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EXPERIMENTAL PERFORMANCE EVALUATION OF A 4.59-INCH RADIAL-INFLOW TURBINE WITH AND WITHOUT SPLITTER BLADES

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16. Abstract

Tests were conducted on the same turbine with and without splitter blades. Removal of the splitter blades resulted in a 0.6 point increase in total efficiency and a 0.7 point increase in static efficiency at design speed and pressure ratio. No significant change in flow rate was detected. Three Lewis computer programs were used to supplement the experimental results. Computer results indicate a flow eddy near blade leading edges with splitters removed.

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SUMMARY

An experimental investigation of the performance of 4.59-inch (11.66-cm) radial-inflow turbine was made. The purpose of the investigation was to determine the effect on performance of removing the splitter blades from the rotor. The turbine was tested in air both with and without splitter blades, using the same rotor.

Removal of the splitter blades resulted in a slight increase in efficiency at the design equivalent speed and pressure ratio. The increase was approximately 0.6 point for total efficiency and 0.7 point for static efficiency. No significant change in the flow rate was detected.

To supplement the experimental results, use was made of three previously published Lewis computer programs. These programs were used to study the relative velocity of the working fluid over the blade surfaces. The computer velocity calculations indicated that removal of the splitters causes eddies to form near the blade leading edges within the rotor passages. In real flow, the eddies may tend to decrease the efficiency. Removal of the splitters would also cause the turbine to operate at a nonoptimum incidence angle. The absence of significant decreases in efficiency was attributed to the reduction of the wetted area with removal of the splitters and the elimination of mixing losses at the splitter trailing edges.

INTRODUCTION

The space-power program conducted at the NASA Lewis Research Center has included the investigation of small aerodynamic components suitable for low-power Brayton cycle power systems. Components for both one-shaft and two-shaft systems have been investigated including both radial- and axial-flow turbomachinery. A discussion of both types of systems is given in reference 1.

As part of this program, a radial-inflow turbine with a tip daiameter of 4.59 inches (11.66 cm), designed to operate in a two-shaft Brayton cycle system having a 10-kilowatt power output, is under investigation.

This turbine has been previously tested to determine its general aerodynamic performance. Reference 2 describes performance tests conducted in cold argon at the design Reynolds number of 82 200. Cold argon tests to determine Reynolds number effects were reported in reference 3. The range of Reynolds number used was from 64 000 to 352 000 at design equivalent speed and pressure ratio. The turbine was also tested over a range of specific speed from 72 to 108 in cold air at design equivalent speed and pressure ratio. The variation in specific speed was accomplished by varying the volume of flow by means of interchangeable stators having different throat areas. The specific speed results were reported in reference 4.

The investigation reported here was prompted by questions regarding the effect of the number of rotor blades on turbine performance. Since the available turbine had 11 full blades and 11 splitters, it was decided to remove the splitters and determine the resulting change in performance. The same rotor was used for all tests, both with and without splitters. This procedure prevented other factors, such as shroud clearance, from influencing the results.

The investigation was made with air as a working fluid. Inlet total pressure was held constant over a range of speeds and pressure ratios. The turbine Reynolds number varied from 210 000 to 365 000 during the testing, which was well above the design value of 82 200.

To supplement the experimental results, use was made of the Lewis computer programs published as references 5 to 7. The purpose of using these programs was to obtain an estimate of the relative velocity distribution over the blade surfaces, both with and without splitter blades. The programs involve the assumption of ideal flow conditions, but empirical corrections are included to simulate losses within the rotor.

The experimental performance is presented for the turbine in terms of mass flow rate, torque, efficiency, and results of exit surveys. The results obtained from computer programs are shown as plots of gas relative velocities for hub, mean, and shroud streamlines.

SYMBOLS

H' isentropic specific work based on total pressure ratio, ft-lb/lb (J/g)

Δh specific work, Btu/lb (J/g)

N turbine speed, rpm (rad/sec)

N_s specific speed, $NQ^{1/2}/(H')^{3/4}$, $(rpm)(ft^{3/4})/sec^{1/2}$; $(rad)(m^{3/2})(kg^{3/4})/(sec^{3/2})(J^{3/4})$

P static pressure, psia (N/cm^2)

Q volume flow (based on rotor-exit conditions), ft^3/sec (m^3/sec)

Re turbine Reynolds number, $w/\mu r_t$

r radius, ft (m)

U blade velocity, ft/sec (m/sec)

V absolute gas velocity, ft/sec (m/sec)

V_j ideal jet-speed corresponding to total- to static-pressure ratio across turbine, ft/sec (m/sec)

w mass flow, lb/sec (kg/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'_{in}/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 static efficiency (based on inlet-total- to exit-static-pressure ratio)

η_t total efficiency (based on inlet-total- to exit-total-pressure ratio)

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

μ gas viscosity, lb/ft sec (N sec/m^2)

ν blade-jet speed ratio, U_t/V_j

τ torque, in.-lb (N-m)

Subscripts:

cr condition corresponding to Mach number of unity

in turbine inlet station

ex turbine exit station

t rotor tip

w outer wall

Superscripts:

' absolute total state

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

— average

TURBINE DESCRIPTION

A radial-inflow turbine was used in this investigation. The tip diameter of the rotor was 4.59 inches (11.66 cm). The rotor was designed with 11 full blades and 11 splitters. The splitters were partial blades which were identical to the axial part of the full blades. Removal of the splitters was the only alteration made on the turbine on the turbine for this investigation.

Figure 1 shows the rotor, scroll, and bearing housing. The shape of the splitter blades can be seen in this photograph. Figure 2 shows the same rotor after the splitter were removed.

