Printed in U.S.A.

73-GT-29

an ASME publication

$3.00 PER COPY
$1.00 TO ASME MEMBERS

Aerodynamic Study of a Turbine Designed For a Small Low-Cost Turbofan Engine

M. G. KOFSKEY

W. J. NUSBAUM

Lewis Research Center,
NASA,
Cleveland, Ohio

A 20.32 centimeter (8.00 in.) mean diameter two-stage turbine was experimentally investigated over a range of speeds from 0 to 110 percent of equivalent design speed and over a range of pressure ratios from 2.2 to 4.2. The principal results indicated that the performance level was substantially higher than that assumed in the design. As part of the program to reduce manufacturing costs, the first stage blading was reduced in thickness for ease in coining. Tests of the modified blades indicated that the aerodynamic performance of a stator or rotor blade with a large amount of reaction was affected very little by a significant change of the pressure surface.

Contributed by the Gas Turbine Division of The American Society of Mechanical Engineers for presentation at the Gas Turbine Conference and Products Show, Washington, D.C., April 8–12, 1973. Manuscript received at ASME Headquarters December 19, 1972.

Copies will be available until February 1, 1974.

THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS, UNITED ENGINEERING CENTER, 345 EAST 47th STREET, NEW YORK, N.Y. 10017

Aerodynamic Study of a Turbine Designed For a Small Low-Cost Turbofan Engine

M. G. KOFSKEY **W. J. NUSBAUM**

INTRODUCTION

The NASA is currently studying the gas turbine engine as a replacement for the piston engine in light aircraft. A comparison of these two types of engine shows the gas turbine engine to be very attractive because of its smaller size and weight. However, as shown in reference (1),[1] the high production cost of current gas turbine engines greatly restricts their use.

Studies have been made of methods for

[1] Numbers in parentheses designate References at end of paper.

NOMENCLATURE

A = area, sq cm; sq in.

D_p = pressure-surface diffusion parameter, $\frac{(\text{Blade inlet relative velocity})^2}{(\text{Minimum blade surface relative velocity})^2}$

D_s = suction-surface diffusion parameter, $\frac{(\text{Maximum blade surface relative velocity})^2}{(\text{Blade outlet relative velocity})^2}$

g = dimensional constant, SI = 1.0; 32.174 ft/sec^2

Δh = specific work, J/g; Btu/lb

J = mechanical equivalent of heat, 778.2 ft-lb/Btu

N = turbine speed, rpm

p = pressure, N/cm^2 abs; psia

R_x = reaction, $\frac{(\text{Blade outlet velocity})^2-(\text{Blade inlet velocity})^2}{(\text{Blade outlet velocity})^2}$

T = absolute temperature, K; deg R

U = blade velocity, m/sec; fps

V = absolute gas velocity, m/sec; fps

V_j = ideal jet speed corresponding to total- to static-pressure ratio across turbine, m/sec; fps

W = relative gas velocity, m/sec; fps

w = mass flow, kg/sec; lb/sec

γ = ratio of specific heats

δ = ratio of inlet total pressure to United States 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 = static efficiency (based on inlet total- to exit-static pressure ratio)

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

θ_{cr} = squared ratio of critical velocity at turbine inlet to critical velocity at United States standard sea-level air, $(V_{cr}/V_{cr}^*)^2$

λ = work-speed parameter, $U_m^2/gJ\,\Delta h$

ν = blade-jet speed ratio, U_m/V_j

τ = torque, N-m; ft-lb

ω = turbine speed, rad/sec

Subscripts

cr = condition corresponding to Mach number of unity

eq = equivalent

m = mean radius

u = tangential component

1 = station at turbine inlet (Fig. 2)

2 = station at first-stage stator exit (Fig. 2)

3 = station at first-stage rotor exit (Fig. 2)

4 = station at second-stage stator exit (Fig. 2)

5 = station at second-stage rotor exit (Fig. 2)

Superscripts

' = absolute total state

* = U. S. standard sea-level conditions [temperature equal to 288.15 K (518.67 R), pressure equal to 10.13 N/sq cm (14.70 psia)]

Table 1 Turbine Design Requirements

	First-stage	Two-stage
For operation with fuel mixture: hydrogen/carbon = 0.167		
Inlet total temperature, T_1', K; ^oR	977.78; 1760	977.78; 1760
Inlet total pressure, p_1', N/cm^2 abs; psia	28.544; 41.40	28.544; 41.40
Mass flow, w, kg/sec; lb/sec	2.994;6.60	2.994;6.60
Turbine rotative speed, N, rpm	28 000	28 000
Rotor blade speed (mean section), m/sec; ft/sec	297.91; 977.38	297.91; 977.38
Total- to static-pressure ratio, $p_1'/p_{3 \text{ or } 5}$	2.238	4.312
Total- to total-pressure ratio, $p_1'/p_{3 \text{ or } 5}'$	1.975	3.551
Blade-jet speed ratio, ν	0.466	0.360
Work-speed parameter, λ	0.582	0.331
Total to static efficiency, η_s	0.747	0.780
Total to total efficiency, η_t	0.870	0.880
Specific work, Δh, joules/gram; Btu/lb	152.77; 65.63	268.02; 115.14
Air equivalent		
Mass flow, $\epsilon w\sqrt{\theta_{cr}}/\delta$, kg/sec; lb/sec	1.989; 4.385	1.989; 4.385
Specific work, $\Delta h/\theta_{cr}$, joules/gram; Btu/lb	45.834; 19.690	80.407; 34.542
Torque, $\tau\epsilon/\delta$ Newton-meter; ft-lb	56.727; 41.840	99.521; 73.403
Rotative speed, $N/\sqrt{\theta_{cr}}$, rpm	15 336	15 336
Rotor blade speed (mean section), m/sec; ft/sec	163.17; 535.33	163.17; 535.33
Total- to static-pressure ratio, $(p_1/p_{3 \text{ or } 5})_{eq}$	2.298	4.640
Total- to total-pressure ratio, $(p_1'/p_{3 \text{ or } 5}')_{eq}$	2.018	3.765
Blade-jet speed ratio, ν	0.466	0.360
Work-speed parameter, λ	0.582	0.331

