Cooled Radial Turbine for High Power-to-Weight Applications

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The design of a high-temperature, high-work radial turbine for advanced small gas turbine engines is described. This turbine is designed to produce nearly 220 Btu/lb gas flow at 87.5% efficiency, with a 5:1 stage pressure ratio. Turbine inlet gas conditions are 17.5 atm and 2300°F. The resulting turbine configuration includes an air-cooled, 12-bladed rotor designed for 67,000 rpm, and a 20-vaned, air-cooled nozzle section of a reflex (supersonic) design. Both parts are IN 100 (PWA 658) investment castings. The discussion includes results of aerodynamic, structural, and thermal analyses. Also discussed is a fabrication study, which indicated that a casting development effort would be required to produce adequate material properties. This effort is currently underway, and consists of both an integral centrifugal casting approach and a bicasting technique. The test rig to be used for the turbine tests is described.

Introduction

IN the past several years, there has been a renewed interest in the radial inflow turbine, sometimes called centripetal turbine or 90° Inflow Radial (IFR) turbine. One of the most promising applications for the radial turbine is in advanced gas turbine engines in the 1–5 lb/sec airflow range. The attractiveness of the radial turbine in this application stems from its potential capability of producing higher stage work at acceptable efficiencies than its conventional axial turbine counterpart. If this advantage can be coupled with a capability to accommodate high turbine inlet temperatures, then the use of a single radial stage in lieu of multiple axial stages might result in simpler, lighter, and more economical gas turbine engines.

The potential of the radial turbine to deliver high stage work is mainly derived from its rotor shape, which is well suited stress wise for high tip speeds. A high tip-speed capability enables a turbine to operate at or near the optimum speed for high values of isentropic head drop across the stage. The tip-speed advantage might conceivably be exploited by operating at increased turbine inlet temperature (TIT), increased stage pressure ratio, or a combination of the two. In nonregenerated gas turbine engines, higher turbine inlet temperatures require increased cycle pressure ratios for optimum performance. Even in regenerated engines, which have lower optimum pressure ratios, there is a strong incentive to design for higher cycle pressure ratios to achieve a flat specific fuel consumption (sfc) characteristic for improved part-load performance. Therefore, the tip-speed advantage of the radial turbine will most likely be exploited in small gas turbines by operating at maximum TIT's comparable to those for advanced axial turbines, but with considerably higher stage pressure ratios.

Another reason that the radial turbine is being considered for small engine applications is its relatively low sensitivity to running clearances.[1,2] In small turbine sizes, the performance penalty incurred by blade-to-blade leakage through the clearance area is more severe for axial turbines than for radial turbines, and this effect becomes more pronounced as the volumetric flow decreases.

Radial Turbine Problem Areas

Most high-temperature turbine designs represent a compromise between conflicting requirements of stress, aerodynamics, and heat transfer, and the radial turbine described in this paper is no exception. There are several design areas where there are insufficient published data or previous experience, but there are two questions of primary importance to the future of high-temperature radial turbine development. 1) Can a radial turbine be air-cooled efficiently enough to ensure adequate material strength without incurring unacceptable aerodynamic and over-all engine performance penalties? 2) Can a radial turbine rotor be fabricated that is structurally adequate to meet design requirements?

Realistic answers to these questions are being sought in a radial turbine research program being conducted by United Aircraft Corporation under Contract DAAJ02-68-C-0003 with the U.S. Army Aviation Materiel Laboratories, Fort Eustis, Va., and Canadian Government Grant No. DRB 4720-18 of the Defence Industrial Research Program, Defence Research Board, Ottawa, Canada. The broad objective of

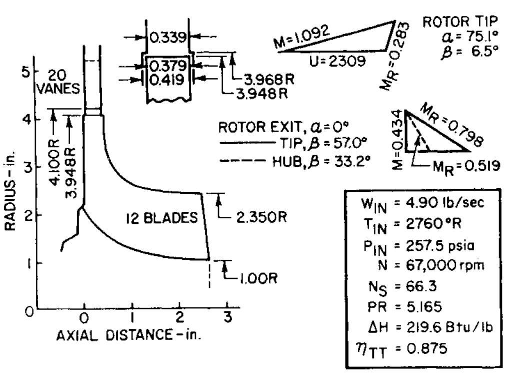

Fig. 1 Rotor velocity triangles.