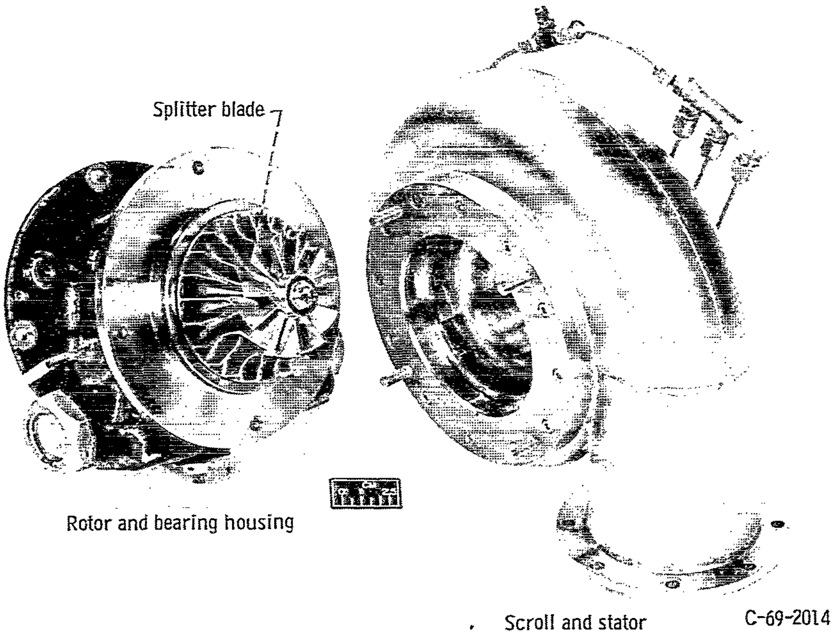

Figure 1. - Turbine with splitter blades.

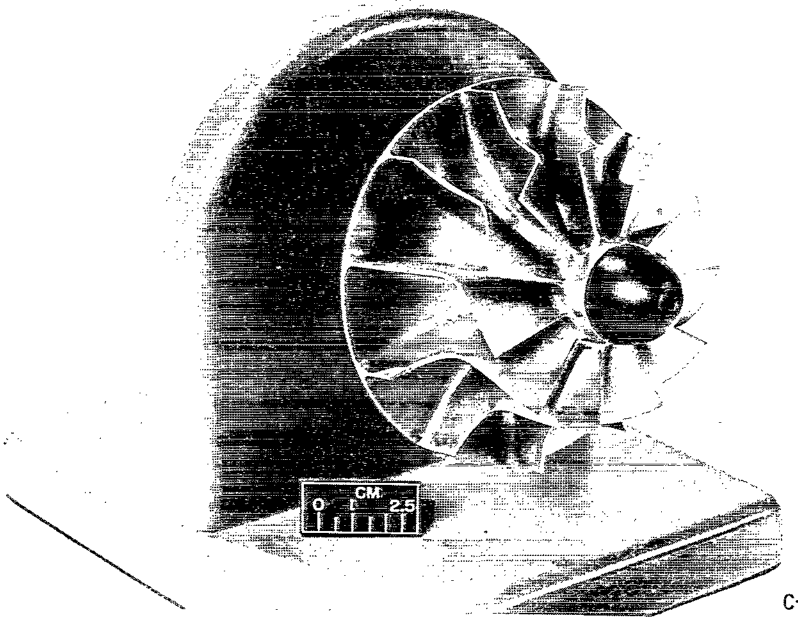

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Figure 2. - Rotor with splitter blades removed.

TABLE I. - AIR EQUIVALENT DESIGN VALUES

Mass flow rate, $w\sqrt{\theta_{cr}}\,\epsilon/\delta$, lb/sec (kg/sec)	0.616 (0.279)
Specific work $\Delta h/\theta_{cr}$, Btu/lb (J/g)	11.9 (27.7)
Speed, $N/\sqrt{\theta_{cr}}$, rpm	29 550
Torque, $\tau\epsilon/\delta$, in.-lb (N-m)	22.12 (2.50)
Total-to-total pressure ratio, P'_{in}/P'_{ex}	1.496
Total-to-static pressure ratio, P'_{in}/P_{ex}	1.54
Blade-jet speed ratio, ν	0.697
Reynolds number, Re	82 200

Air equivalent design values, taken from reference 2, are given in table I. A more complete description of the turbine is contained in reference 2.

APPARATUS AND INSTRUMENTATION

The apparatus and instrumentation used in these tests were essentially the same as those used in the general performance tests described in reference 2. The layout of the test apparatus is shown in the diagram of figure 3. Turbine inlet and exit stations indicate where temperature and pressure measurements were taken. Mass flow rate was

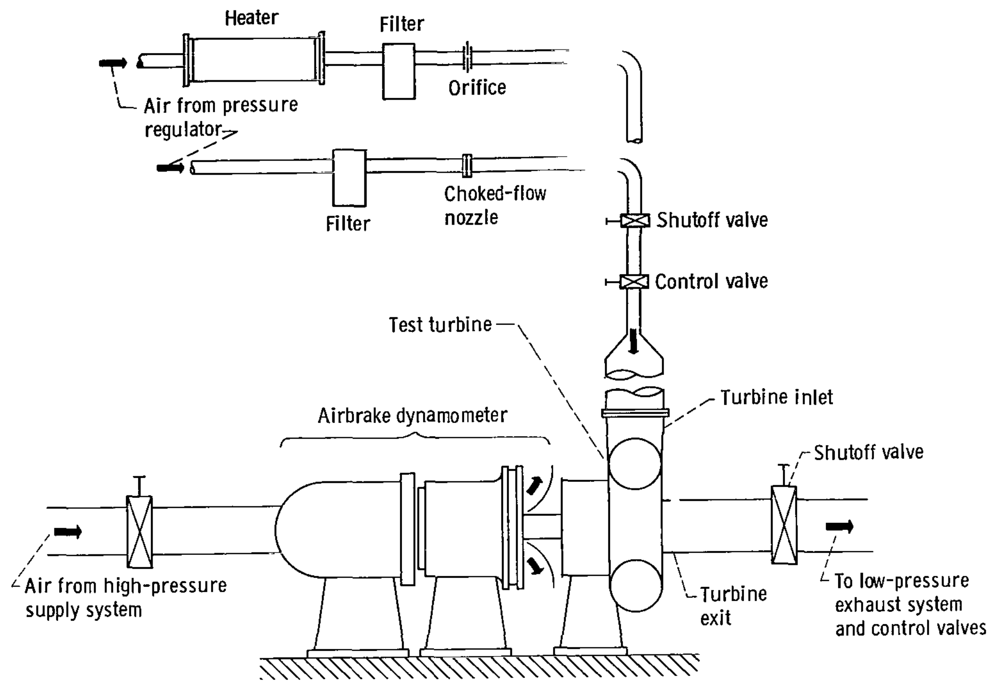


Figure 3. - Test equipment.

measured by means of a choked-flow nozzle or orifice, and torque was measured by means of an airbrake dynamometer. The entire air turbine and torque arm of the dynamometer was supported on gas bearings to reduce friction. The torque arm acted on a strain gage load cell which was read out on a digital voltmeter. Varying the inlet air flow to the dynamometer controlled the speed of the turbine.