achieving a sizable reduction in manufacturing costs of these engines (1). One approach to the problem involved the trade-off of engine performance for values of design parameters which are conducive to a low cost engine. This approach limits the design operating temperature to a level compatible with low cost materials. The pressure ratio is limited to a value which will permit a reduction in tip speeds, stress levels, and the number of stages. A second approach used in the study was concerned with design simplifications and low cost fabrication techniques. A detailed investigation was made of techniques for fabrication of low cost compressor and turbine rotors. The results of this phase of the study indicated that a stamping process, using coined blade profiles, might have the best potential for low cost and reliability.

A detailed design study (1) was made of a low cost turbofan engine incorporating a low temperature, low pressure ratio cycle. This engine is designed to operate at cruise conditions of Mach 0.65 at an altitude of 7620 m (25,000 ft). It has a cruise turbine inlet temperature of 980 K (1760 R), a pressure ratio of 6, and a bypass ratio of 2.5. These conditions would give 4890 Newtons (1100 lb) of takeoff thrust. The turbine for this engine would operate at an inlet total pressure of 28.5 N/sq cm abs (41.4 psia), an inlet total- to exit-static pressure ratio of 4.3 and a rotative speed of 28,000 rpm.

As part of the overall cost-reduction program, the turbine component suitable for this application was designed and fabricated. It was a two-stage, 20.32-cm (8.00-in.) mean diameter, axial flow turbine. The design values permitted operation at moderate temperature levels and pressure ratios. As a result, there was a reduction in rotor blade speed, stress levels, and number of stages from those used in present-day engines. Use of moderate temperatures could facilitate the use of low-cost fabrication techniques as well as low-cost materials. The subject turbine was fabricated with conventional blade profiles and machining methods. An experimental, cold air investigation was made to determine the aerodynamic performance of both first-stage and two-stage configurations. The results of this investigation are reported in reference (2).

In the second phase of the turbine component study, fabrication cost-reduction techniques were investigated. The first-stage conventional blading was modified to achieve an airfoil shape easily fabricated by a stamping or coining process. This modification involved a change of the

