The Effect of Rotor Blade Thickness and Surface Finish on the Performance of a Small Axial Flow Turbine

Richard J. Roelke
National Aeronautics and Space Administration
Lewis Research Center

and

Jeffrey E. Haas
Propulsion Laboratory
AVRADCOM Research and Technology Laboratories
Lewis Research Center

Work performed for
U.S. DEPARTMENT OF ENERGY
Conservation and Renewable Energy
Office of Vehicle and Engine R&D

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Richard J. Roelke
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by Richard J. Roelke
National Aeronautics and Space Administration
Lewis Research Center
Cleveland, Ohio 44135

and Jeffrey E. Haas
Propulsion Laboratory
AVRACOM Research and Technology Laboratories
Lewis Research Center
Cleveland, Ohio 44135

ABSTRACT

An experimental investigation was conducted to determine the effect of blade profile inaccuracies and surface finish on the aerodynamic performance of a 11.15 cm tip diameter turbine. The as-received cast rotor blades had a significantly thicker profile than the design intent and a fairly rough surface finish. Stage test results showed an increase of one point in efficiency by smoothing the surface finish and another three points by thinning the blade profiles to near the design profile. Most of the performance gain between the as-cast thick and the thinned rotor blades, both with the same surface finish, was attributed to reduced trailing edge losses of the re-contoured blades.
INTRODUCTION

The efficiency of small (under about 15 cm tip diameter) axial turbines has not equaled that demonstrated in larger machines. The chief reasons for this are Reynolds number effects and compromises made in the aerodynamic design to accommodate limitations in mechanical design and fabrication processes. A practical small turbine design will almost always have a lower blade aspect ratio, higher trailing edge blockage, and a higher rotor tip clearance than a similar large turbine. Further performance degradation may also be caused by manufacturing imperfections because it is difficult to make the blade profiles with the same precision or relative surface smoothness as large turbines. The effect of these manufacturing imperfections on the performance of a small single stage turbine is the subject of this paper.

Few reports have appeared on the effect of these manufacturing imperfections in comparison to the other causes affecting the performance of small turbines. Bammert and Sandstede (1) reported on a series of cascade tests and a four stage turbine test where the surface roughness was changed and the blade profiles were either uniformly thinned or thickened to simulate manufacturing errors. Their results indicated dramatic changes in blade losses.

The results of the investigation described herein are an outgrowth of the automotive gas turbine technology program conducted at the NASA-Lewis Research Center. A part of that program consisted of a series of component performance tests of the compressor-drive turbine for the Department of Energy automotive gas turbine demonstrator engine. The engine and technology program are described in (2). The turbine blading used in the subject tests consisted of duplicates of the stator and rotor castings used in the demonstrator engine. Inspection of the blading made before the start of the turbine component tests showed significant deviations from design in the profile shape and a fairly rough surface. The initial tests were made to determine the performance of the as-cast blading. After these initial tests two subsequent turbine builds were evaluated. One build had reduced rotor blade surface roughness and in the other build the rotor blade profiles were reworked to more nearly approach the design profile.

All performance tests were conducted with air at a nominal inlet temperature of 320 K and an inlet pressure of 0.827 bars. The results reported in this paper were obtained by measuring the overall stage performance for a range of pressure ratios with the turbine operating at design speed. Rotor-exit radial surveys of angle, total pressure and total temperature were taken at design equivalent values of speed and specific work. Results are presented in terms of efficiency and mass flow for each of the three turbine builds. Also included is the effect of the blading changes on the static pressures within the stage and the calculated changes in local efficiency based on the rotor-exit surveys. The results of the complete series of performance tests conducted with this turbine are reported in (3).

SYMBOLS

m  mass flow rate, kg/sec
r  radius, m
U  blade velocity
V  absolute gas velocity, m/sec
W  relative gas velocity, m/sec
α  absolute gas angle measured from axial direction, deg
were duplicates of the stator and rotor castings.

0.929 file tolerance

ever, moderate diffusion was predicted for both the
dicted on either of the stator blade surfaces; how­

The inspection traces made showed profile variations
before the start of the turbine testing showed sig­
ificant deviations from design in the profile shape
and a fairly rough surface. Figure 3 compares in­
spection tracings of the mean and tip sections with
those measured after polishing and coating the
hub. The average trailing edge blockage for the
reworked rotor was about 13 percent. The respective
suction and pressure surface roughness measurements
of the reworked rotor were essentially the same as
those measured after polishing and costing the
as-cast rotor. Tests were then conducted using the
reworked rotor blading.