Speed was measured by a magnetic pickup which was actuated by a toothed wheel mounted on the coupling which connected the turbine and dynamometer. The speed readout was a digital counter.

For the tests involving the rotor with splitter blades, an inlet air heater was used. An automatic control held the inlet air temperature constant at 540° R (300 K). For the tests made after removal of the splitter blades, the heater was not available, and inlet air temperature increased somewhat during a test run. Speed was set according to the inlet temperature to give the required percent of design equivalent speed.

TEST PROCEDURE

For all tests, the inlet pressure was held constant at 16 psia (11.03 N/cm^2). Total-static pressure ratios were varied from 1.2 to 2.3. Speed was regulated by means of the dynamometer in order to give the required percentages of the design equivalent speed. The percentages used were 0, 20, 40, 60, 80, 100, and 110 for constant speed test runs.

Total temperature and static pressure were measured at the turbine inlet. The mass flow was used with these to calculate the inlet total pressure. Static pressure was measured at the turbine exit. Exit total pressure for calculating total efficiency was calculated from exit static pressure, flow rate, temperature, exit flow angle, and area. A survey was made at the turbine exit to determine the variation with radius in exit flow angle and total pressure both with and without splitter blades. The surveys were made at design equivalent speed and pressure ratio.

For calculating efficiencies, the calculated exit total pressure at the exit was used instead of the measured total pressure.

All calculations made for the purpose of processing data were made as described in reference 2. This reference also gives details of the testing procedure used.

THEORETICAL VELOCITY CALCULATION

Flow velocities within the rotor were studied by the use of three computer programs previously developed at Lewis by Katsanis and McNally. Descriptions of these programs were published in references 5 to 7. A brief description will be given in this section and velocity plots obtained by use of the programs are presented in the section RESULTS AND DISCUSSION.

The first program used was one described in reference 5. With this program, a meridional plane solution is first made. Isentropic conditions are used but a correction is made to represent the loss in total pressure relative to the blade. Having calculated the conditions on the meridional plane, the program then calculates velocities at the surfaces of the blades by an equation based on absolute irrotational flow and linear velocity distribution between blades. Part of the output of this program is the streamline geometry in the meridional plane which is used as input for the other two programs.

The second program used was called TANDEM, and is described in reference 6. The program makes a blade-to-blade solution to the flow problem and was written primarily for tandem-blade turbomachines. However, it may be used for radial-inflow turbines with splitters. Reversible conditions are used, but a correction is applied to the stream sheet thickness to represent the effect of total pressure loss. The program makes use of a stream function to calculate the surface velocities on a surface of revolution defined by a streamline. The streamline geometry is an input to this program and was taken from the output of the first program. Three streamlines were used for this report: the hub, shroud, and the 50-percent streamline.

The third program used was TURBLE, described in reference 7. This program is a modification of the TANDEM program. For present purposes, it represents the case of a radial-inflow turbine with the splitter blades removed. The input to this program is the same as for the TANDEM program except for a few items which reflect the absence of splitter blades.

RESULTS AND DISCUSSION

In this section, the test results are presented first. This is followed by a discussion of the velocity distributions obtained from the computer programs. Finally, the calculated velocity distributions are used to give an interpretation of the results obtained experimentally.

Test Results

Test results are presented as plots involving flow rate, torque, efficiency, and rotor-exit surveys. Comparison is made, in each case, between the performance with splitters and without splitters.

Mass flow rates. - Figures 4(a) and (b) show mass flow rates for the turbine with and without splitter blades. Mass flow rates are plotted against the ratio of exit static pressure to inlet total pressure for lines of constant speed. Differences for the two cases are small with the best agreement at speeds higher than 60 percent of design. At design equivalent speed and pressure ratio the mass flow is the same. The greatest differences for the two cases occur in the 40- to 60-percent design speed range.

A crossplot of figures 4(a) and (b) is shown in figure 4(c) with the flow rate scale expanded. This figure shows the manner in which the mass flow rate varies with turbine speed at design equivalent pressure ratio. The turbine with splitters shows about a 1-percent-higher mass flow rate in the 40- to 60-percent speed range. The variation is similar for other pressure ratios.

Torque. - Torque-speed curves are shown in figures 5(a) and (b) for the turbine with and without splitter blades. At design equivalent speed and pressure ratio, the torque is about 1-percent higher for the turbine without splitters. As with mass flow rate, differences in torque with and without splitter blades are small. Figure 5(c) is a comparison of equivalent torque as a function of total to static pressure ratio for the 100- and 60-percent speed lines. The 60-percent speed line was selected because this is the speed at which the greatest differences appear. At 60 percent of design equivalent speed the torque is about 2 percent higher for the turbine with splitters.