Received May 27, 1969; presented as Paper 69-524 at the AIAA 5th Propulsion Joint Specialist Conference, U.S. Air Force Academy, Colo., June 9–13, 1969; revision received November 10, 1969. Part of the research efforts reported in this paper were conducted under contract with the U.S. Army Aviation Materiel Laboratories (AVLABS), the Army's aviation research center at Fort Eustis, Va., and the Canadian Defence Research Board (DRB), Ottawa, Canada, under the Defence Industrial Research Program.

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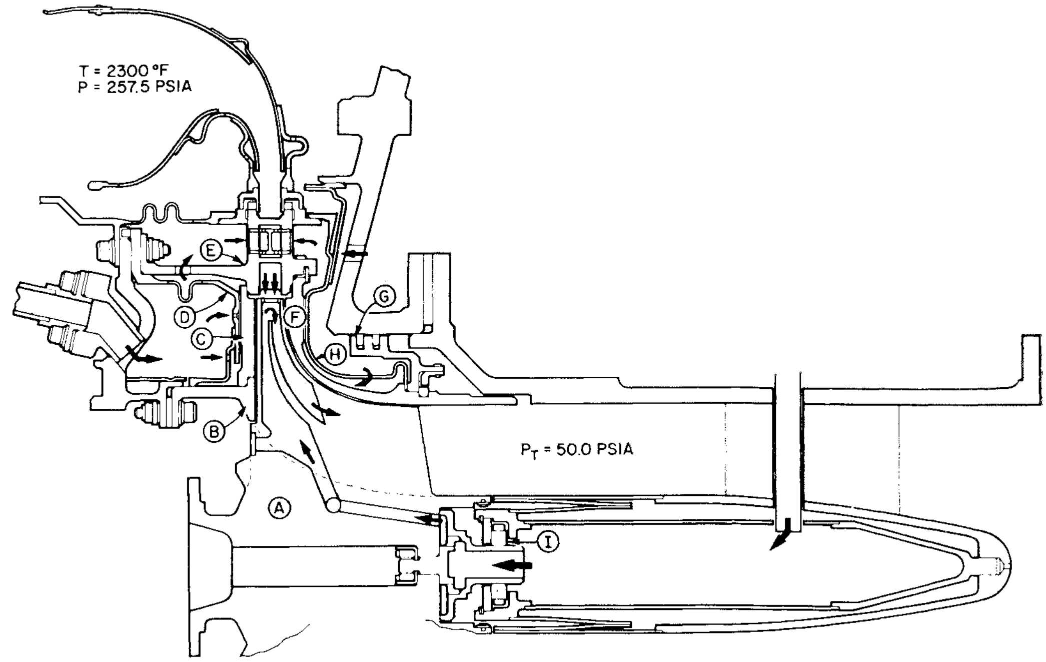

Fig. 2 Meridional view of turbine.

the program is to develop radial turbine technology to a level that will permit small-engine manufacturers to make a choice between the radial and axial turbines. This paper describes the first phase of the program, which includes the design and fabrication studies. The radial turbine design was performed by United Aircraft of Canada, Ltd. (UACL), with assistance in the heat-transfer analyses from Pratt & Whitney Aircraft, East Hartford, Conn. UACL also designed the gas generator test rig that will be used to evaluate the turbine design experimentally. Pratt & Whitney Aircraft's Florida Research and Development Center (FRDC), the prime contractor, is fabricating the turbine and test rig, and will conduct the experimental program.

Turbine Design

Hypothetical Engine

The radial turbine has been designed to drive the compressor in a hypothetical twin-spool turboshaft engine having the following sea level standard-day characteristics: airflow rate = 4.9 lb/sec; engine pressure ratio = 18:1; turbine inlet temperature = 2300°F.

The compressor was assumed to consist of a 10:1 centrifugal stage of specific speed = 80, boosted by a single axial stage. This choice fixed the rotational speed at 67,000 rpm for the gas generator shaft, the turbine inlet pressure at 257.5 psia, and, allowing for a power requirement for auxiliaries, the turbine total enthalpy drop at 219.6 Btu/lb. With a predicted turbine efficiency of 87.5%,‡ the stage pressure ratio became 5.165 for the cooled turbine.

Velocity Triangles

Over-all dimensions of the turbine were evolved as follows: since the application of this gas generator turbine was to be a general one, i.e., the direction of rotation of the power turbine, and the detailed geometries of the power turbine and the interturbine duct had not been specified, the exit swirl of the turbine was chosen to be zero, absolute exit Mach number 0.43, and rotor exit hub radius 1.0 in. This design choice, a nonoptimum one from the exclusive viewpoint of rotor design, was taken in order to demonstrate that the subsequent test results will be valid not only for an especially favorable downstream gas path, but also for designs where the downstream geometry has been compromised for other requirements.

