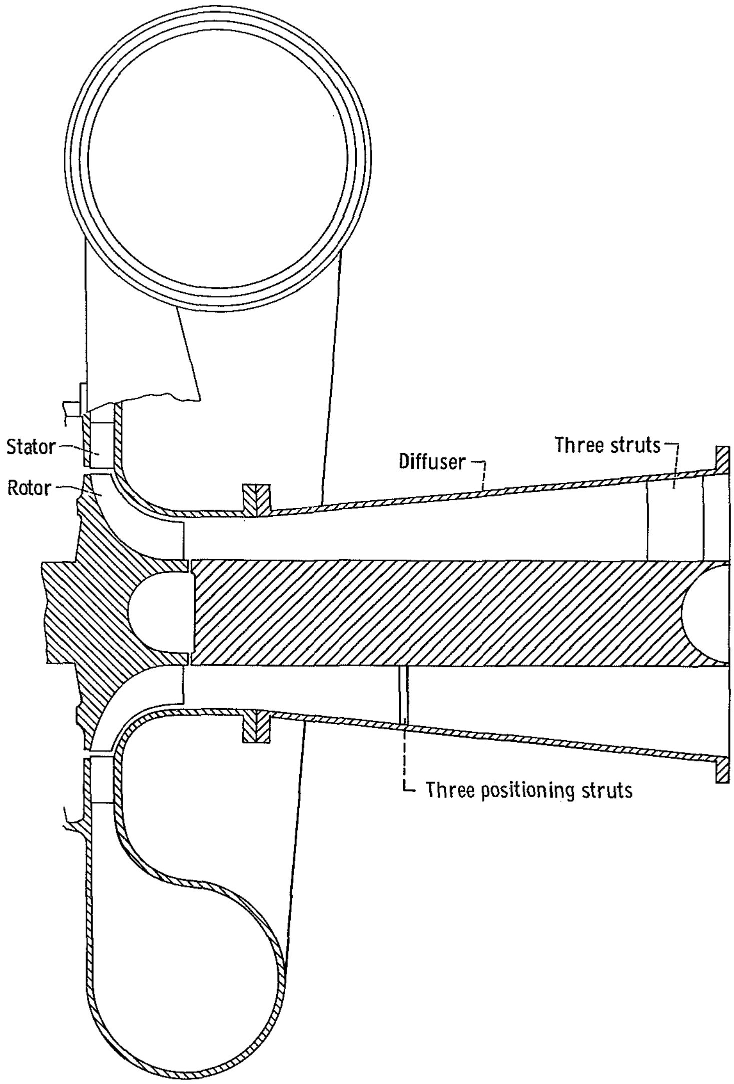


Figure 2. - Turbine with original diffuser.

Diffuser Designs

The original turbine exit diffuser (fig. 2) has inlet and exit areas of 44.452 and 118.000 square centimeters (6.890 and 18.290 in.^{2}), respectively. The overall axial length of the diffuser is 24.003 centimeters (9.450 in.); the dimension from the rotor blade trailing edge to the diffuser exit is 24.318 centimeters (9.574 in.). The cylindrical inner wall and conical outer wall provide a nearly linear flow area variation with axial distance.

Calculations were made of the gas flow velocities and associated static pressures through the diffuser. These calculations were based on continuity and a design total-pressure loss occurring linearly with distance from the inlet to the outlet. There is a large deceleration of the flow at the inlet with very little at the exit. The associated inlet static-pressure gradient is approximately twenty times as large as that at the exit.

The calculated pressures were then used to obtain a diffuser effectiveness value, defined as $(p_3 - p_2)/(p_2' - p_2)$. This value corresponding to design total-pressure loss is 0.664. An effectiveness for isentropic, incompressible diffusion is defined as $1 - (A_2/A_3)^2$. This value is 0.858. Diffuser efficiency, defined as the ratio of actual to isentropic effectiveness, would therefore be 0.77 with design flow and design total-pressure loss.

