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Performance of a High-Efficiency Radial/Axial Turbine

This paper presents the test performance of a lightly loaded, combination radial/axial turbine for a 420-hp, two-shaft gas turbine. This two-stage turbine configuration, which included an interstage duct and an exhaust duct discharging vertically to ambient pressure conditions, was shown to be capable of attaining an overall isentropic efficiency of 89.7 percent. The influence of exhaust diffuser struts on the turbine performance under stalled power turbine conditions was shown to significantly affect compressor and turbine matching.

Introduction

Virtually all gas turbines with multistage turbines use the axial turbine configuration. A few gas turbines have been produced and developed, however, with a single-stage, radial inflow turbine followed by a single-stage axial turbine. Most notable among those produced are the AiResearch GTCP 105, the AiResearch GTCP 601, and Solar's T351 Spartan gas turbine.

The results of extensive experimental testing of a high pressure ratio radial/axial turbine configuration for an advanced auxiliary power unit are described by Kidwell and Large [1]. At an overall pressure ratio of 9.0, cold turbine rig tests showed an isentropic expansion efficiency of 87 percent.

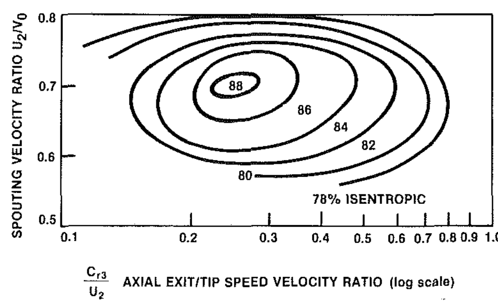
The radial/axial turbine arrangement is particularly suitable for small gas turbines with reverse flow combustors or scroll combustors, since the combustor exit 180-deg bend can be supplanted by the radial nozzle and radial inflow rotor. Furthermore, the radial rotor is less sensitive to tip clearance than the small blade height axial rotor. Counteracting these advantages are the losses of the interstage duct between the exducer and the inlet of the axial nozzle, and the potentially higher inertia of the radial rotor.

Providing the interstage duct losses are minimized, the optimum radial/axial turbine arrangement is usually more efficient than the optimum two-stage axial arrangement at the same pressure ratio in the 2.0 to 10.0 pps flow engine category.

This paper describes the development test of a radial/axial turbine for the 420-hp, two-shaft T351 gas turbine, where it was demonstrated that an overall total-to-ambient turbine efficiency of 89.7 percent was attained based upon actual gas turbine test calibrations.

Radial Turbine Design Optimization

The aerodynamic excellence of both radial inflow and axial turbines is dominated by two major performance parameters:



- Total-Static Efficiency Including Exhaust Diffuser Turbines 6.25 to 10.5 in. Tip Diameter.
- Inlet Blade Angle 90°
- Pressure Ratio 3.0 to 5.0
- Axial Clearance/Tip Diameter .005 to .01
- U_2 Tip Speed
- C_{r3} Rotor Axial Exit Velocity
- V_0 Isentropic Spouting Velocity $(2gJCp\Delta Tisen)^{0.5}$

Fig. 1 Attainable efficiency levels of radial inflow turbines

- Velocity ratio $= U_2/Vo$
- Exit flow coefficient $= \phi = Cr_3/U_2$

The velocity ratio is a direct measure of the blade loading. The exit flow coefficient for a radial turbine is an indirect measure of the specific speed since for zero exit swirl and incompressible flow it can be shown that:

Dimensionless specific speed $N_s \sim \phi^{1/2}\left(\frac{U_2}{Vo}\right)^{1.5}\frac{2.98}{\epsilon}$

where

$$\epsilon = \text{Rotor diameter ratio} = D_2/D_{3\ \mathrm{rms}}$$

State-of-the-art levels for the total-to-static efficiency of uncooled radial inflow turbines are shown in Fig. 1, indicating peak efficiencies are obtained with velocity ratios close to 0.7 with exit flow coefficients of 0.2 to 0.3.