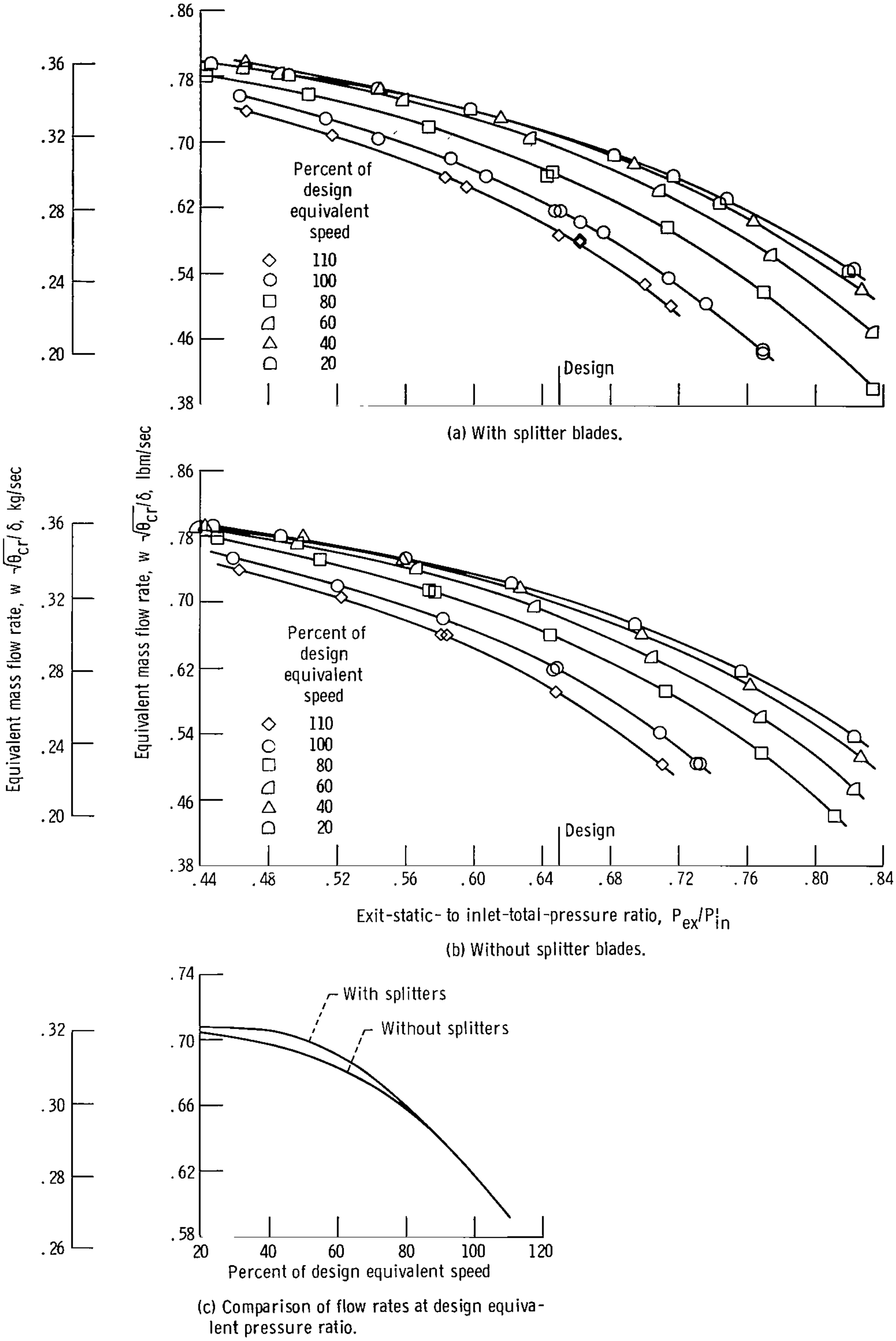
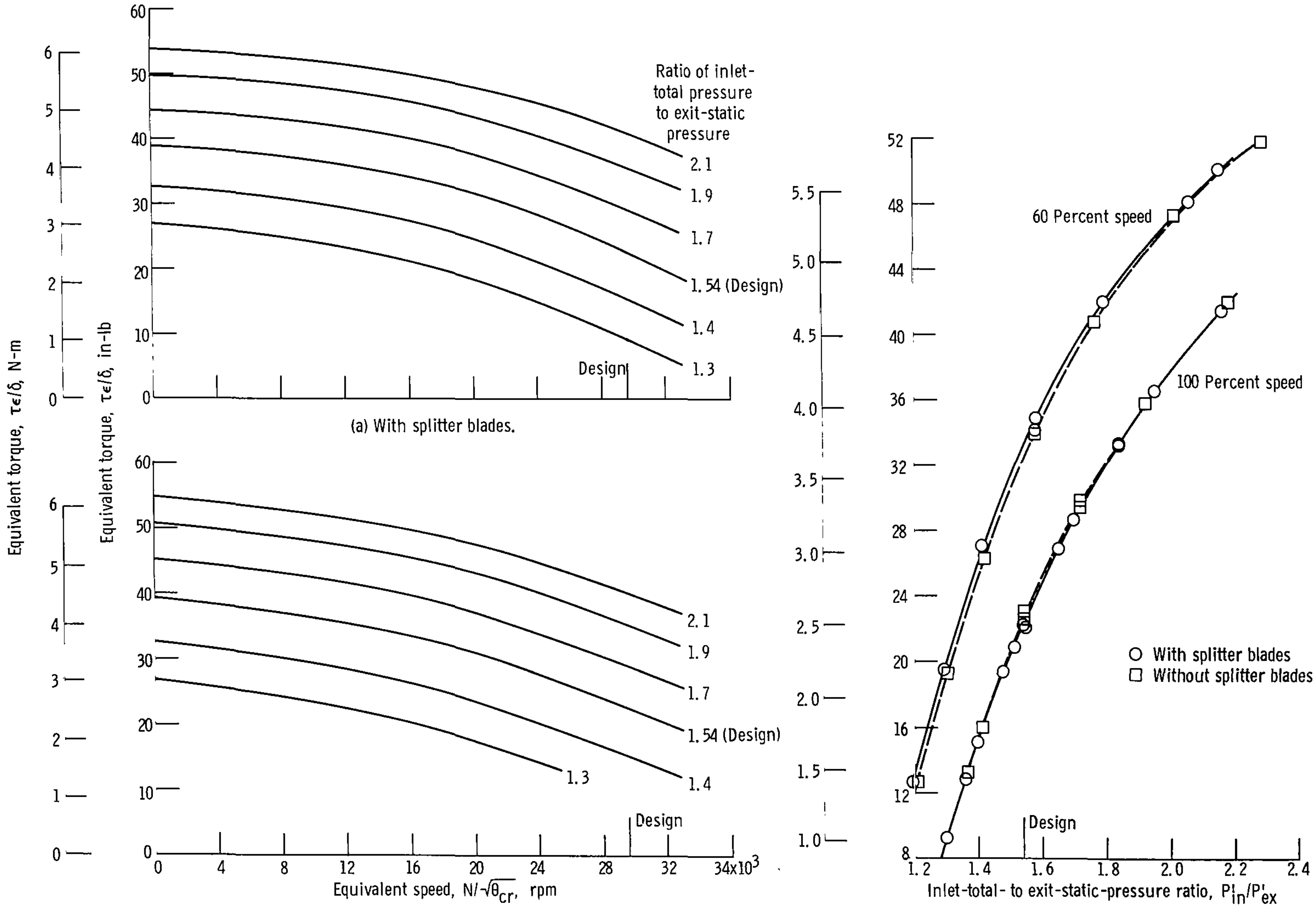


Figure 4. - Mass flow rate.