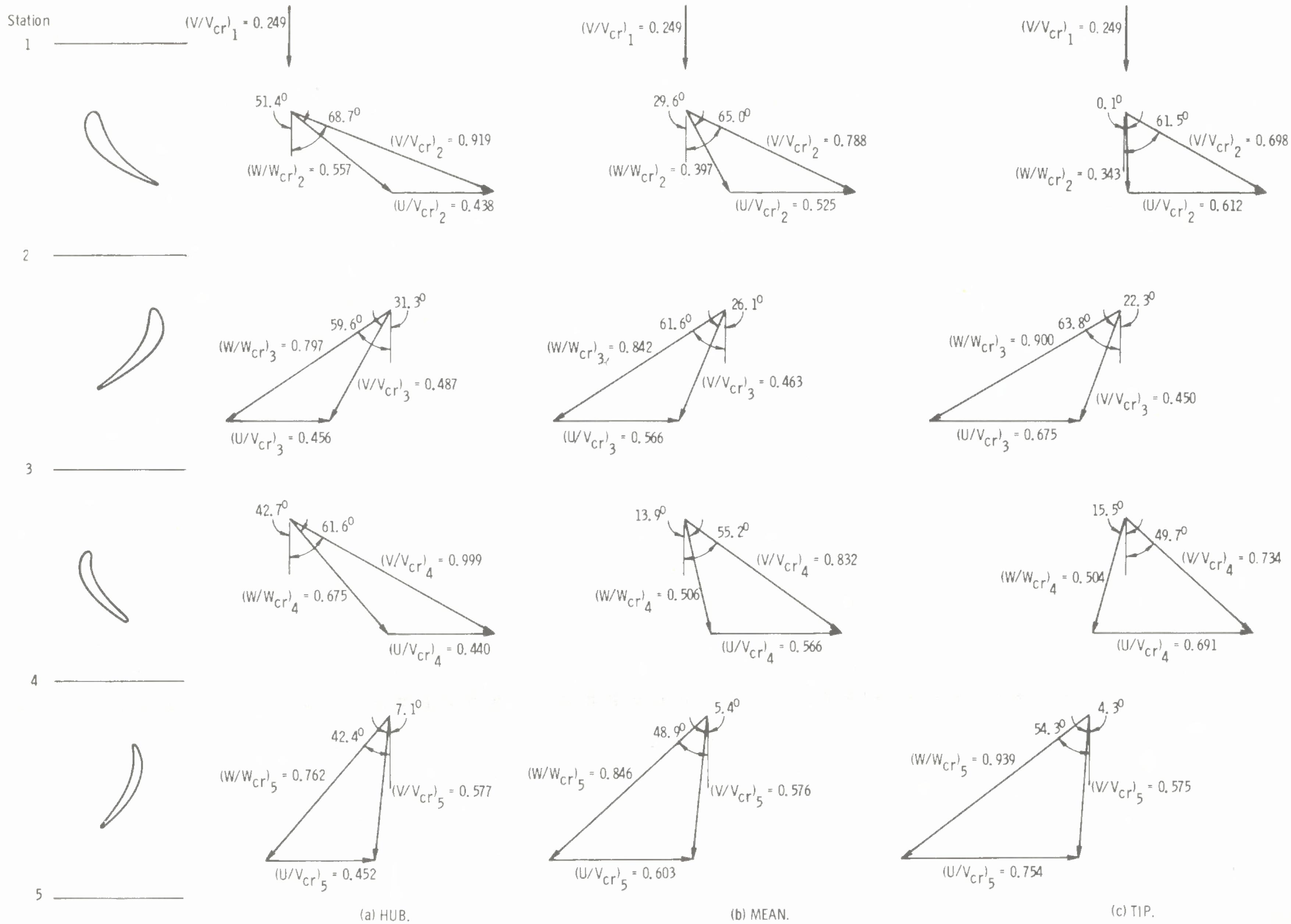

Fig. 1 Design free-stream velocity diagrams

Table 2 Turbine Aerodynamic Parameters

Turbine blading	Blade row	Section	Blade-surface diffusion parameter (a)		Blade turning, deg	Blade chord, cm; in.	Solidity	Reaction (a)	Aspect ratio (b)	Number of blades	Tip clearance, cm; in.
			Suction surface, D_s	Pressure surface, D_p							
Original profile	Stator	Tip	1.232	1.241	68.7	2.946; 1.160	1.39	0.873	1.29	35	------------
		Mean	1.233	1.309	65.0	2.616; 1.030	1.43	.900			
		Hub	1.284	1.032	61.5	2.400; 0.945	1.35	.927			
	Rotor	Tip	1.302	1.892	63.9	2.654; 1.045	1.50	0.855	1.39	42	0.030; 0.012
		Mean	1.180	2.176	91.2	2.606; 1.026	1.71	.778			
		Hub	1.213	2.267	111.0	2.758; 1.086	2.18	.512			
Thin profile	Stator	Tip	1.232	5.635	68.7	2.946; 1.160	1.39	0.873	1.29	35	------------
		Mean	1.233	5.531	65.0	2.616; 1.030	1.43	.900			
		Hub	1.284	5.742	61.5	2.400; 0.945	1.35	.927			
	Rotor	Tip	1.302	4.223	63.9	2.654; 1.045	1.50	0.855	1.39	42	0.025; 0.010
		Mean	1.180	6.108	91.2	2.606; 1.026	1.71	.778			
		Hub	1.213	8.675	111.0	2.758; 1.086	2.18	.512			

[a] See section SYMBOLS for definition.

[b] Based on average blade height and mean chord.

blade pressure surface to give near uniform blade thickness from leading to trailing edge. This change resulted in thin blades with the maximum thickness equal to the leading edge diameter. Both the suction surface and the throat remained unchanged. The first stage was tested with the modified stator and rotor blades in order to determine their effect on turbine performance. Results of this investigation are reported in reference (3).

The objectives of this paper are twofold. The first objective is to review the principal results obtained from the experimental study of the reference two-stage turbine. The second objective is to show the effects on first-stage performance of reducing the stator and rotor blade thickness in the manner described in the foregoing. These effects are shown by a comparison of first-stage performance results as obtained with the design blades (2) and the thin blades (3).

DESCRIPTION OF TURBINE

As stated in the INTRODUCTION, the 20.32-cm (8.0-in.) mean diameter two-stage turbine was designed as a component for a low-cost turbofan engine. Table 1 presents the pertinent design requirements for this turbine. An overall work-speed parameter, λ, of 0.331 indicates a conservative design which should result in good aerodynamic performance. A turbine-inlet temperature of 977.78 K (1760 R) and a rotor-blade speed (mean section) of 297.91 m/sec (977.38 fps) should not create any serious stress problems.

Design Velocity Diagrams

The free-stream velocity diagrams were computed to meet the design requirements and are based on the following assumptions:

1 Free vortex flow
2 A 57 to 43 percent work split between the first and second stages, giving respective values of work-speed parameter, λ, of 0.582 and 0.771
3 Respective losses in total pressure through the first- and second-stage stators of 4.0 and 5.0 percent of stage inlet pressure
4 First-, second-, and two-stage total efficiency values of 0.870, 0.871, and 0.880, respectively.