A second diffuser (fig. 3) was designed with the purpose of shifting the maximum

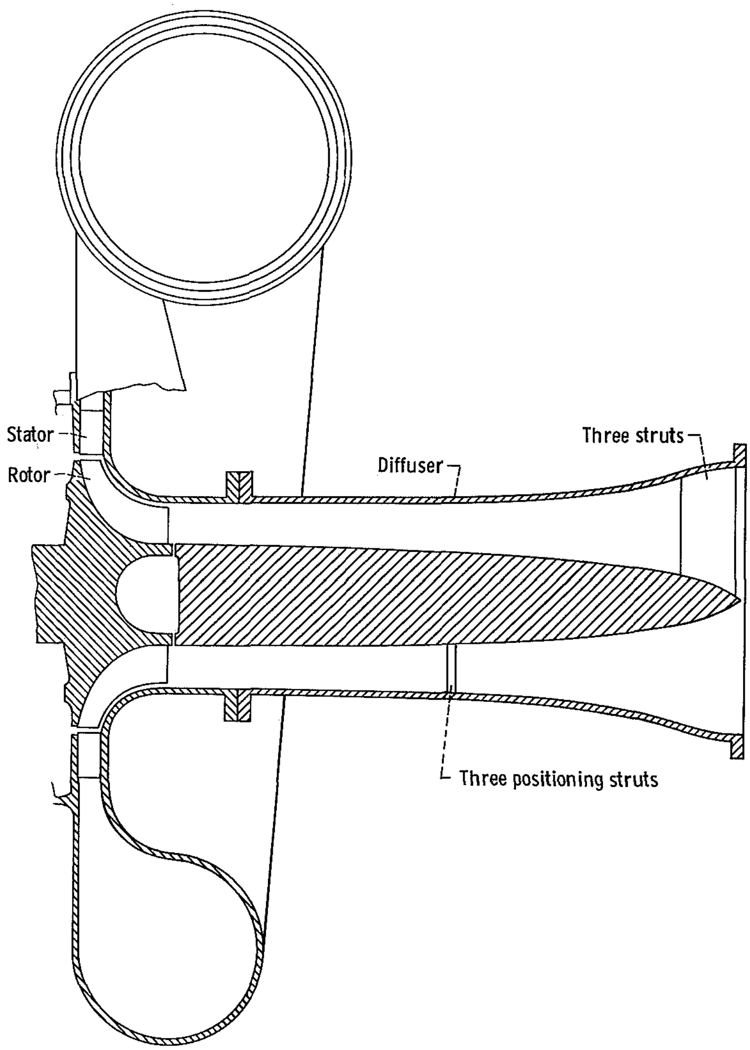


Figure 3. - Turbine with second diffuser.

rate of deceleration to the downstream part of the diffuser, where the kinetic energy level of the fluid is lowest. The inlet and exit flow areas were the same as for the original diffuser. However, a slightly greater length was permissible without exceeding the limiting size of the turbine-compressor-alternator package. This length, 26.492 centimeters (10.430 in.), is about 10 percent greater than that of the original. The flow area was determined to obtain a linear increase of static pressure with distance from the inlet. For 80 percent of the diffuser length the increase in annular area is accomplished by an increase in the diameter of the outer wall at a rate equal to that of the decrease in diameter of the inner wall. The mean diameter of the flow passage thereby remains constant. The inner body is then tapered to a small radius at the exit, and the outer surface is contoured to provide the desired pressure distribution. This diffuser and the associated turbine performance were described in reference 1.

A third diffuser (fig. 4) was designed with the same flow area variation and length

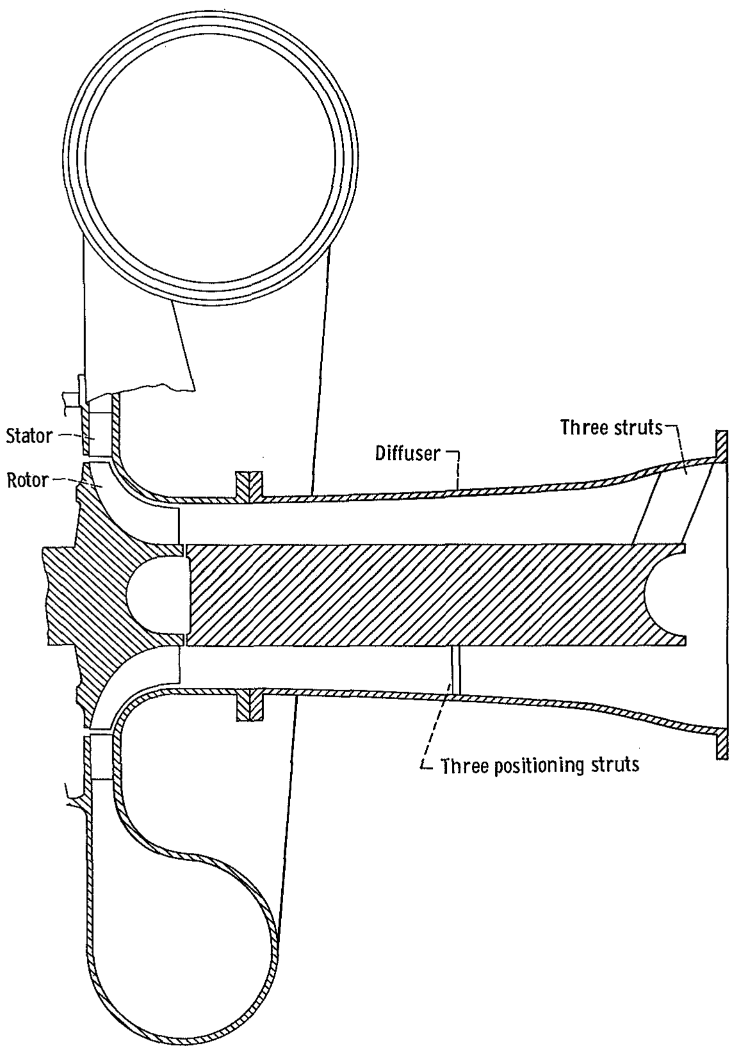


Figure 4. - Turbine with third diffuser.

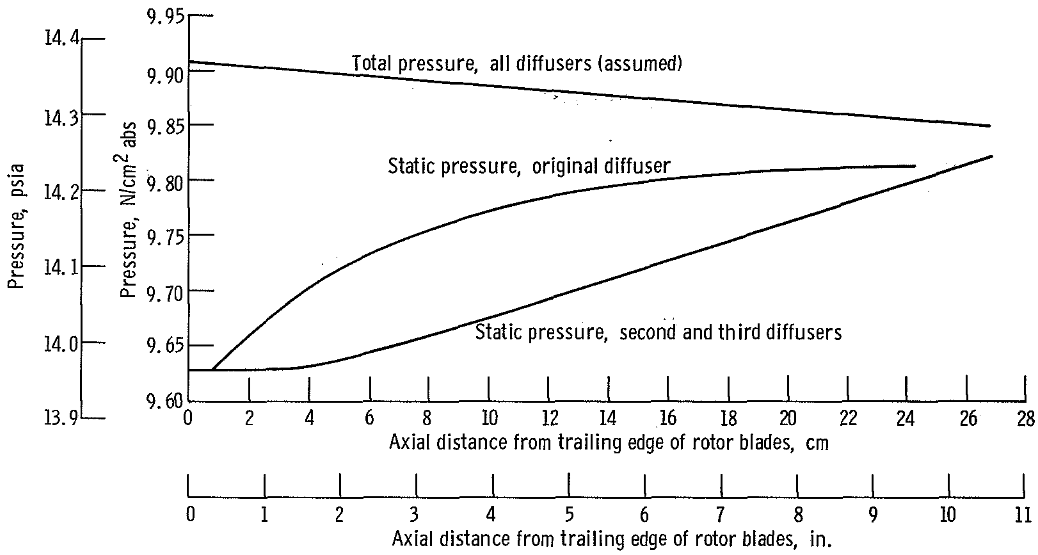


Figure 5. - Calculated pressure distributions in diffusers.