TURBINE DESCRIPTION

The turbine used in this program was the NASA­
designed compressor drive turbine for the Department
of Energy automotive gas turbine demonstrator
engine. A cross-section of the turbine is shown in
Fig. 1. Reference 4 describes the aerodynamic de­
sign of the turbine. The turbine was designed with
a tip diameter of 11.15 cm, stator and rotor blade
heights of nominally 1.12 cm and trailing edge
thicknesses of 0.038 cm for both blade rows. There
were 15 stator blades and 62 rotor blades. The
design trailing edge blockages were nominally 4.3
and 11.8 percent for the stator and rotor respec­
tively. The turbine was designed for a work factor
(specific work/mean blade speed squared) of 2.1.
The design mean section velocity diagram and blade
surface velocity distributions are shown in Fig. 2.
The stator exit absolute critical velocity ratio was
0.64 and the rotor exit relative critical velocity
ratio was 0.818. Very little diffusion was pre­
dicted on either of the stator blade surfaces; how­
ever, moderate diffusion was predicted for both the
pressure and suction surfaces of the rotor blade.

The turbine blading used for component testing
were duplicates of the stator and rotor castings
used in the engine. Inspection of the rotor blading
before the start of the turbine testing showed sig­
nificant deviations from design in the profile shape
and a fairly rough surface. Figure 3 compares in­
spection tracings of the mean and tip sections with
the design profile of two randomly selected rotor
blades. Hub section tracings were not obtained
because the tracing stylus was too large to fit in
the small hub area. The inspection tracings show
some waviness in the blade profiles and an increase
in the blade thickness. Measurements made indicated
local regions of the blade profiles fell outside the
profile tolerance band by up to 0.05 mm. The pro­ile tolerance was ±0.1 mm. The average trailing
edge thickness based on these and other inspection
tracings and the hub throat measurement was 0.053 cm
resulting in a trailing edge blockage of 16.5 per­
cent. Surface roughness measurements were made on
the pressure and suction surfaces of several blades
and averaged 1.35 microns.

Surface velocity distributions were not gen­
erated for the as-cast rotor blades primarily
because these blades were not "typical" as-cast profile.
The inspection tracings made showed profile variations
from blade to blade. Also tracings of the hub sec­
tion profile could not be obtained.

Profile tracings for the stator were not gen­
erated since the stator blades were cast integrally
with the endwalls. However, stator throat measure­
ments indicated that the stator flow area was undersized
by 4.1 percent. The reduced flow area was caused by
the size of the fillets and draft angles used in
casting the stator. The as-cast stator trailing
edge blockage was nominally 4.5 percent.

Turbine aerodynamic performance tests were made
using the as-cast blading. After these tests were
made, two modifications were made to the rotor blad­
ing. The first modification consisted of polishing
the blade surface roughness. This process consisted
of polishing the suction surface of each of the
blades (reducing the average suction surface rough­
ness to 0.33 microns) and applying a thin coat of
lacquer to the pressure surfaces. The average pres­
sure surface roughness was 0.05 micron resulting in
an average surface roughness for the blade of
0.64 microns. Tests were then made on this con­
figuration. The second modification consisted of
electric discharge machining the rotor profiles to
the design profile. The process consisted of slowly
removing metal from the rotor profiles until inspec­
tion traces at the mean and tip agreed closely with
the design profile. Rotor throat measurements indi­
cated that the hub section was still thick. How­
ever, any further hub machining may have resulted in
undersized profiles away from the hub, and perhaps
steps in the hub endwall if the machining electrode
had touched the hub. Figure 4 shows a comparison of
the throat dimensions for the design, as-cast, and
reworked rotor profiles. This figure shows the
close agreement in the throat dimension between the
design and reworked rotors, near the mean and tip
sections, and the difference that remained near the
hub. The average trailing edge blockage for the
reworked rotor was about 13 percent. The respective
suction and pressure surface roughness measurements
of the reworked rotor were essentially the same as
those measured after polishing and costing the
as-cast rotor. Tests were then conducted using the
reworked rotor blading.

RESEARCH EQUIPMENT AND PROCEDURE

The apparatus used in this investigation con­
sisted of the research turbine, an airbrake dyna­
ometer used to control the speed and absorb and
measure the power output of the turbine, an inlet
and exhaust piping system including flow controls,
and appropriate instrumentation. A schematic of the
experimental equipment is shown in Fig. 5. The
rotational speed of the turbine was measured with an
electronic counter in conjunction with an electronic
counting device, a magnetic pickup and a shaft-mounted gear. Mass flow was mea­
sured with a calibrated venturi. Turbine torque was
determined by measuring the reaction torque of the
airbrake which was mounted on air trunion bearings,
and adding corrections for rarefied losses. The torque
load was measured with a commercial strain-gage load cell.