The free-stream velocity diagrams and station nomenclature are shown in Fig. 1. The amount of turning and the magnitude of velocities throughout the turbine indicate a conservative design. The turning at the mean diameter is 91.2 and 62.8 deg for the first- and second-stage rotors, respectively. All free-stream velocities are subsonic, with near sonic conditions existing at the exit of both stators at the hub. There is an increase in relative velocity through both rotors, indicating some positive reaction that increases from hub to tip. Values of reaction are presented in Table 2 along with other aero-

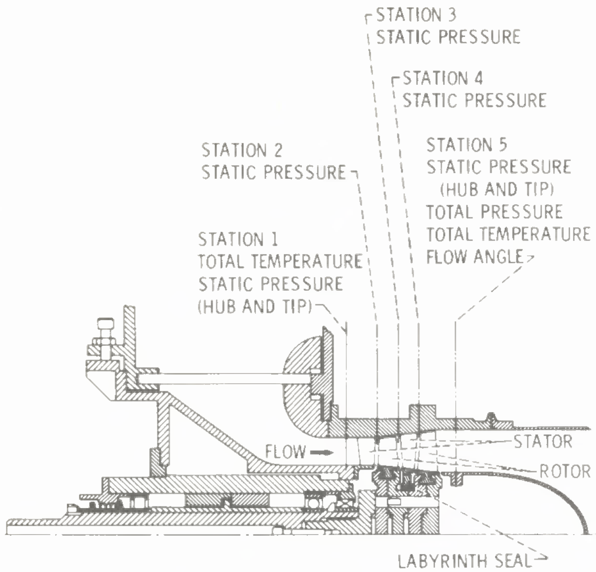

Fig. 2 Schematic of turbine

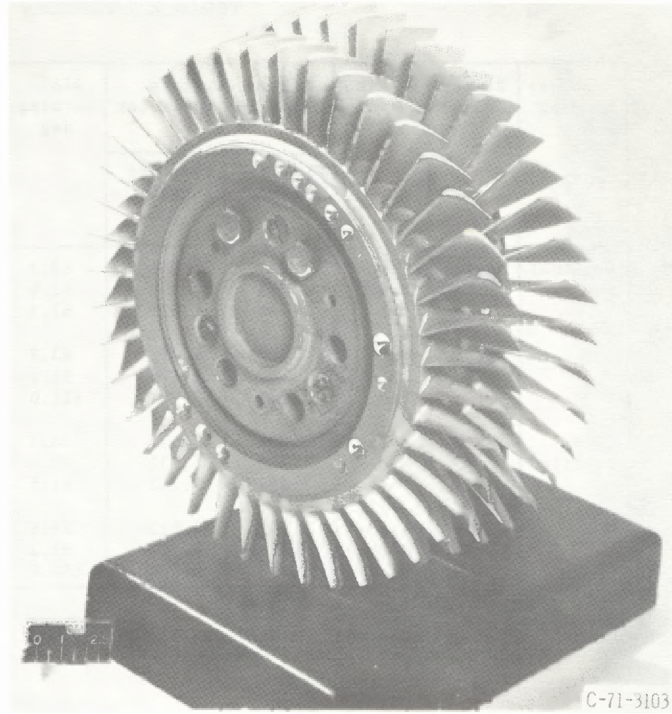

Fig. 3 Turbine rotors

dynamic parameters.

Blade Description

The selection of blade number was based on the results of optimum solidity studies (4) and by the axial space allocated to the turbine by the engine design. The first- and second-stage stator assemblies contain 35 and 43 blades, respectively, with mean section blade chords of 2.616 and 2.182 cm (1.030 and 0.859 in.). Values of solidity at the mean sections are 1.43 and 1.47. The first-stage stator blade row has a constant blade height of 3.363 cm (1.324 in.); the second-stage stator blade height increases from 3.945 cm (1.553 in.) at the inlet to 4.483 cm (1.765 in.) at the exit. Both first- and second-stage blade rows have a constant mean diameter of 20.32 cm (8.0 in.). The schematic of the turbine (Fig. 2) shows the flow passage through the turbine with the blading arrangement. The rotor assembly shown in Fig. 3 includes 42 and 44 blades in the first and second stages, respectively, giving values of solidity at the mean section of 1.71 and 1.66. The inner and outer walls of the flow passage diverge equally at both blade rows (Fig. 2) giving average rotor blade heights of 3.658 and 4.801 cm (1.44 and 1.89 in.). The tip clearance for the first-stage rotor is 0.030 cm (0.012 in.). The 0.038-cm (0.015 in.) tip clearance for the second-stage rotor is provided by a recess in the outer wall, thereby reducing leakage flow. These values of tip clearance are 0.8 percent of the respective annulus heights.