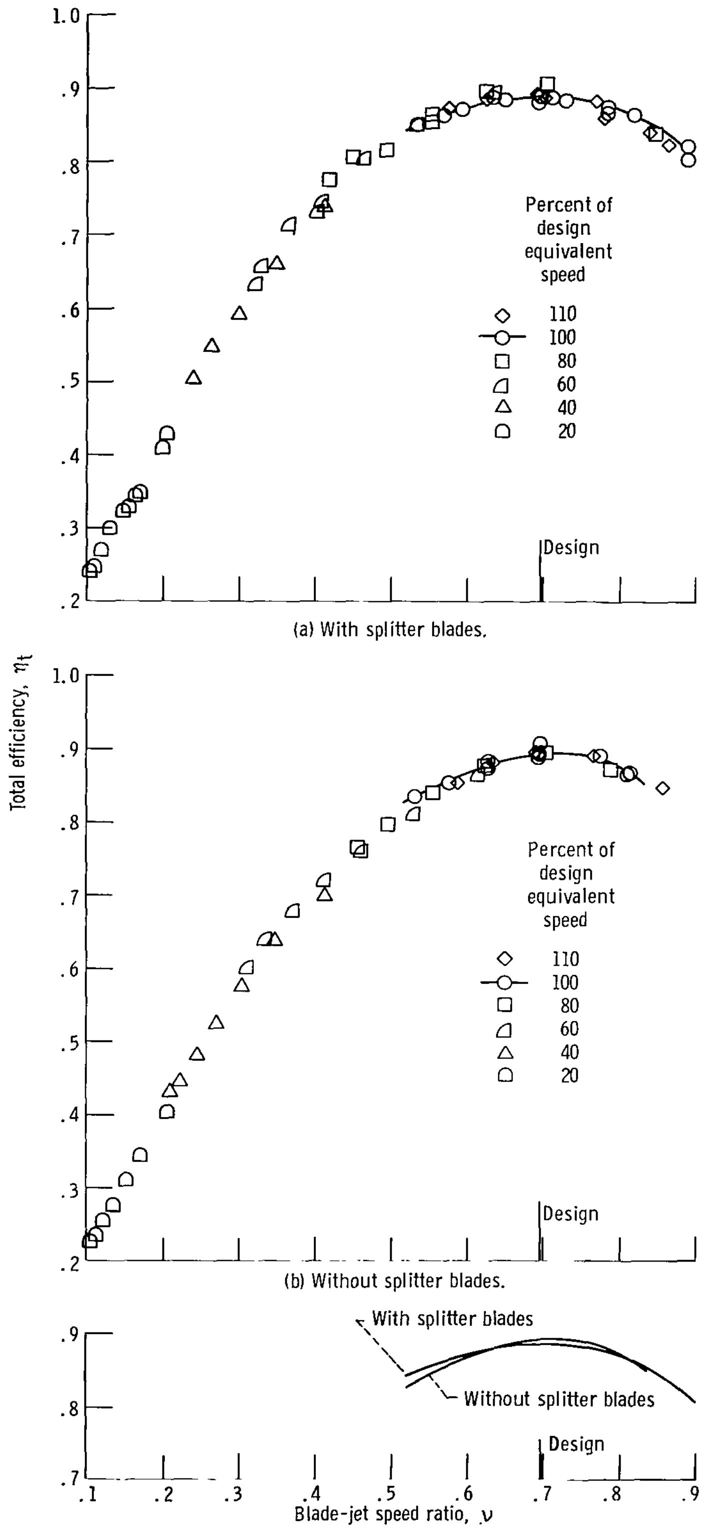


(a) With splitter blades.

(b) Without splitter blades.