The turbine instrumentation stations are shown
in Figs. 1 and 6. Instrumentation at the manifold
inlet (Station 0) measured wall static pressure,
total pressure, and total temperature. At both the
stator inlet (Station 1) and stator exit (Station 2)
static pressures were measured with six taps with
three each on the inner and outer walls. The inner
and outer wall taps were located opposite each other
at different intervals around the circumference.

The rotor exit instrumentation station
(Station 3) was located in a constant area exhaust
duct approximately three axial chord lengths down­
stream of the rotor. This location was determined
using a hot-wire anemometer survey probe so that the
rotor exit instrumentation could be located in a position where the rotor wakes were mixed out.

At the rotor exit static pressure, total pres­
sure, total temperature and flow angle were mea­
A probable explanation for a major portion of the inefficiency between the as-cast and reworked blade. The rotor tip clearance was the same for all turbine configurations tested and equaled 1.7 percent of the blade length.

For each rotor configuration a rotor exit radial survey was first conducted at design equivalent speed and specific work. Mass averaged values of flow angle, total temperature, and total pressure were obtained for each of the three survey locations. These mass-averaged values were then arithmetically averaged to obtain overall values. The survey probes were then positioned with one each near the tip, near midspan, and near the hub so that the average flow angle from these three positions would correspond closely to the overall mass-averaged value obtained from the survey. The radial positions of the survey probes, so determined, were not changed during the remainder of the testing of that rotor configuration. Performance data were then obtained over a range of turbine stage total-pressure ratio at design equivalent speed. Performance data were then obtained over a range of turbine stage total-pressure ratio at design equivalent speed. The rotor exit survey data together with the overall stage measurements and the results of a stator exit survey (5) were used to calculate the stage velocity diagrams for the three turbine builds at the design work condition. Selected results from those calculations are tabulated in Table I. The velocity diagram information listed in the table shows that the flow velocities generally decreased in the rotor and increased in the stator as the rotor configuration was changed. This indicates that the lowest mass flow had the highest incidence among the three turbine configurations.

The changes in stage efficiency for the three turbine builds are shown in Fig. 8. The difference in efficiency between the as-cast and reduced roughness rotor was nominally one point and between the as-cast and reworked profile was nominally four points. These differences could be expected to increase further if it were possible to thin the rotor profile near the hub and to further smooth the surface finish of the rotor and stator blades. A probable explanation for a major portion of the increase in performance of the reworked rotor configuration is the reduced trailing edge losses. An analysis of these losses is discussed later in this paper.

The increases in wall static pressure through the turbine for the design equivalent total-to-total pressure ratio of 2.01 are shown in Fig. 9. All pressures were ratioed to the inlet total pressure at Station 0. As the rotor blades were first smoothed and then thinned the static pressure between the stator and rotor decreased slightly. This change in reaction was more evident in the stator and reduced it across the rotor; however, positive rotor reaction was always maintained. It was felt that the decrease in rotor reaction was not large enough to decrease either the rotor or stage efficiency.

The rotor exit radial surveys of total temperature, total pressure, and flow angle conducted at design equivalent values of speed and specific work were used to determine radial variations in stage performance. The radial variations in turbine efficiency calculated from these survey data are shown in Fig. 10. These results show that the largest benefits of reducing the surface roughness and thinning the blade profile occurred from midspan out to the tip. This may have occurred because, as mentioned earlier, it was difficult to improve the blade surface finish and profile near the hub so that the average flow angle from these three positions was the same as the original rotor. The highest flow was measured with the reworked profile but the difference is small, only about 0.7 percent. The flow increase with the reworked rotor was much less than the increase in flow area of that rotor which was three percent. These results indicate, that at this rotor speed, the as-cast rotor choked before the stator but with the reworked rotor installed the stator choked first and therefore controlled the stage mass flow.

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ing edge difference between the as-cast and reworked blades. This change in rotor efficiency was then used in a turbine performance computer code (9) to predict the effect on the stage performance. The results of the analysis indicated an increase in stage efficiency of 2.7 points by thinning the rotor blades. The difference in efficiency shown in Fig. 8 between the reduced roughness and reworked rotors was nominally three points. Therefore, it appears that the reduction in rotor trailing edge loss was the main reason for the performance gain.

An analysis procedure similar to that described above for the effect of blade trailing edge thickness was used to try to predict the effect of blade surface roughness. However, the results were inconclusive because of present limitations in the referenced boundary layer code to account for surface roughness changes.