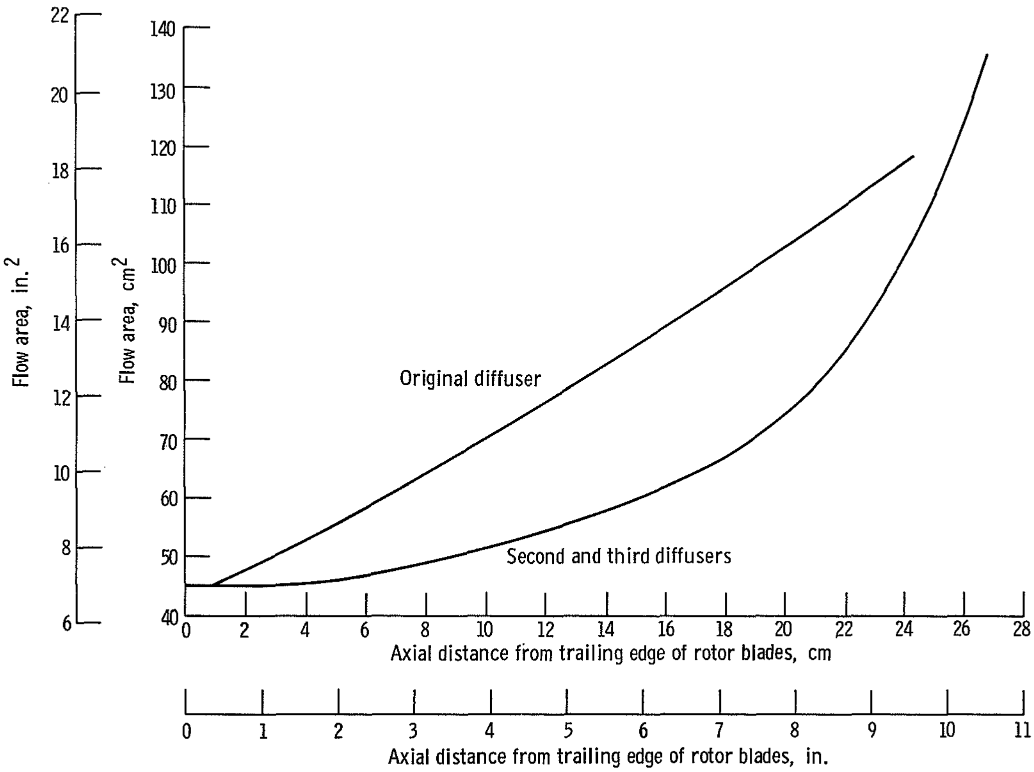


Figure 6. - Calculated flow area variation through diffusers.

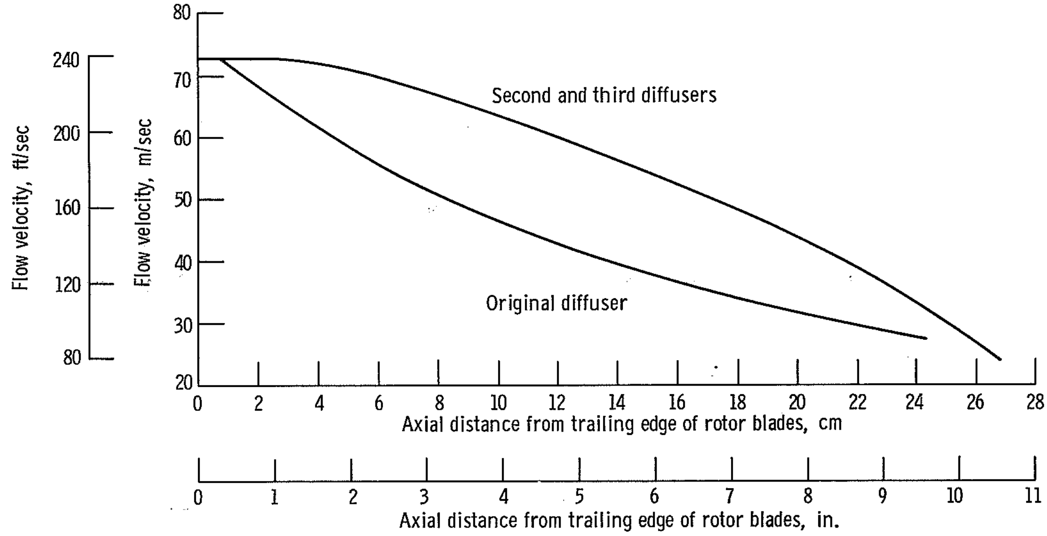


Figure 7. - Calculated velocity distributions through diffusers.

requirements which were used for the second diffuser but with the cylindrical inner body of the original diffuser. The desired flow area variation was obtained by contouring the outer wall only. A comparison can then be made between the results obtained with the two designs to determine the effect on performance of passage contour.

The design variation in total and static pressures through the three diffusers is shown in figure 5. Linear total-pressure variation was assumed as shown. The rate of static-pressure rise with axial distance is the same for the two alternate diffusers but differs markedly from that for the original diffuser, particularly near the inlet.