Similar charts have been produced by Smith [2] for axial turbines showing comparable trends. For a particular turbine design, the level of efficiency is dependent upon specific criteria, such as:

- Solidity of nozzle and rotor
- Effects of diameter ratio ϵ
- Tip clearance losses

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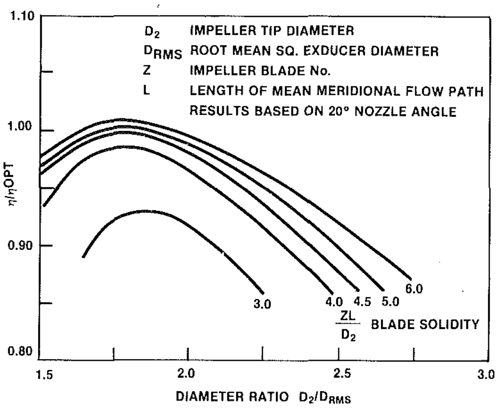

Fig. 2 Effect of rotor solidity on radial turbine efficiency

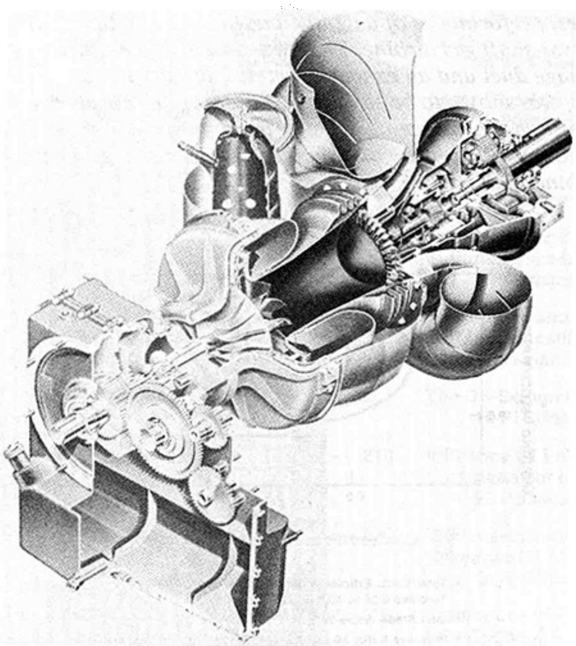

Fig. 3 T351 gas turbine

- Vane and blade trailing edge thickness
- Entry flow conditions
- Mach and Reynolds numbers

The effects of rotor solidity and diameter ratio upon radial turbine efficiency are correlated in Fig. 2, wherein small diameter ratios decrease efficiency as a consequence of nonuniform shroud loading.

A high-performance radial inflow turbine should be selected close to the maximum efficiency island of Fig. 1 and have an optimum diameter ratio ϵ and blading solidity. These basic guidelines were adopted in choosing the geometry of the T351 radial inflow turbine described herein.

Description of Test Gas Turbine

The T351 was designed as a two-shaft version of the T350 single-shaft gas turbine described in [3].

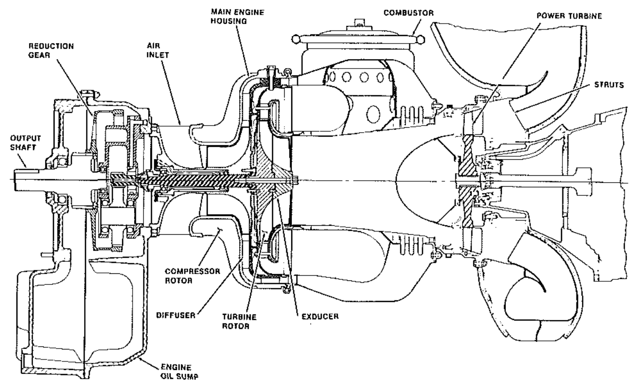

Fig. 4 T351 gas turbine cross section

Table 1 Test instrumentation

Instrumentation	Number	Type	Accuracy
Barometer	1	- - -	±.05 psi
Bellmouth Inlet Air Temperature	1	C/A Thermocouple	±3°
Bellmouth Static Pressure	3	.03 inch	±.05 psi
Compressor Inlet Temperature	3	C/A Thermocouple	±1°
Compressor Inlet Total Pressure	3	Kiel	±0.1 psi
Compressor Exit Total Pressure	3	Kiel	±0.1 psi
Compressor Exit Total Temperature	2	C/A Thermocouple	±3°
Gas Producer Turbine Inlet Total Pressure	2	Kiel	±0.1 psi
Gas Producer Turbine Exit Total Pressure	3	Kiel	±0.1 psi
Power Turbine Inlet Total Temperature	1	I/C Thermocouple	±10°F
Exhaust Total Temperature	6	I/C Thermocouple	±10°F
Rotational Speeds	2	Tachometers	±0.2%
Output Power	1	Water Dynamometer	±5 hp
Fuel Flow	1	Rotometer	±3 pph

The T351 gas turbine with its separate shaft power turbine is shown in Figs. 3 and 4. This gas turbine has a single-stage centrifugal compressor, driven by a single-stage gas producer radial inflow turbine, feeding into a single-stage axial power turbine, followed by an exhaust collector with a vertical discharge.