TEST FACILITY

The apparatus used in the performance evaluation of the subject turbine is shown in Fig. 4. In addition to the turbine, the apparatus consisted of a cradled dynamometer to absorb the power output of the turbine while controlling the speed and an inlet and exhaust piping system with flow controls. High-pressure dry air was supplied from the laboratory air system. The air passed through a filter, a weight flow measuring station (a flat-plate orifice), a remotely controlled pressure-regulating valve, an inlet plenum and into the turbine. After passing through the turbine, the air was exhausted through a system of piping and a remotely operated valve into the laboratory exhaust system. The pressure control valve upstream from the turbine regulated the turbine-inlet pressure. With a given inlet pressure, the remotely controlled valve in the low-pressure exhaust line was used to maintain the desired pressure ratio across the turbine. Overall performance of the two-stage turbine was based on measurements of speed, torque, and mass flow together with inlet and exit flow conditions. These included temperature and pressure data at station 1 (Fig. 2), with pressure and flow angle at station 5. The tip static pressure variation through the turbine, as reported in reference

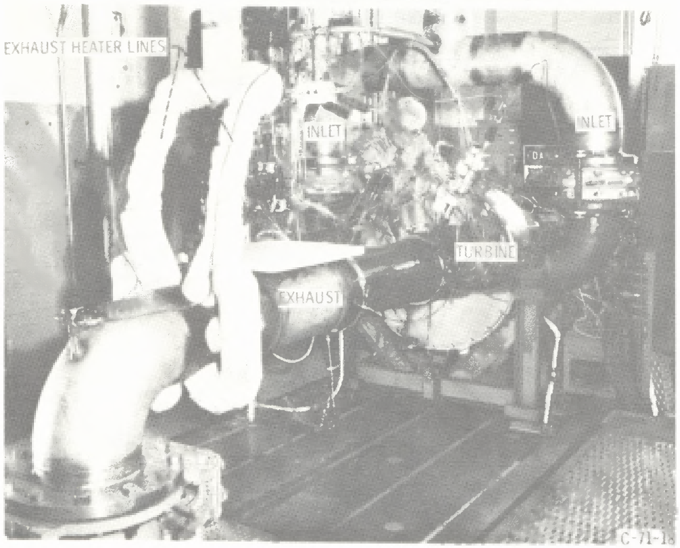

Fig. 4 Turbine test apparatus

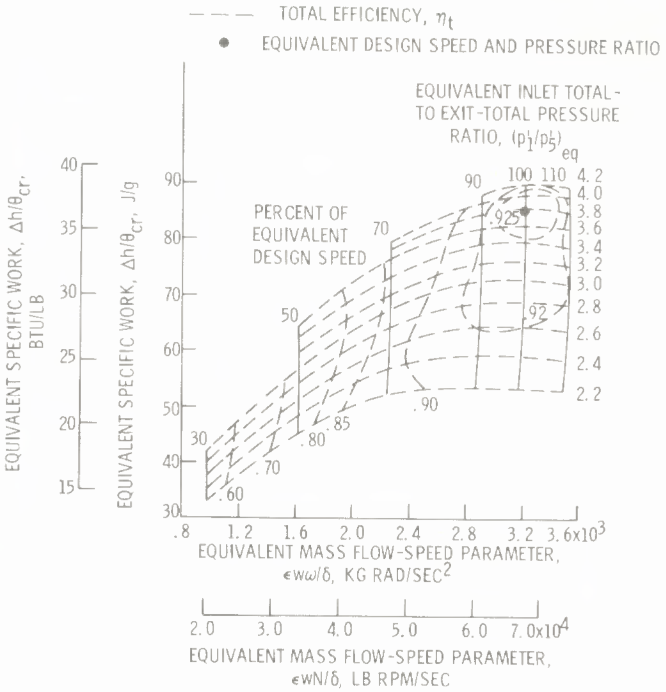

Fig. 5 Overall turbine performance map of two-stage turbine

(2), was determined by measurements taken at stations 1, 2, 3, 4, and 5. For the first-stage investigation, the second-stage was removed and appropriate fairing pieces were inserted to insure smooth flow on the inner and outer walls. The first-stage performance was then determined from measurements taken at stations 1 and 3, the instrumentation at station 3 being identical to that at station 5 for the two-stage turbine. A more complete description of the apparatus, instrumentation, and procedure can be found in reference (2).

RESULTS AND DISCUSSION

The experimental results of the turbine program will be presented in two sections. First, the experimental performance of the two-stage turbine will be discussed in terms of efficiency and mass flow. This will be followed by a discussion of the effect of a reduction in blade thickness on the performance of the first stage of the subject two-stage turbine.

Two-Stage Performance

Turbine total efficiency is presented in Fig. 5 as a performance map. The map consists of equivalent specific work, $\Delta h/\theta_{cr}$, as a function of the mass-flow-speed parameter, $\varepsilon w\omega/\delta$ for the various equivalent speeds investigated. Lines of constant pressure ratio and efficiency contours are superimposed. The figure shows that a total efficiency value of about 0.93 was obtained at equivalent design speed and pressure ratio. This is significantly higher than the design value of 0.88 assumed in the design. The figure also shows that the efficiency varied from 0.60 near 30 percent design speed to about 0.93 at 100 percent design speed. The vertical speed lines indicate that the turbine was choked over most of the range of pressure ratios investigated.

Performance of the first stage as reported in reference (2) was 0.93 at equivalent design speed and pressure ratio. This value was also significantly higher than the 0.870 assumed in the design.