The flow area variation through the three diffusers is shown in figure 6. This variation is nearly linear for the original diffuser. The second and third diffusers, however, have a very gradual increase near the inlet and then an increasingly rapid rate of increase toward the exit.

The variation in flow velocities which corresponds to the total- and static-pressure variation is shown in figure 7. As stated previously, the region of maximum change in velocity (deceleration) has been shifted from the inlet region of the original to the exit region of the two alternate diffusers.

APPARATUS, INSTRUMENTATION, AND PROCEDURE

The test facility, instrumentation, and procedure for testing and calculating performance parameters were the same as those described in detail in reference 3 with some additional instrumentation.

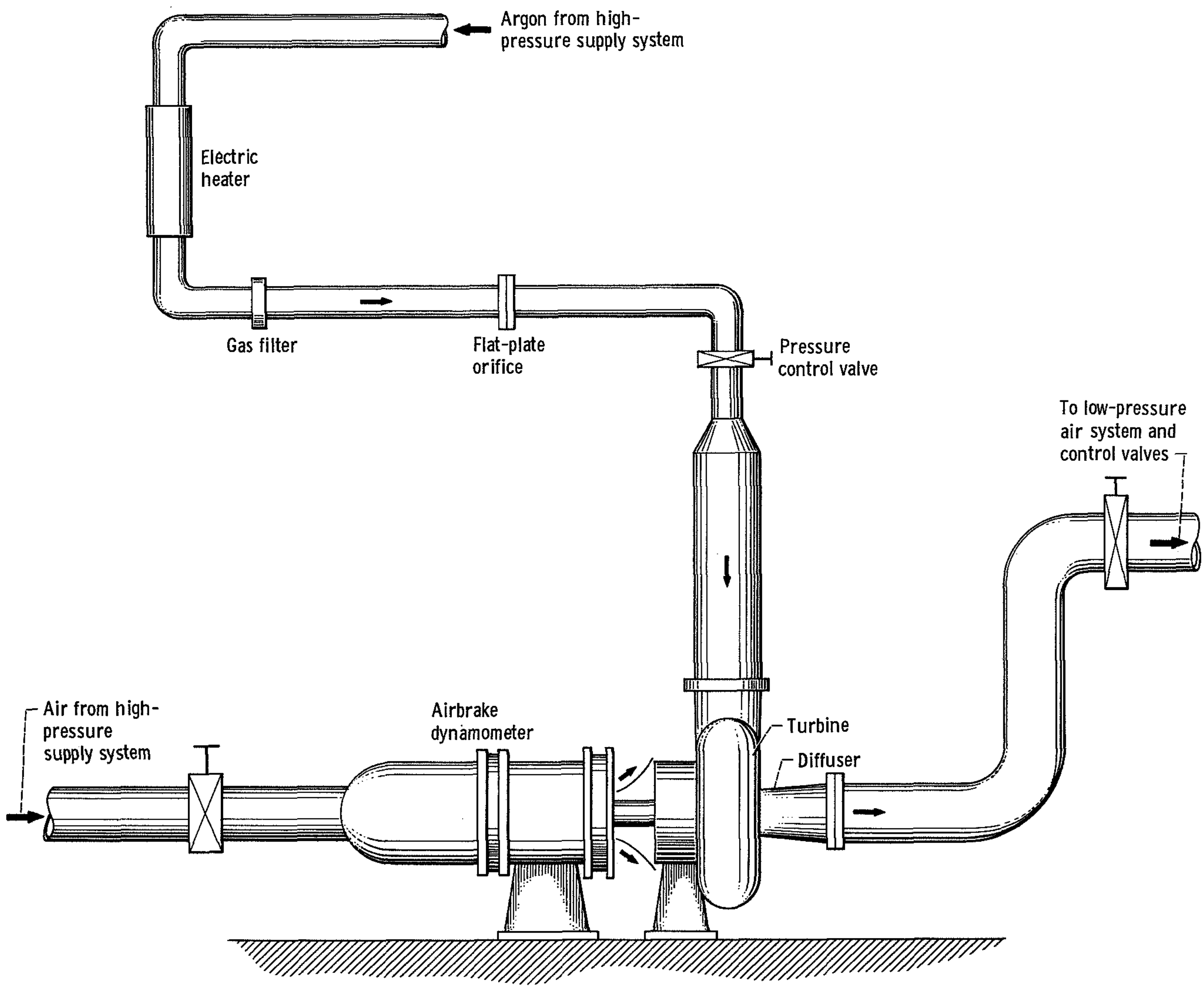


Figure 8. - Schematic of experimental equipment.

The apparatus included a 12.62-centimeter (4.97-in.) radial-inflow turbine with an airbrake dynamometer and two alternate design diffusers described in the previous section. Figure 8 shows a schematic of this apparatus connected to the inlet and exhaust piping system with flow control, heating, filtering, and measuring devices. The turbine test facility is shown in figure 9 with a diffuser installed.