The gas turbine was designed to deliver an output power of 420 hp at sea level, 80°F conditions with a corresponding conservative turbine inlet temperature of 1450°F.

Compressor discharge air was collected in the outer combustor scroll and passed through a single can combustor to the inner turbine scroll. Combustor exit swirl entering the gas producer radial nozzles was on the order of 40 deg.

The axial power turbine was a modified second-stage turbine from Solar's Saturn gas turbine [4] fitted with interstage transition ducting to permit matching to the gas producer tailcone and existing Saturn turbine exhaust collector. The manner in which the interstage transition was accomplished is of particular interest in that for simplicity and cost reasons an inner cone was not used. Rapid acceleration into the power turbine nozzle was accomplished with a large centerbody.

The exhaust collector consists of an annular diffuser transitioning into a vertical collector. The centerbody and power turbine bearing housing was supported by eight airfoil, zero-camber struts located approximately one chord downstream of the power turbine blades. The effect of the close proximity of the struts on stalled power turbine performance is discussed later in this paper.

Nomenclature

C = velocity
D = diameter
g = gravitational constant
J = Joules equivalent
L = blade length
N_s = dimensionless specific speed
P = total pressure
p = static pressure
T = total temperature
U = tip speed
Vo = spouting velocity
W = flow
Z = blade number
η = isentropic efficiency
ϵ = diameter ratio
ϕ = exit flow coefficient

Subscripts

2 = radial turbine inlet
3 = radial turbine exit
4 = axial turbine inlet
5 = exhaust diffuser exit (ambient)
rms = root mean square
m = mean
t = turbine

Table 2 Radial turbine aerodynamic details

Parameter	Dimension
Inlet Total Temperature, °R	1910
Inlet Total Pressure, psia	58
Inlet Flow, pps	6.52
Rotational Speed, rpm	36,200
Nozzle Entry Diameter, in.	13.4
Nozzle Exit Diameter, in.	10.7
Nozzle Throat Area, sq. in.	10.6
Nozzle Vane Height, in.	0.75
Number of Nozzle Vanes	19
Number of Rotor Blades	12
Rotor Tip Height, in.	0.75
Rotor Tip Diameter, in.	9.8
Inlet Blade Angle, deg	90.0
Inlet Incidence, deg	-15.0
Exducer Tip Diameter, in.	7.5
Exducer Hub Diameter, in	1.5
Exducer RMS Blade Angle, deg	57
Exducer Exit Swirl, deg	-6.0
Pressure Ratio (Total-to-Total)	2.2
Velocity Ratio U_2/Vo	0.71
Exit Axial Velocity/Tip Speed, Cr_3/U_2	0.34
Total-to-Total Isentropic Efficiency, *	0.92
Dimensionless Specific Speed, N_s	0.77