Performance of the second stage under two-stage operation was estimated by the use of first- and two-stage performance and the first-stage rotor exit tip static pressure at design pressure ratio. The results indicated that, at equivalent design speed and pressure ratio, a total efficiency of 0.91 was obtained for the second stage. This value was four points higher than the design value of 0.870. At design point operation, the work split was 58 percent for the first stage and 42 percent for the second stage. This work split compares closely with the 57 to 43 percent work split selected in the design.

Although the experimental tests indicated that the turbine was performing significantly better than design, it does not necessarily follow that the same turbine performance would be

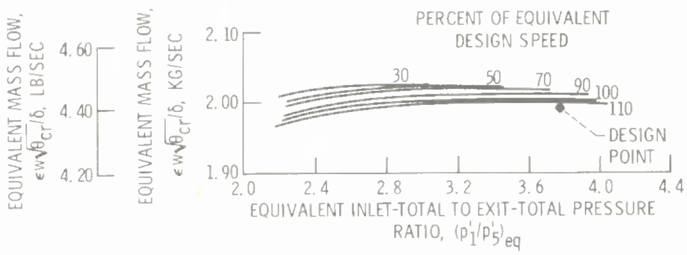

Fig. 6 Variation of mass flow with total pressure ratio and speed for two-stage turbine

obtained in the actual engine. Factors, such as non-uniform inlet conditions of pressure and temperature, increased leakage, and increased tip clearance would reduce the performance to a value somewhat closer to the design value. The inlet conditions to the turbine, for the reported investigation, were optimum for minimum losses. There was no leakage around the first-stage stator, and the second-stage stator had a labyrinth seal to minimize leakage. In addition, the recessed casing with minimum tip clearance also contributed to minimizing the losses.

Fig. 6 shows the variation of mass flow with turbine inlet total to exit total pressure ratio for lines of constant equivalent speed. The turbine chokes at a pressure ratio of about 2.6 at 30 percent design speed and at a pressure ratio of about 3.1 at 110 percent of design speed. The equivalent mass flow was 2.004 kg/sec (4.418 lb/sec) at equivalent design speed and pressure ratio (3.765). This mass flow is about 0.8 percent higher than the design value of 1.989 kg/sec (4.385 lb/sec).

Effect of a Reduction in Blade Thickness on Performance

As was mentioned in the INTRODUCTION, engine costs could be reduced if the turbine blades were coined. The coining process would be simplified if the blades were of near uniform thickness from leading to trailing edge. The first-stage blading of the two-stage turbine was modified to give near uniform blade thickness from leading to trailing edge, and tests were made to determine the effect of this unusual blade shape on performance. The modification of the blade profile was accomplished by changing the pressure surface with the leading edge diameter being used as the value of maximum thickness (Fig. 7). This method was considered satisfactory since the throat dimensions would be unchanged. Fig. 7 shows the comparison of the stator and rotor blade profiles and flow passages at the mean section for both the original and thin blade configurations. It can be seen that there was a substantial reduction in thickness in the region of one third of the chord length as measured from the inlet. Inspection of the stator flow passages, Fig. 7(a), shows that there was no abrupt change in flow area and that the flow passage width was converging satisfactorily. Fig. 7(b) shows the rotor blade profiles and flow passages. The channel width formed by the thin blade profiles is seen to diverge and then converge. This could cause an unfavorable velocity gradient with increased loss.

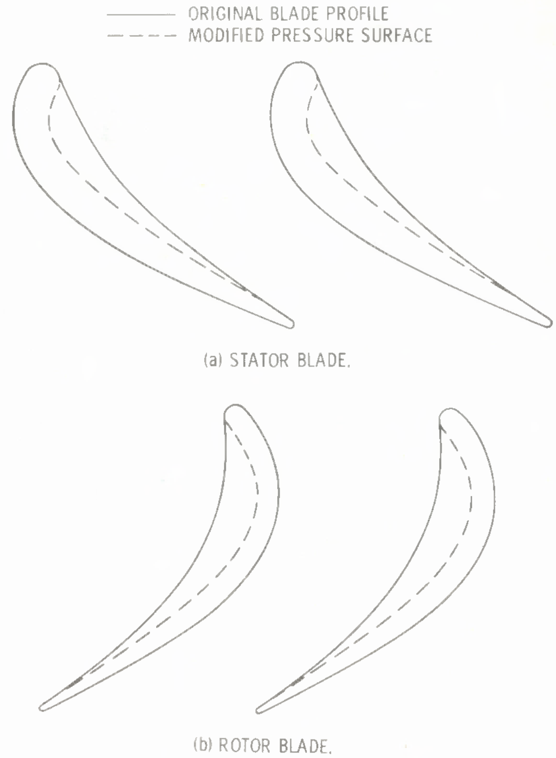

Fig. 7 First-stage mean section profile and flow passage

Fig. 8 shows the stator and rotor blade surface velocities for both the original and the thin blade configurations. The surface velocities were determined by the computer program described in reference (5). The program obtains a transonic flow solution on a blade-to-blade surface of a turbomachine. The transonic solution is obtained by a velocity-gradient method, using information obtained from a finite-difference stream-function solution at a reduced weight flow. Fig. 8(a) shows the blade surface velocities for both stator configurations for the hub, mean, and tip sections. The velocity levels on both suction and pressure surfaces were lower for the thin blade