The instrumentation was the same as that described in reference 3 except for some additional measuring stations. In this reference we described the evaluation of the same turbine with the original diffuser. For the subject investigation radial survey instrumentation was added at station 2 immediately downstream of the turbine rotor (fig. 10). This instrumentation consisted of a self-balancing probe which measured flow angle, total pressure, and total temperature. At the same station eight static-

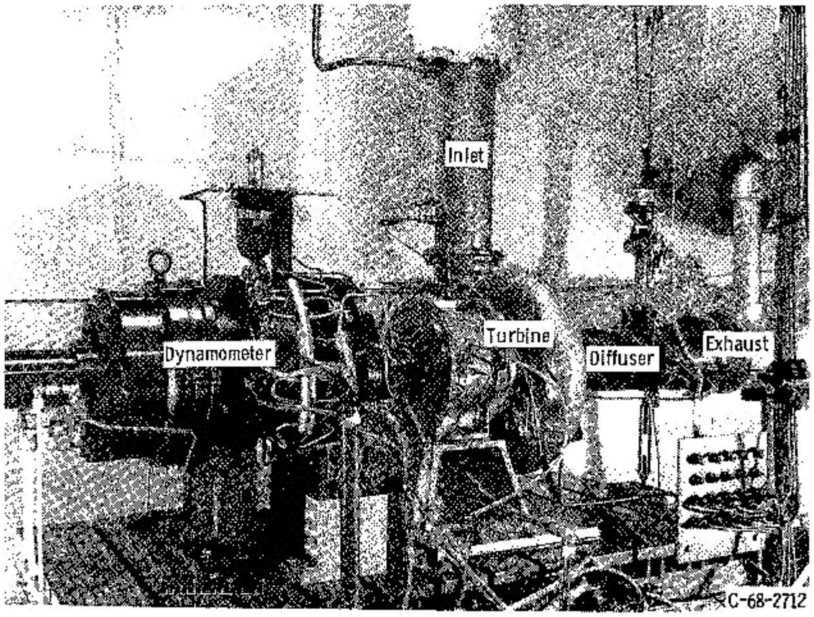


Figure 9. - Installation of turbine in turbine-component test facility.

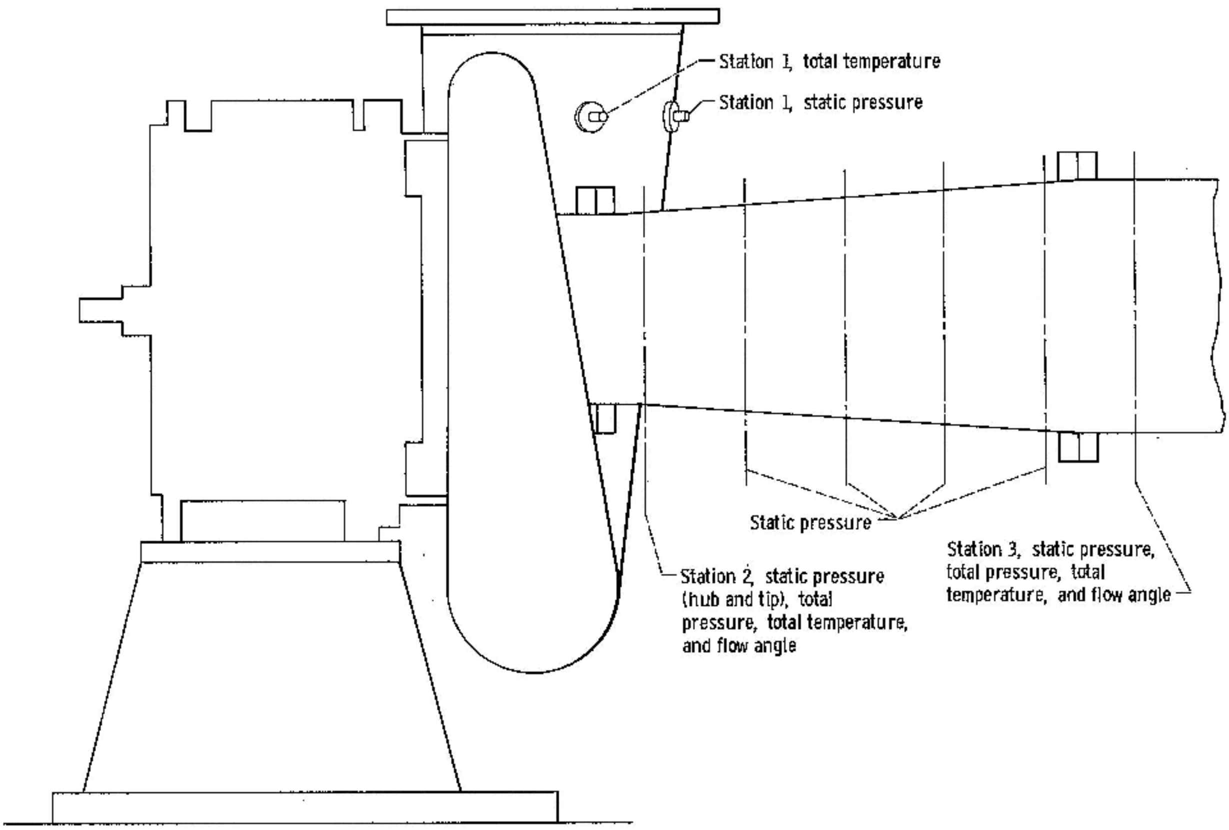


Figure 10. - Turbine instrumentation stations.

pressure taps were added, four in the inner body and four in the outer wall, equally spaced circumferentially. Four additional static taps were added in the diffuser. These taps were installed in the same circumferential position and spaced axially between station 2 and the diffuser exit (fig. 10).

The diffusers were tested only as parts of the turbine, never as individual components. Performance measurements, therefore, were made with flow conditions of high turbulence and radial gradients in flow, pressure, temperature, and flow angle.

The tests described in this report were made with cold argon at inlet total conditions of 4.8 newtons per square centimeter absolute (7.0 psi) and 339 K (610° R). These values of pressure and temperature correspond to a Reynolds number of about 75 500 at design operation. Data were obtained over a range of turbine equivalent total- to static-pressure ratios $(p_1'/p_2)_{eq}$ from 1.45 to 2.10 and a speed range from 30 to 100 percent of equivalent design. Evaluation of the diffuser performance at a low pressure depends on small differences in measured pressures and could result in significant errors. Accordingly, additional runs were made with the second diffuser at a higher turbine-inlet pressure of 13.8 newtons per square centimeter absolute (20 psi) in order to verify the results obtained at the lower pressure.