*Indicates interstage duct

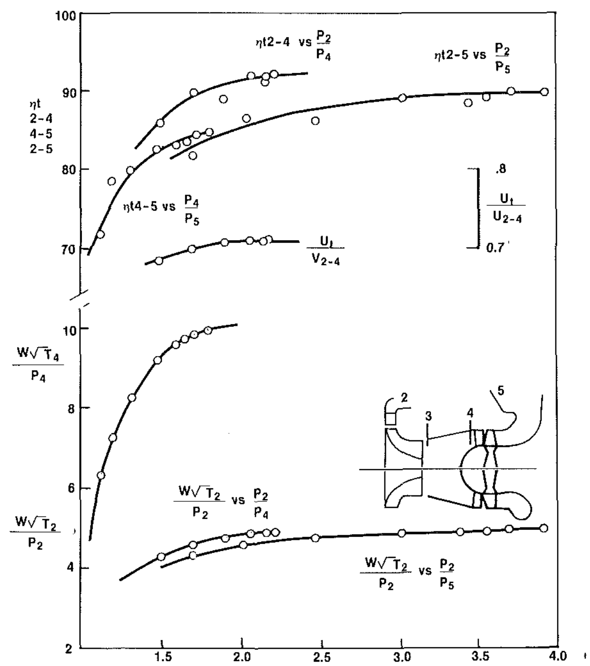

Fig. 5 T351 gas turbine test performances

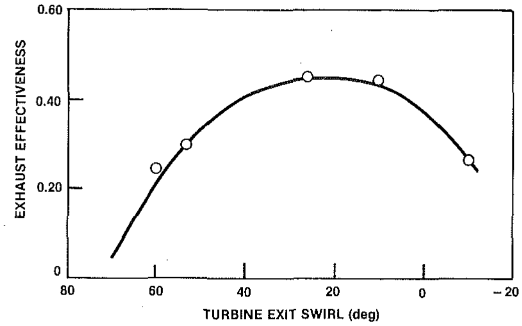

Fig. 6 Exhaust diffuser effectiveness

Table 3 Axial turbine aerodynamic details

Parameter	Dimension
Inlet Total Temperature, °R	1601
Inlet Total Pressure, psia	26.0
Inlet Flow, pps	6.52
Rotational Speed, rpm	22,000
Inlet Axial/Exducer Axial Velocity, *	1.16
Nozzle Entry Annulus Area, sq. in.	46.8
Nozzle Exit Angle*, deg	64.0
Nozzle Vane Number	41.0
Nozzle Solidity	1.46
Rotor Solidity	1.45
Rotor Blade Angle*, deg	11.0
Rotor Inlet Incidence*, deg	1.0
Rotor Blade Number	58
Rotor Exit Annulus Area, sq. in.	55.5
Rotor Exit Swirl*, deg	4.0
Exit Axial Velocity/Blade Speed, Cr_3/U_2	0.53
Work Coefficient, $gJ\Delta H/U_m^2$	1.07
Reaction	0.43
Velocity Ratio, Um/Vo	0.59
Overall Isentropic Efficiency, %	84.7
Total-to-Total Isentropic Efficiency, %	92.0
Pressure Ratio (Total-Ambient)	1.792

*Mean section 11.1-inch diameter

Several test calibrations and gas turbine builds were completed, mostly operating the power turbine along its optimum efficiency locus. Some stalled power turbine testing was also conducted by locking the dynamometer arm.

The data from the gas turbine test instrumentation listed in Table 1 provided the input for a computerized performance analysis code. Gas producer radial turbine efficiency was computed from a work balance with the single-stage centrifugal compressor. Power turbine efficiency was computed from measurement of the pressure ratio, exit temperature, airflow, and output power. An average specific heat of 0.276 Btu/lb R and a specific heat ratio of 1.33 was used for the expansion process.

T351 Radial Turbine Performance

Extensive development previously conducted on the single-shaft version of the T350 Spartan gas turbine [3] produced relatively good performance with overall total-to-ambient efficiencies of up to 88 percent including the exhaust at stage pressure ratios of up to 4.2. Design analyses for the T351 two-shaft gas turbine application showed that the total-to-total gas producer stage pressure ratio would decrease to 2.2, requiring a nozzle rematch and decrease in rotor tip diameter of 11 percent for operation at optimum velocity ratio.

Aerodynamic details of the gas producer radial turbine are listed in Table 2. The rotor was 9.8 in. in diameter with 12 radial element blades. At the design velocity ratio of 0.71, the measured total-to-total turbine efficiency was 92.0 percent with an inlet incidence of −15.0 deg and exit swirl of −6.0 deg.

Test performance of the radial turbine, as determined from test calibrations of the T351 gas turbine using the instrumentation itemized in Table 1, is shown in Fig. 5. This performance is based upon nozzle inlet total-to-power turbine inlet total pressures (i.e., including the interstage transition duct).

T351 Power Turbine Performance

Extensive development was also conducted on the three-stage axial turbine and exhaust diffuser for the Saturn gas turbine program. Design analyses for application of the Saturn's second-stage turbine for the T351 power turbine revealed that the only change required was to rematch the second-stage nozzle by increasing the throat area 7 percent.