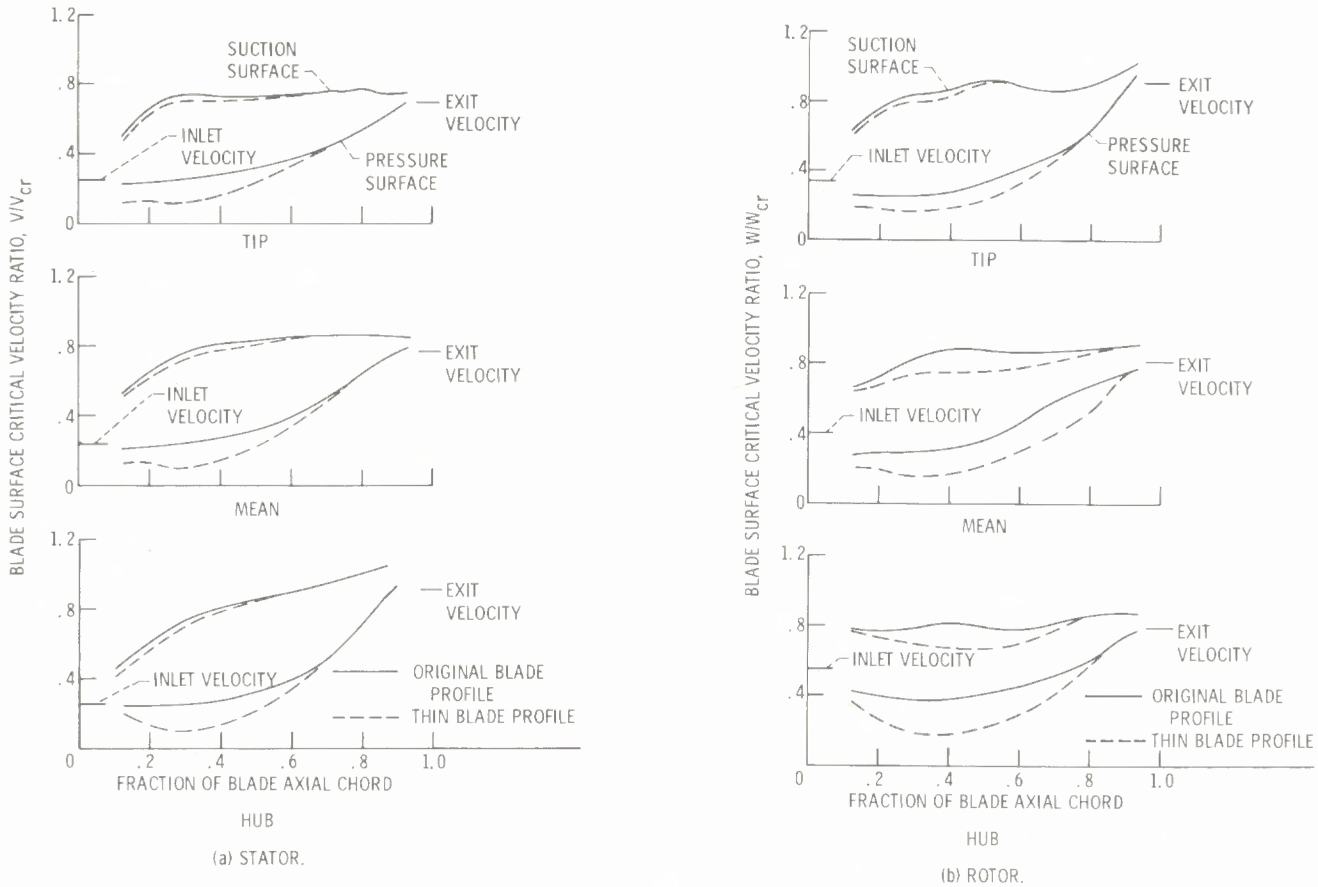

Fig. 8 Design blade surface velocities for blade profiles investigated

configuration. This was primarily a result of the larger passage area with the exception of the throat or minimum area. The figure also indicates that the suction surface diffusion was the same for both blades and that the pressure surface diffusion was higher for the thin blade profile.

Fig. 8(b) shows the rotor blade surface velocities for both blades. Again, the surface velocities for the thin blade are seen to be lower than those obtained for the original blade with the exception of the region near and at the throat of the two blade configurations. Since the throat area was the same for both blades, the mass flow would, therefore, be the same if the losses remained constant. The figure indicates that the suction surface diffusion was the same for both blade configurations. It will also be noted that the pressure surface diffusion would be higher for the thin blade configuration particularly at the mean and hub section. Table 2 gives the Turbine Aerodynamic Parameters for both the original and thin blade profiles. The table shows that there was a substantial increase in pressure surface diffusion when the rotor blade was modified to a thin profile. All other aerodynamic parameters listed in the table were the same for each configuration with the exception of the rotor blade tip clearance. The difference in tip clearance was due to fabrication. A photograph of both rotor blade configurations investigated is shown in Fig. 9.