A comparison was also made between the effect of blade surface roughness measured in this test program and the results reported in (1). According to the information in the reference a decrease in the surface roughness from 1.35 microns to 0.64 microns could result in an increase in stage efficiency of about 1.2 points. This predicted increase in efficiency is generally consistent with the increase measured during the test program. An increase of nominally one point was measured with the turbine operating at design equivalent speed and work.

CONCLUDING REMARKS

The results obtained in this experimental investigation showed that inaccuracies in the manufacture of small turbine blades can cause significant turbine performance penalties. Small dimensional deviations from design that may be acceptable in large machines must be critically examined to judge the impact in a small machine. Ultimately, analytical methods must be developed to predict these effects. For this turbine, obtaining an accurate blade profile and a smooth surface finish significantly improved its performance. Analytical predictions of these effects compared closely with the experimental results. Finally, the quality of the castings procured for this demonstrator engine program may not be indicative of what may be reasonably achieved given more time to further develop the manufacturing processes.

REFERENCES


<table>
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<tr>
<th>TABLE 1. - CALCULATED STAGE VELOCITY DIAGRAMS AT DESIGN WORK FACTOR</th>
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</thead>
<tbody>
<tr>
<td><strong>Percent span</strong></td>
</tr>
<tr>
<td>(hub is zero)</td>
</tr>
<tr>
<td><strong>Station 2</strong></td>
</tr>
<tr>
<td>Absolute velocity</td>
</tr>
<tr>
<td>Relative velocity</td>
</tr>
<tr>
<td>Relative flow</td>
</tr>
<tr>
<td>angle, deg</td>
</tr>
<tr>
<td><strong>Station 3</strong></td>
</tr>
<tr>
<td>Absolute velocity</td>
</tr>
<tr>
<td>Relative velocity</td>
</tr>
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### Table 1 - Calculated Stage Velocity Diagrams at Design Work Factor

<table>
<thead>
<tr>
<th>Percent span (hub is zero)</th>
<th>Design</th>
<th>As-cast</th>
<th>Reduced roughness</th>
<th>Reworked</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>50</td>
<td>5</td>
<td>25</td>
<td>50</td>
</tr>
<tr>
<td>Station 2</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Absolute velocity ratio</td>
<td>0.929</td>
<td>0.945</td>
<td>0.899</td>
<td>0.869</td>
</tr>
<tr>
<td>Relative velocity ratio</td>
<td>0.559</td>
<td>0.632</td>
<td>0.551</td>
<td>0.495</td>
</tr>
<tr>
<td>Relative flow angle, deg</td>
<td>45.8</td>
<td>44.7</td>
<td>47.6</td>
<td>46.0</td>
</tr>
<tr>
<td>Rotor incidence, deg</td>
<td>0.1</td>
<td>-4.3</td>
<td>-0.2</td>
<td>0.3</td>
</tr>
<tr>
<td>Station 3</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Absolute velocity ratio</td>
<td>0.521</td>
<td>0.524</td>
<td>0.593</td>
<td>0.570</td>
</tr>
<tr>
<td>Relative velocity ratio</td>
<td>0.818</td>
<td>0.808</td>
<td>0.862</td>
<td>0.845</td>
</tr>
<tr>
<td>Absolute flow angle, deg</td>
<td>-21.1</td>
<td>-27.6</td>
<td>-23.1</td>
<td>-19.2</td>
</tr>
</tbody>
</table>
Figure 1. - Cross-sectional schematic of turbine.

Figure 2. - Design mean section velocity diagram and blade surface velocity distributions.
Figure 2. - Concluded. Figure 3. - Comparison of design and as-cast rotor blade profiles.
Figure 4. - Radial variation in rotor throat dimension.

Figure 5. - Test installation schematic.
INSTRUMENTATION

- STATIC PRESSURE
- TOTAL PRESSURE
- TOTAL TEMPERATURE
- TOTAL PRESSURE - FLOW ANGLE

Figure 6. - Flow path instrumentation, viewed looking downstream.

Figure 7. - Variation of equivalent mass flow with pressure ratio at design speed.

Figure 8. - Variation of efficiency with pressure ratio at design speed.
Figure 9. - Variation of static pressure with axial location at design speed and stage total pressure ratio.

Figure 10. - Radial variation in efficiency at design speed and work.
# Title and Subtitle
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# Authors
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# Abstract
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# Key Words
Axial turbine
Blading imperfections
Aerodynamic performance

# Security Classification
Unclassified

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