Because of the hostile thermal environment, the aerodynamic design of the rotor was compromised because of the anticipated high stress levels in the rotor blades. At the rotor inlet the absolute flow angle was chosen at 75.1° from the radial direction, and the nominal relative flow angle (i.e., incidence) was set at +6.5° to minimize blade length. The selection of a positive incidence angle should result in a substantial performance compromise. The number of rotor blades is similarly below optimum to reduce hub stresses. Major dimensions and gas angles of the turbine are shown in Fig. 1. Allowing for the aerodynamic compromises, the final design (which was based on previous experience at UACL) agrees very well with the maximum efficiency geometry proposed in Ref. 3.

Mechanical Design

The mechanical design of the turbine and surrounding components is shown in Fig. 2. The basic components of the turbine section are the rotor, backplate assembly, nozzle, and shroud assembly. The downstream duct with its struts and centerbody are test rig parts and do not simulate any specific engine configuration. The rotor (A, in Fig. 2) is a single-piece casting of IN 100 (PWA 658), which is overhung from the gas generator shaft. The backplate assembly, consisting of the lower backplate (B), upper backplate (C), and air cover (D), is bolted to the bearing housing, as is the nozzle (E). IN 100 is also the material for the backplates, shroud, and nozzle section. The nozzle section is an integral casting that has a brazed insert in each of its 20 hollow vanes. The stationary shroud ring (F) is held concentric with the nozzle by means of radial dogs and slots. The remainder of the shroud assembly consists of a pressure balance piston (G) and air cover (H). A close concentricity thus can be maintained between the rotor and the stationary components. Both the upper backplate and shroud ring form a face seal with the nozzle casting, with the sealing force being maintained by the large pressure difference between the cooling air and the hot primary gas stream.

‡ Turbine efficiency is defined as the measured shaft horsepower divided by the isentropic horsepower available to the stage. In the calculation of isentropic horsepower, the vane coolant is included but the rotor coolant is not.

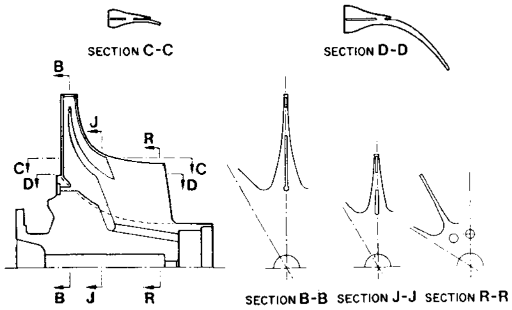

Fig. 3 Rotor.

Rotor cooling is accomplished through cast cooling-air passages in each blade. Cooling air, at 170 psia and 850°F, is supplied to the downstream end of the rotor through the centerbody, and is exhausted at the blade tips and the suction surfaces of the exducer blades after making a double pass through each blade. Leakage is minimized by a carbon ring seal (I). Total rotor cooling-air requirement is 3% of primary flow, of which 2.5% completes the double pass.

The nozzle is cooled by two separate streams of cooling air, which, after having cooled the backplate and shroud, respectively, converge in the nozzle vanes and are ejected into the primary gas stream ahead of the rotor. Both cooling streams are supplied initially at 850°F and 257.5 psia, which corresponds to compressor delivery conditions of the assumed engine cycle after allowing for pressure losses in supply ducts. Backplate cooling air is progressively admitted into the space formed by the backplate and its air cover and passes through a number of channels machined into the surface of the upper backplate, then through holes in the nozzle support cylinder and finally into the nozzle insert at a temperature of 1080°F. Shroud cooling air is similarly admitted into the space between the shroud ring and its air cover and passes through a number of channels machined into the shroud at its outer radius and into the nozzle inserts. From the insert, both airflows cool the nozzle vanes before being ejected into the primary gas stream. The total cooling air required to cool these stationary components is 6% of primary flow. However, since this air is ejected upstream of the rotor, it does contribute to the work produced and is not completely lost to the gas generator cycle.