RESULTS AND DISCUSSION

In an effort to reduce diffusion loss and improve overall turbine performance, two alternate diffusers were designed, fabricated, and tested. Tests were performed at two values of turbine inlet pressure as described in the preceeding section. The smaller pressure corresponds to design Reynolds number; the larger pressure furnishes a verification of the data taken at the lower pressure. There was no difference in performance as calculated from measurements at the two pressure levels. This report presents only those results obtained at design Reynolds number. The results are first presented in terms of overall total and static efficiencies for operation with the original (from ref. 3) and two alternate design diffusers. A comparison of these results is included. A second section presents the performance of the two alternate diffusers in terms of pressure distributions and effectiveness. A comparison is made with the original diffuser.

Overall Performance

The overall total efficiency for operation with each of the three diffusers is presented in figure 11. Total efficiency (including the diffuser) is plotted as a function of

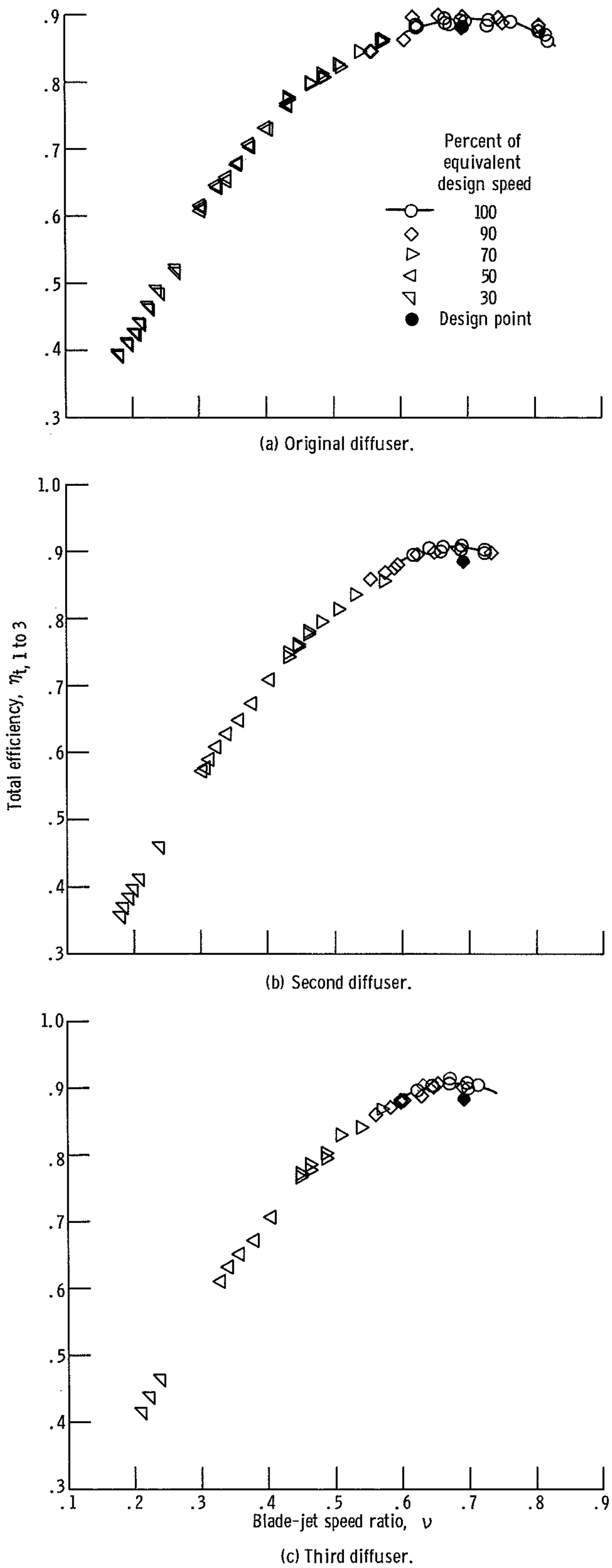


Figure 11. - Variation of total efficiency with blade-jet speed ratio. (Ideal work is calculated from conditions at the turbine inlet and diffuser exit.)

blade-jet speed ratio for the five values of turbine speed covered in the investigation. Figure 11(a), taken from reference 3, shows results obtained for operation with the original diffuser. The solid line is faired through the equivalent design speed data points. At design blade-jet speed ratio (0.690) this curve shows a total efficiency of 0.894, which is near the peak value for the entire range of operation. The corresponding efficiency based on rotor exit total pressure was 0.913 (ref. 3). The diffusion penalty was therefore 0.019 in total efficiency. Similarly, figures 11(b) and (c) show the total efficiency values for operation with the second and third diffusers, respectively. The results obtained for these two diffusers differed very slightly. At design point the curves show a total efficiency value of about 0.906 for both diffusers. The diffusion penalty in total efficiency for either of the alternate diffusers was, therefore, about 0.007 as compared to 0.019 for the original diffuser. The increase in overall turbine efficiency represents a potential of 3.0 percent increase in system electrical net power output.