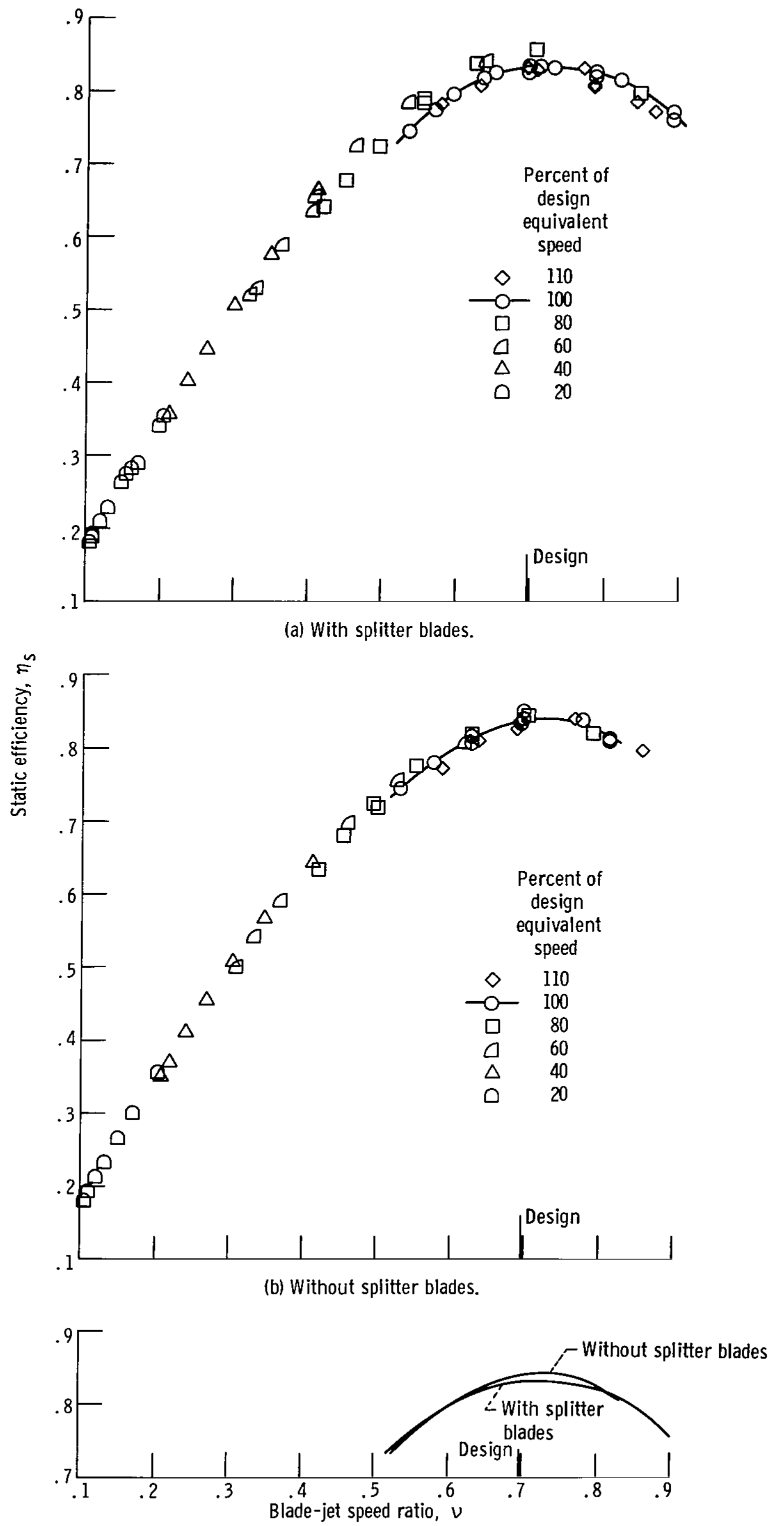
(c) Comparison of equivalent torque for 60 and 100 percent of design equivalent speed.

Figure 5. - Torque speed curves.



(c) Comparison at design equivalent speed, with and without splitter blades.

Figure 6. - Turbine total efficiency.



(a) With splitter blades.

(b) Without splitter blades.

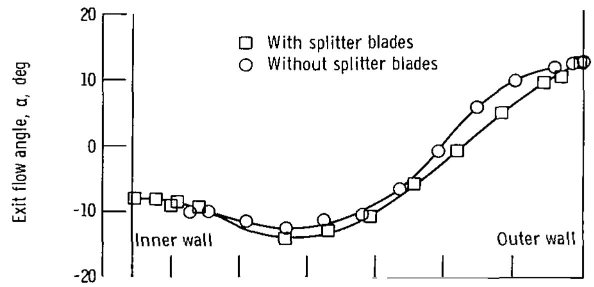
(c) Comparison at design equivalent speed, with and without splitter blades.

Figure 7. - Turbine static efficiency.

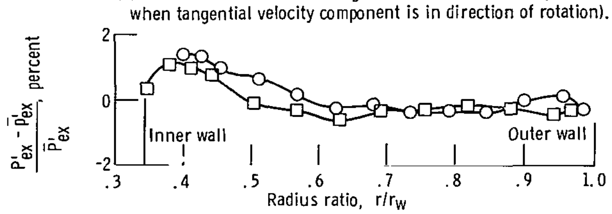
Efficiency. - The experimental efficiencies are presented in figures 6 and 7 against blade-jet speed ratio. Figures 6(a) and (b) show the total efficiency plots for the turbine with and without splitter blades. At lower speeds, the total efficiency is slightly higher with splitters than without. However, above 80 percent of design equivalent speed, the efficiency is slightly higher without splitters. Figure 6(c) shows a comparison of total efficiency for the two cases using only the design equivalent speed curves. The total efficiency is about 0.6 point higher without splitters at design blade-jet speed ratio.

A similar set of plots for static efficiency is shown in figures 7(a) and (b). No noticeable difference in static efficiency is found at low speeds. However, at design equivalent speed and blade-jet speed ratio, the static efficiency is about 0.7 point higher without splitters. A comparison at design equivalent speed is shown in figure 7(c).

The efficiency differences observed are small, less than one point. Therefore, the question arises as to whether these differences are real or simply a result of the experimental accuracy of the tests. Reference was made to a report by Holeski and Futral (ref. 8). This report included an error analysis based on a turbine geometrically similar to the one used for the present investigation. The error analysis indicates that the probable error in a single data point would be about 0.9 point for efficiency. For a faired curve, such as in figures 6(c) and 7(c), the actual experimental error would be less than the probable error for a single point. Therefore, it is concluded that the efficiency differences shown in figures 6(c) and 7(c) are probably real, rather than the result of experimental error. However the differences are so small that the performance must be considered essentially unchanged by removing the splitters.



(a) Variation in turbine exit angle across channel (α is positive when tangential velocity component is in direction of rotation).



(b) Variation in total pressure across channel. Pressure measurements taken from total pressure tap on angle probe.

Figure 8. - Turbine exit survey results at design equivalent speed and pressure ratio.

Rotor exit surveys. - The radial variation in the flow angle at the turbine exit is shown in figure 8(a). The turning was slightly less with the splitters removed, with a maximum difference of about 4°. Figure 8(b) is a plot of total pressure variation across the channel. The exit total pressures for this plot were taken from the total pressure tap on the angle probe. The total pressure variation for the two cases is similar. However, points near the inner wall, for the case without splitters, are missing as a result of a different probe design.