Rotor

The choice of 12 rotor blades was based on results of cold-flow rig tests and analysis of the hypothetical engine cycle. The rig tests were carried out with an existing uncooled but similar radial turbine design, the number of rotor blades being progressively reduced from 14 to 12 to 10. Turbine efficiency at design speed and pressure ratio was found to decrease as the blade number was reduced below 14. On the other hand, each rotor blade requires a given amount of cooling air. In the engine cycle, an increase in cooling-air requirements tends to counteract the turbine efficiency improvement resulting from an increase in the number of blades. A cycle analysis incorporating the two effects indicated that 12 blades is the best compromise; the cycle sfc is only slightly below optimum while the rim load to be carried by the hub is reduced to a manageable level.

Final rotor shroud and hub contour, axial length, and exducer blade curvature were the results of aerodynamic designs that were analyzed by the method outlined in Ref. 4, assuming a linear velocity distribution in the blade-to-blade direction. Using the calculated velocity distribution on an earlier UACL rotor of known performance as a guide, the rotor was designed to minimize local blade surface diffusion. With the rather low number of blades employed, this resulted in a relatively long rotor, as is evident in Fig. 3.

Various rotor-blade cooling schemes were investigated before the design was finalized. Early preferences were for cooling passage configurations in which the cooling air was ejected at the blade tips. Tip ejection offered a simple design solution, though at the cost of low cooling effectiveness because of its single-pass feature. To achieve a better understanding of the flow pattern at the blade tip, and to explore the possibility of cooling air/primary stream interaction effects at the tip, a water flow visualization rig was constructed. This rig consisted of a nozzle ring and working rotor, operating within glass endwalls.

An optical system, driven by the rotor, permitted observation of the relative flow pattern in the rotor tip region. The primary stream was made visible by means of minute air bubbles, injected into one of the nozzle channels, while the cooling flow was colored with dye. High-speed moving pictures were taken at various conditions of head and load, both with and without tip ejection of cooling flow. Analysis of these films revealed that in the case of the low blade numbers (i.e., high blade loadings) employed, a positive incidence is induced locally at the tip of the blade, even under conditions of zero nominal incidence. The induced positive incidence produced a region of separation on the blade suction surface immediately downstream of the tip. The ejection of cooling flow, radially outward and at a high velocity, was seen to increase this region of separation. Since the design point nominal incidence of the prototype turbine is +6.5°, suction surface separation is certain to exist even without tip ejection, and the single-pass scheme was abandoned in favor of the double-pass scheme.

Aside from the previous considerations, the double-pass cooling path shown in Fig. 3 offers a better utilization of cooling air as well as the potential for boundary-layer energization in the exducer. By virtue of its large stage pressure drop, the high-work radial turbine is in the fortunate position of having a large pressure difference available between rotor cooling air and the primary stream almost anywhere on the rotor. Therefore the air can be ejected onto the rotor surface at a high-velocity upstream of a region of diffusion to energize the boundary layer and reduce the likelihood of flow separation.

In the cooled-rotor design, such a region exists on the exducer surface, over the outer portion of the blade. The aerodynamic optimum ejection location nearly coincides with the intersection of the blade surface and the plane of the cooling passage core, thus producing a natural ejection slit. This is the configuration finally adopted.

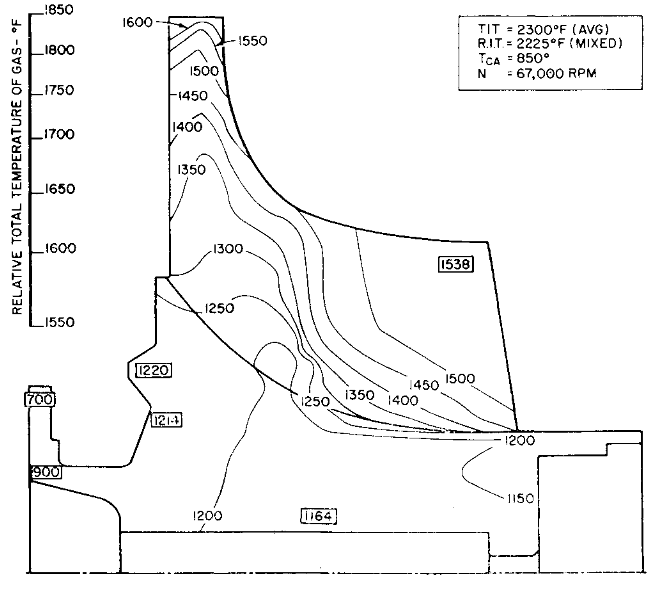

Fig. 4 Rotor temperature distribution.

Cooling air is introduced into the rotor hub at its trailing face, and is guided into the blade cavities through 12 holes. The choice of this flowpath was dictated by stress and sealing considerations. Alternative supply locations on the flanged face of the rotor hub proved less desirable because of the large seal diameter, which made effective sealing difficult. The final hole location shown in Fig. 3 permits sealing at a very low radius by means of a carbon seal (I in Fig. 2).