The design speed curves of figure 11 are shown in figure 12 for comparison. The

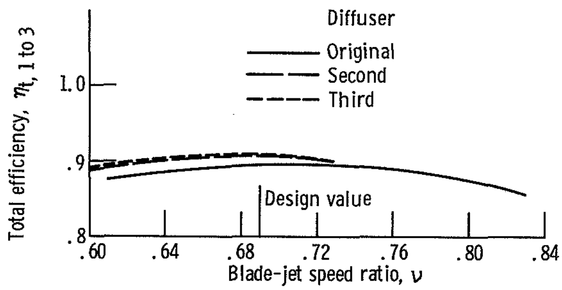
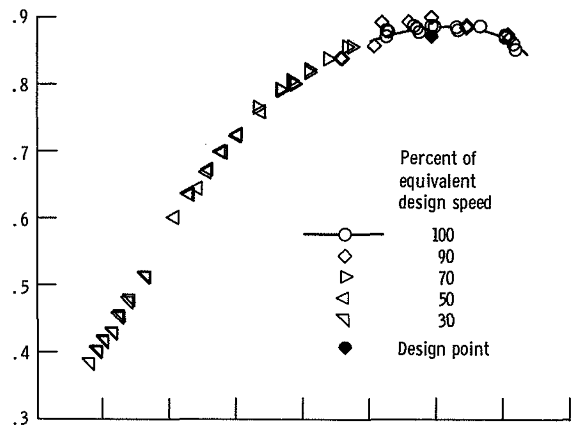


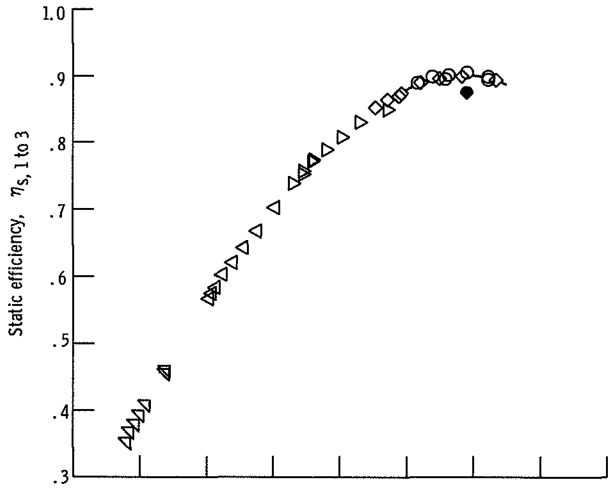
Figure 12. - Variation of overall total efficiency with blade-jet speed ratio at equivalent design speed for the three diffusers.

efficiency values for operation with the two alternate diffusers are about equal over the entire range of blade-jet speed ratios. These values are 1/2 to 2 points higher than those for operation with the original diffuser at corresponding values of blade-jet speed ratio.

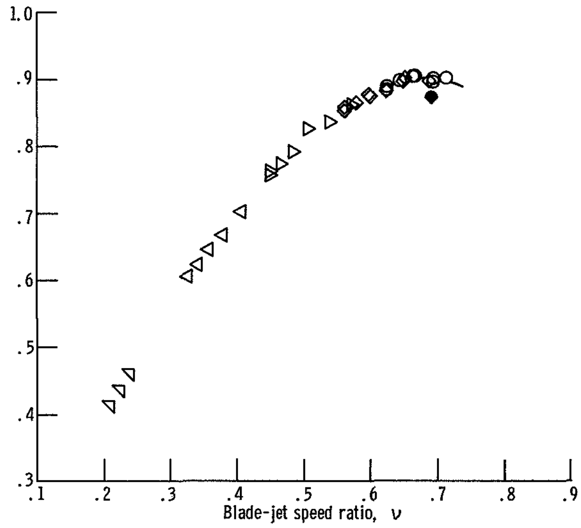
The overall static efficiency for operation with each of the three diffusers is presented in figure 13 as a function of blade-jet speed ratio. Results are shown for five values of turbine speed. Figure 13(a), taken from reference 3, shows results obtained for operation with the original diffuser. The curve through the data points for equivalent design speed shows the peak static efficiency value of 0.888 at design blade-jet speed ratio. The corresponding efficiency based on rotor exit static pressure was 0.872 (ref. 3). Thus, pressure recovery through the diffuser resulted in an increase



(a) Original diffuser.



(b) Second diffuser.



(c) Third diffuser.

Figure 13. - Variation of static efficiency with blade jet speed ratio. (Ideal work is calculated from conditions at the turbine inlet and the diffuser exit.

of about $1\frac{1}{2}$ points in static efficiency. Corresponding curves for the second diffuser (fig. 13(b)) and the third diffuser (fig. 13(c)) show a static efficiency value of about 0.901 at equivalent design speed and pressure ratio. The pressure recovery through these two diffusers, therefore, resulted in an increase in static efficiency of about 3.0 points. Compared with the original diffuser, the alternate diffusers show an improvement of about $1\frac{1}{2}$ points in overall static efficiency at the design point of operation.

Figure 14 shows the design speed curves of figure 13 for comparison. The curves for the two alternate diffusers are nearly identical. They show an improvement over the original diffuser for all values of blade-jet speed ratio. This improvement in overall static efficiency varies from about 1/2 point at the higher blade-jet speed ratios to about 2.0 points at the lower values. Table I summarizes the overall turbine efficiency values for design point operation.

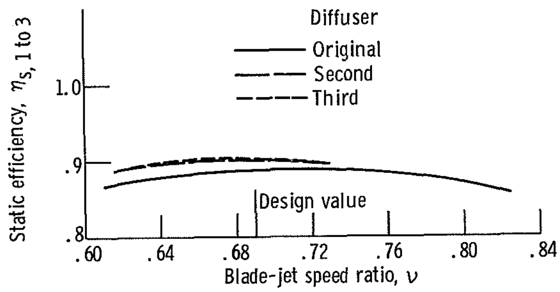


Figure 14. - Variation of overall static efficiency with blade-jet speed ratio at equivalent design speed for the three diffusers.