Aerodynamic details of the axial power turbine are listed in Table 3. With a design rotational speed of 22,000 rpm, the meanline-velocity ratio was 0.59, at which a total-to-ambient isentropic efficiency including the exhaust diffuser of 84.7 percent was measured. The power test performance shown in Fig. 5, in combination with the radial gas producer turbine,

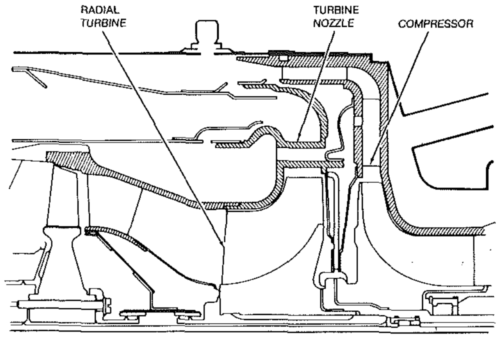

Fig. 7 S140 gas turbine configuration

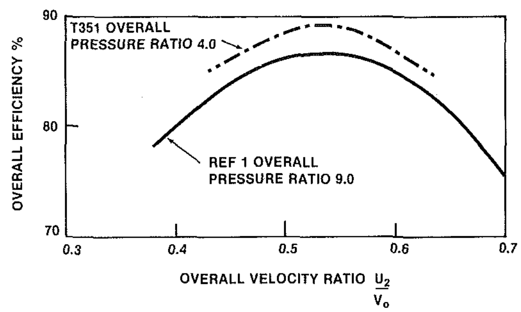

Fig. 8 Radial/axial turbine overall efficiency potential

resulted in the attainment of an overall expansion efficiency for both turbines of 89.7 percent at design point.

Meanline vector conditions were very close to optimum with an inlet incidence of one degree exit swirl of 4.0 degrees and reaction of 0.43. The ratio of the power turbine inlet and gas producer exducer axial velocities was 1.16.

Stalled Torque Performance

Application of the T351 two-shaft gas turbine was earmarked for cement cracking in the petroleum industry where the power turbine could be operated over a range of speeds for stall (zero speed) to runaway conditions.

Gas turbine development testing was therefore conducted operating the power turbine under simulated field conditions. During stalled power turbine testing, the compressor match point moved adjacent to the compressor surge line, and random surge "popping" was experienced close to design gas producer speed.

Component performance data indicated a reduction of power turbine flow function of 9 percent at stalled conditions. This prompted traversing of the power turbine exit to assess the influence of exit swirl upon the exhaust diffuser performance.

Exhaust diffuser static pressure effectiveness as calculated from the traverse pressure measurements is plotted in Fig. 6 versus diffuser inlet swirl angle. Maximum effectiveness was 0.45 near 15-deg swirl, rapidly reducing to zero as stalled conditions were approached. Similar trends are reported in [5]. Twelve equally spaced straight axial struts, as shown in Figs. 3 and 4, were used to support the power turbine bearing housing.

The reduction in power turbine flow capacity reduced the gas producer pressure ratio and flow capacity at rated speed, displacing the operating line closer to surge. The eventual solution to improve surge margin at stalled conditions was to open both the gas producer and power turbine nozzles by approximately 3 percent.

Discussion

Although the radial/axial turbine configuration is capable of attaining high overall turbine efficiency levels in small gas turbines, its popularity is limited. This may be due, in part, to selection of relatively low pressure ratio uncooled designs, for which a closely coupled, lightweight, two-stage axial design may attain comparable efficiencies.

An example of a closely coupled radial/axial turbine configuration for a small 140-hp gas turbine is shown in Fig. 7. This particular gas turbine, Solar's S140, was used as a gas turbine starter system and attained an overall turbine efficiency of 86 percent with an axial diffuser.

The desire to improve thermal efficiencies has motivated increased pressure ratios for single-stage centrifugal compressors to the level where the two-stage axial turbine becomes overloaded, and either a three-stage axial or radial/axial turbine offers higher performance.

The test performance of the radial/axial turbine described in this paper and supported by the work in [1] verifies that the radial/axial turbine configuration is an attractive candidate for fuel-efficient, low-cost, small gas turbines.

The combined overall performance potential of radial/axial turbines with fixed nozzles including the interstage and exhaust ducting is presented in Fig. 8, where overall efficiency is plotted versus velocity ratio based upon the tip speed of the radial stage. These potential performance levels will be attractive for higher pressure ratio small gas turbines, providing the cooling flow considerations for higher temperature operation of the radial turbine stage (with its larger hub surface area) do not incur additional cycle performance losses.

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