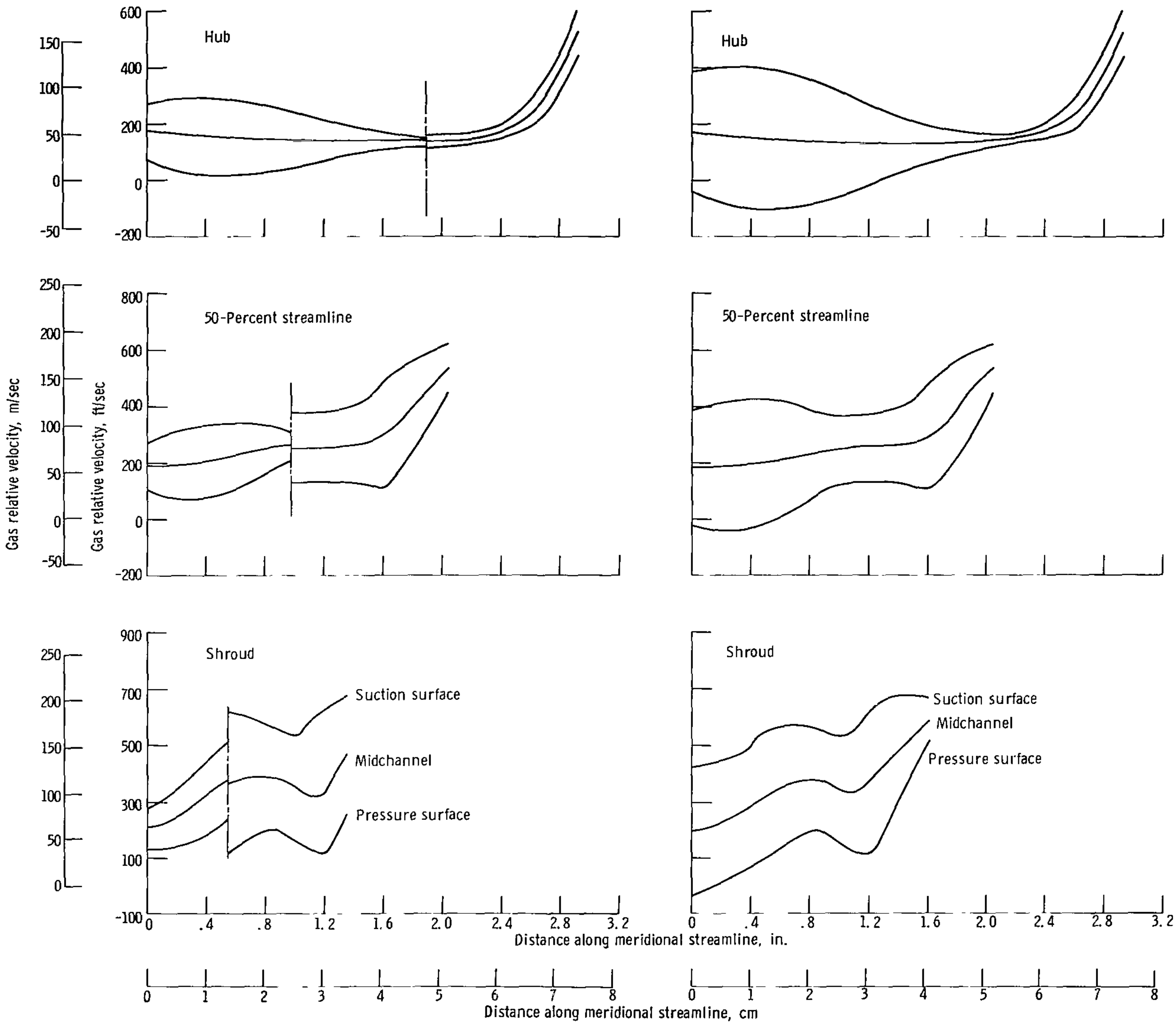
Theoretical Velocity Distribution

The computer programs were used for the purpose of seeking explanations for the experimental results. It was also thought that the computer results might indicate ways of futher improving the performance of turbines of this type.

The first computer program used was the meridional plane solution described in reference 5. The velocity plots from this program are shown in figure 9. Figure 9(a) is for the turbine with splitter blades, and figure 9(b) is for the turbine without splitter blades. The discontinuities in the plots of figure 9(a) at the end of the splitters are caused by change in blade spacing, which results in changes in the loading and the blockage. Figure 9(b) shows a negative surface velocity near the rotor entrance at the pressure surface hub, mean, and shroud. This indicates the existence of an eddy in the flow brought about by the removal of the splitter blades. Although the flow is treated as isentropic in the program, this would be expected to reduce the efficiency for a real fluid. A decrease in the flow rate could also be expected.

In order to get a better approximation to the surface velocities, use was made of the potential flow solutions of the TANDEN and TURBLE programs, which are described in references 6 and 7. The resulting surface velocity plots are shown in figures 10(a) and (b). Figure 10(b) also indicates that an eddy occurs near the leading edge when splitters are removed. However, this is confined to the region of the blades between the hub and the 50-percent streamline.