Fig. 10 shows the performance results obtained with the four possible combinations of original and thin stator and rotor blades. The figure shows the variation of total efficiency with inlet total- to exit-static pressure ratio at equivalent design speed. The best performance was obtained with the two thin stator configurations. However, the maximum difference in efficiency between the four cases was not more than two points at any pressure ratio. At equivalent design pressure ratio, the curves for the two thin stator configurations show an efficiency of about 0.94 as compared to about 0.93 for the

Fig. 9. Rotor blade configurations investigated

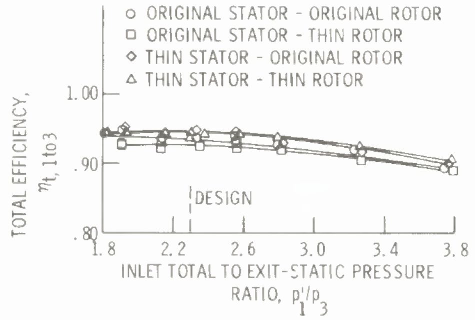

Fig. 10 Variation of total efficiency at equivalent design speed for first-stage operation

original stator configurations. This increase in performance was apparently due to the thin stator. Use of the thin rotor had little effect on performance and appeared to have had an adverse effect on performance when tested with the original stator. The high-pressure surface diffusion for the thin blade may have resulted in some flow separation, but the effects of the flow separation were minimized by the rapid acceleration of pressure surface velocity to the trailing edge.

The reason for the improved performance with the thin stator configuration is not known at this time. Tests of both stator configurations in an annular cascade may give some information for the improved performance.

Values of measured mass flow for each of the four blade configurations are plotted in Fig. 11. Equivalent mass flow is shown as a function of inlet total- to exit-static pressure ratio. Although the figure would indicate large differences due to the scale used in the plot, mass flow for the four configurations did not differ by more than 1.0 percent at any pressure ratio. The mass flows for both original stator configurations are slightly larger than those for the thin. This difference could be attributed to a larger stator throat area in the original stator assembly as a result of fabrication.

CONCLUDING REMARKS

This paper has summarized the principal results obtained during the aerodynamic investigation of a two-stage turbine designed for a small low-cost turbofan engine.

Although the turbine performance was about 5 points better than design, the high efficiency values would not necessarily be expected in actual engine operation. Losses due to non-uniform turbine inlet conditions, increased interstage leakage, and increased tip clearance could result in efficiencies closer to the design values than those obtained in this cold-air investigation. In order to determine the potential of the turbine, inlet conditions to the turbine were optimum for minimum losses. There was no leakage around the first-stage stator, and the second-stage stator had a labyrinth seal with small radial clearance to minimize leakage. In addition, the recessed casing with minimum tip clearance also contributed to minimizing the losses.

Test results have also indicated that the aerodynamic performance of a stator or rotor blade with a large amount of reaction is affected very little by a significant change of the pressure surface. This change resulted in a 50 percent reduction in maximum blade thickness for the turbine investigated. Tests have shown that the first-stage performance of the subject two-stage turbine was actually improved when the stator blade thickness was decreased by approximately 50 percent. The reason for the improved performance is not known at this time. Detailed surveys of both stator configurations in an annular cascade may supply some reasons for the apparent increase in performance.

REFERENCES

1 Cummings, R. L., and Gold, H., "Concepts for Cost Reduction on Turbine Engines for General Aviation," TM X-52951, NASA, Cleveland, Ohio, 1971.

2 Kofskey, M. G., and Nusbaum, W. J., "Design and Cold-Air Investigation of a Turbine for A Small Low-Cost Turbofan Engine," TN D-6967, NASA, Cleveland, Ohio, 1972.

3 Nusbaum, W. J., and Kofskey, M. G., "Effect of a Reduction in Blade Thickness on the Performance of a Single Stage 20.32-Centimeter Mean Diameter Turbine," TN D-7128, NASA, Cleveland, Ohio, 1972.

4 Miser, J. W., Stewart, W. L., and Whitney, W. J., "Analysis of Turbomachine Viscous Losses Affected by Changes in Blade Geometry," RM E56F21, NACA, Cleveland, Ohio, 1956.

5 Katsanis, T., "Fortran Program for Calculating Transonic Velocities on A Blade-to-Blade Stream Surface of A Turbomachine," TN D-5427, NASA, Cleveland, Ohio.

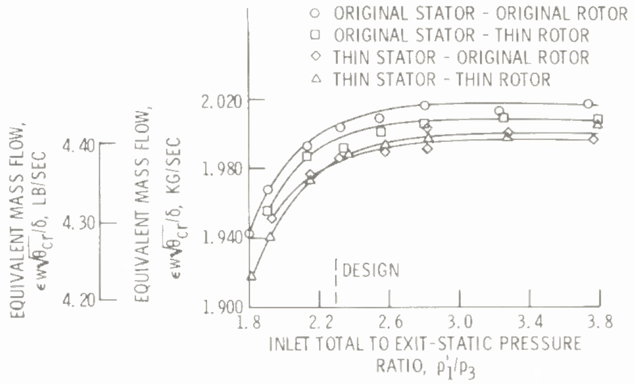

Fig. 11 Variation of equivalent mass flow at equivalent design speed for first-stage operation