TABLE I. - PERFORMANCE VALUES

	Diffuser effectiveness[a]	Diffuser efficiency[a]	Overall turbine total efficiency	Overall turbine static efficiency
Design	0.664	0.77	0.886	0.875
Experimental				
Original diffuser	.40	.47	.894	.888
Second diffuser	.60	.68	.905	.900
Third diffuser	.65	.73	.907	.902

[a]See the section Diffuser Designs for definitions.

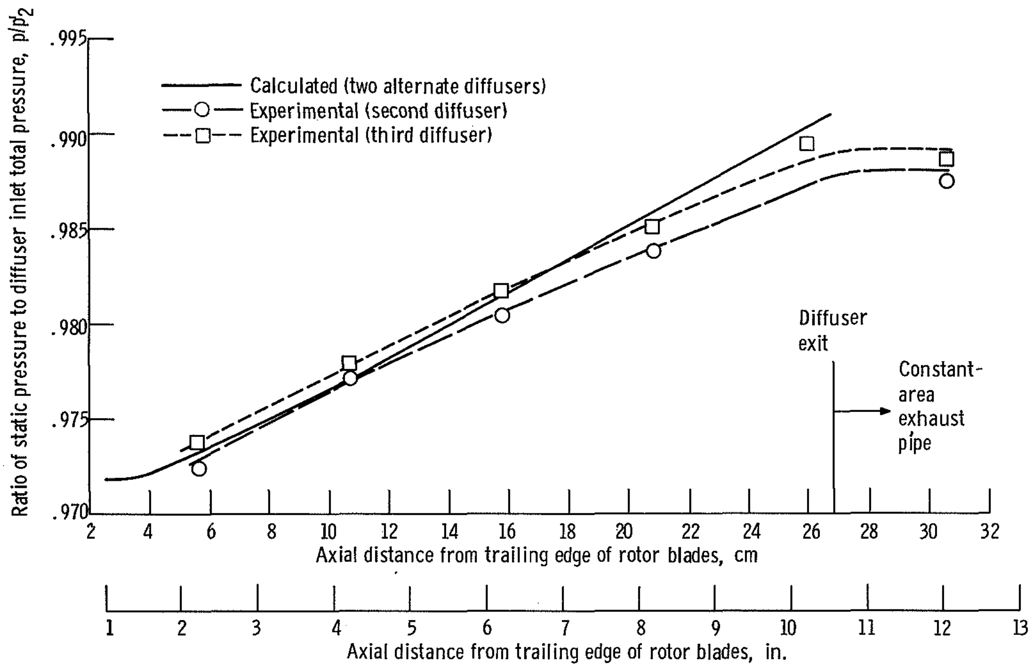


Figure 15. - Static-pressure distribution in diffusers.

Static-Pressure Distribution

As stated in the APPARATUS, INSTRUMENTATION, AND PROCEDURE section, static pressures were measured at five locations between the rotor exit and the diffuser exit and at one additional station immediately downstream of the diffuser. Figure 15 shows the values of these measured pressures at equivalent design speed and pressure ratio. The calculated curve is also shown for comparison. Pressures are shown as percentages of diffuser inlet total pressure in order to eliminate the effects of any variations in operating conditions. The effect of scatter in the individual readings was reduced by averaging the results of 18 data recordings. Experimental results for both alternate diffusers agree closely with the calculated curve. Nearly constant static pressure downstream of the diffuser exit results from a constant flow area in the exhaust pipe.

The pressure recovery in a diffuser can be used to express its performance, as discussed in the section Diffuser Designs. Diffuser effectiveness was defined as the ratio of static-pressure rise to the diffuser inlet dynamic pressure. Diffuser efficiency, then, is the ratio of actual to isentropic effectiveness. The performance of the two diffusers at the design point of operation is presented in table I. The performance of the original diffuser is included for comparison. Both the second and third diffusers showed a significant improvement in performance over the original diffuser. This improvement is attributed to a more favorable pressure distribution and is reflected in an increase of 1 to $1\frac{1}{2}$ points in overall efficiency (table I).

Radial surveys of total temperature, total pressure, and flow angle at the diffuser exit showed similar flow patterns for the two alternate diffusers. There appeared to be a region with very little mass flow near the center of the exit pipe for both designs. Thus, for a given variation in flow area, a change in the contour of the inner and outer walls did not have a significant effect on the performance of the diffuser.

SUMMARY OF RESULTS

Two exit diffusers were designed for a 12.62-centimeter (4.97-in.) radial-inflow turbine. This turbine is a component of a 2- to 10-kilowatt Brayton cycle space power system. The diffusers were designed in an effort to obtain an improvement in performance over the one originally designed for this turbine. The two alternative diffusers were designed for a linear variation of static pressure with axial distance through the diffuser. They differed only in the contours of the inner and outer walls. The diffusers were tested as parts of the turbine assembly. Tests were made with cold argon at the inlet total conditions of 4.8 newtons per square centimeter absolute (7.0 psi) and 339 K (610° R). These values of pressure and temperature correspond to a Reynolds number near design at design operation. Data were obtained over a range of turbine equivalent total- to static-pressure ratios from 1.45 to 2.10 and a speed range from 30 to 100 percent of equivalent design. The pertinent results of the investigation are as follows:

1. Operation of the turbine at the design point with either of the alternative diffusers resulted in an overall total efficiency of about 0.906. This value is about 1.2 points larger than that obtained with the original diffuser and represents a diffusion penalty in total efficiency of only 0.007. This improvement indicates a potential increase of 3.0 percent in system electrical net power output.

2. The overall static efficiency obtained with either alternative diffuser at the design point was about $1\frac{1}{2}$ points larger than that obtained with the original.

3. The axial static-pressure variation through both alternative diffusers agreed very well with design. These pressures indicated a significant improvement in diffuser efficiency over the original.

Lewis Research Center,
National Aeronautics and Space Administration,
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