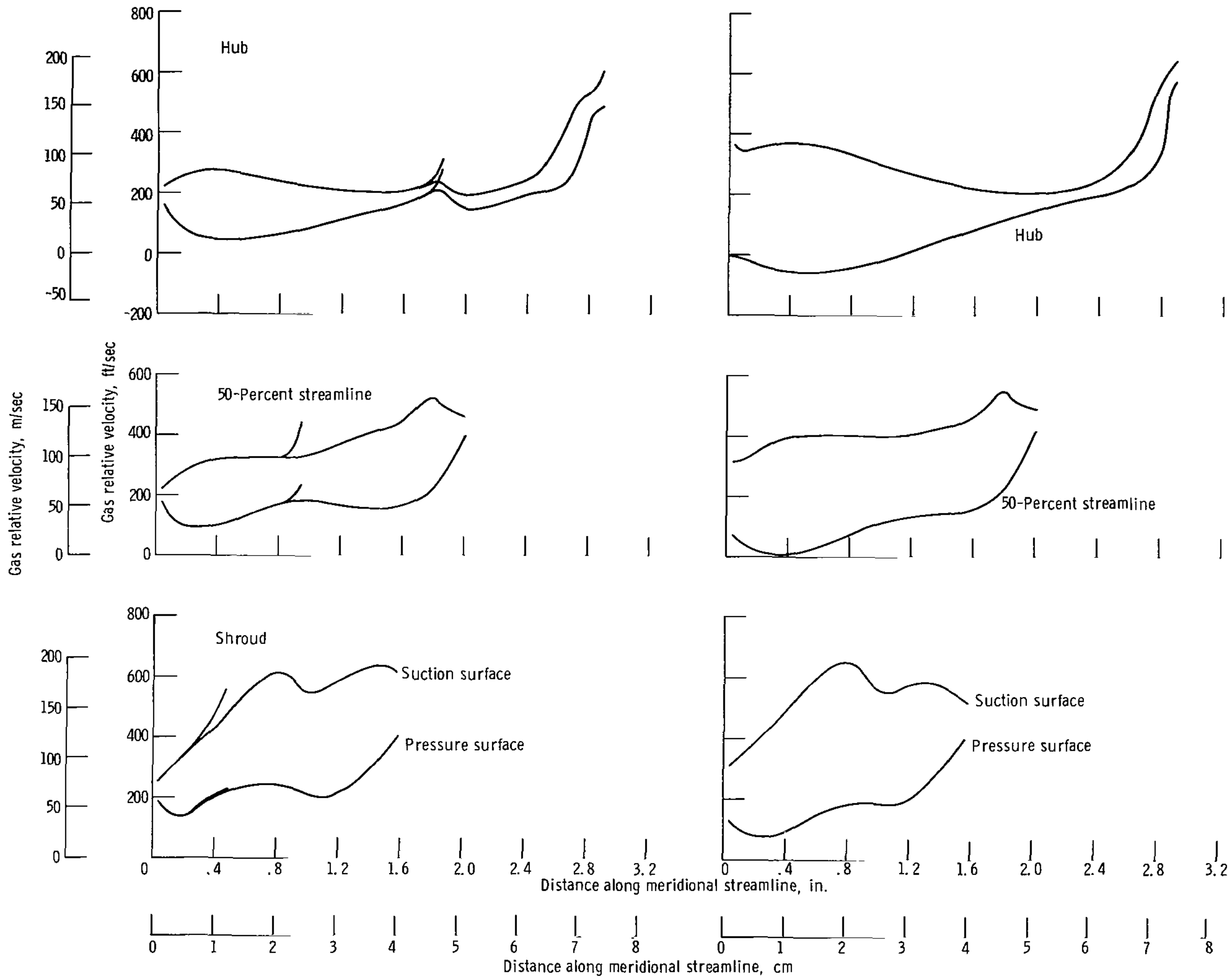
In figure 10(a) the surface velocities over the splitter blades match the surface velocities over the full blades near the leading edge. However, near the end of the splitters, the velocities at the splitter surfaces rise rapidly since in this region the streamlines near the surface of the blade are forced into an axial direction by the shape of the blade.



(a) Rotor with splitter blades.

(b) Rotor without splitter blades.

Figure 9. - Gas relative velocities at blade surface by program of reference 5.



(a) Rotor with splitter blades. Calculations by TANDEM program (ref. 6).

(b) Rotor without splitter blades. Calculations by TURBLE program (ref. 7).

Figure 10. - Gas relative velocities at blade surface.

Discussion of Results

The experimental results indicate that a slight increase in efficiency occurred at equivalent design speed and pressure ratio when the splitter blades were removed. The computer results indicated flow eddies which should result in a decrease in efficiency. In addition, removal of the splitter blades reduced the number of rotor blades by half at the rotor entrance. Since the optimum incidence angle depends on the number of blades at the rotor entrance, it follows that the turbine without splitter blades runs with an incidence angle which is not optimum. However, the removal of splitter blades also represents an appreciable reduction in the wetted area of the blade surface and eliminates the mixing loss at the splitter ends. Thus, in the case of the turbine used here, the reduction in viscous and mixing losses have slightly more than compensated for the losses incurred by eddies and nonoptimum incidence angle when the splitter blades were removed. Although the gain was small, it suggests that, if the optimum incidence angle were used and the flow eddy were eliminated, the efficiency of a turbine without splitters might be improved. The computer results also indicate that, for a turbine with splitters, a redesign of the splitters near their trailing edges would be desirable to reduce the rapid accelerations in that region, as indicated in figure 10(a).

For purposes of comparison, an examination was made of a method (see ref. 9) for determining the optimum number of blades for a radial-inflow rotor. In this reference the optimum number of blades is the minimum number required to prevent local flow reversals within the rotor passages. For the turbine used in this investigation the minimum number of blades is 22, which is the number the rotor had before the splitters were removed. Examination of figure 10(a) indicates that the rotor, as originally designed, probably could have had fewer blades without any flow reversals.

A general observation may be made as a result of conclusions reached here and those of reference 10. This reference deals with the effects of varying the blade-shroud clearance for a radial-inflow turbine. It was found that the turbine performance was not affected strongly by clearance increases at the rotor entrance so long as tight clearance was maintained at the rotor exit. Combining this result with the present results suggests that it may be possible to take considerable liberties with the blade design near the rotor entrance without seriously lowering the efficiency. In particular, where blade cooling is required as a result of high turbine inlet temperature, the splitters could be eliminated. This would leave space for thickened full blades containing cooling passages. With the correct blade design, this could also eliminate eddies in the flow at the blade surfaces.

SUMMARY OF RESULTS

An investigation was conducted to determine the effect on performance of removing the splitter blades from a 4.59-inch (11.66-cm) radial-inflow turbine. The same rotor was used with and without splitter blades. Three Lewis computer programs were used as an aid in interpreting results. The following results were obtained:

1. A slight increase in efficiency, at design equivalent speed and pressure ratio resulted from removal of the splitter blades. Results indicate that the total efficiency increased by 0.6 point and that the static efficiency increased by 0.7 point.

2. No significant change was noted in mass flow rate at design equivalent speed and pressure ratio as a result of removing splitter blades.

3. Removal of the splitter blades caused the turbine to operate at a nonoptimum incidence angle, with the formation of flow eddies near the blade leading edges. The absence of any significant decrease in efficiency was attributed to the reduction of wetted area and the elimination of mixing losses with removal of the splitters.

Lewis Research Center,
National Aeronautics and Space Administration,
Cleveland, Ohio, September 10, 1970,
120-27.

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