Thermal analysis of the rotor was complicated by the incompletely understood leakage flow pattern between the rotor blade and backplate, and by the anticipated flow separation at the blade tips (as observed during water flow test). Gas-side heat-transfer coefficients in these regions were generally calculated by assuming several probable flow patterns and settling on a coefficient which erred on the pessimistic side of the average. Rotor analysis was carried out with 250 internal and 200 surface nodes/blade segment. The resulting rotor temperature distribution is shown in Fig. 4. Metal temperatures in the cooled portions of the blade are approximately 300° lower than the relative gas temperatures.

Rotor stress analysis was carried out with a UACL technique that considers the structure as a set of discrete elements, rings, or flat plates. The stiffness of each element was calculated (effect of plasticity was taken into account), and formed into a system stiffness matrix. This matrix was solved and stresses in each element were determined from the calculated deflections.

Design point stress levels in the rotor are shown in Fig. 5 as contours of constant ratio of effective stress to ultimate tensile strength at the prevailing temperatures. The highest stress is 85% of ultimate tensile strength at the bore. The blades were designed to the criteria of 1% creep in 100 hr, and burst at 130% speed, assuming an adverse casting tolerance on blade contours. Blade stresses therefore appear quite moderate at design speed. At 130% speed, however, stresses in the nominal blade shape approach the ultimate tensile strength for the material, while the peak stress in the hub has increased to only 91% of ultimate tensile strength because of local stress relief through plastic redistribution (Fig. 6). Predicted burst speeds are 90,000 rpm for the blade and between 90,000 and 95,000 rpm for the hub, assuming that material properties meet the specification for PWA 658.

Nozzle

The integral nozzle casting has 20 hollow vanes with inserts, and a pair of end wall shields consisting of 10 segments each, welded together on assembly (Fig. 7).

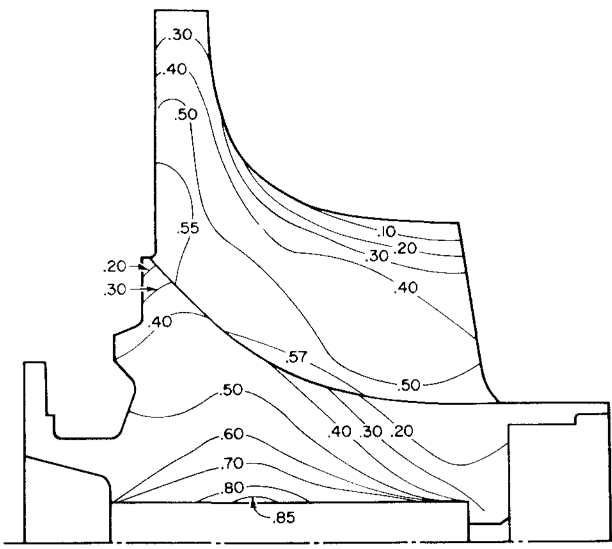

Fig. 5 Effective stress/uts for rotor at 67,000 rpm.

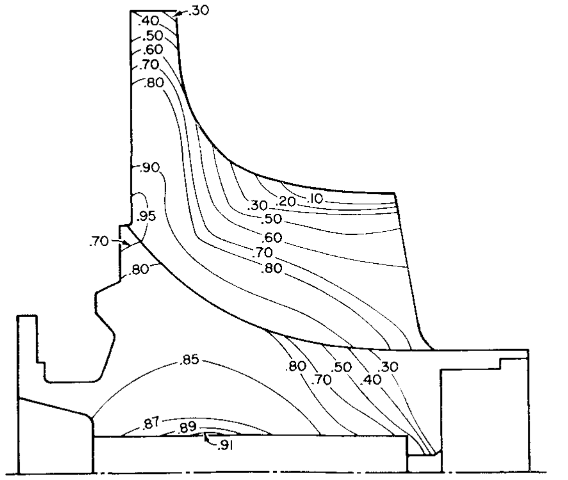

Fig. 6 Effective stress/uts for rotor at 87,100 rpm.

The choice of the number of vanes was influenced by mechanical design considerations, but was finally decided on the basis of cold-flow test results. Three nozzles, having 15, 20, and 25 vanes each, were tested with an existing rotor at UACL. All three nozzles had identical flow-channel geometries upstream of their throats (identical vane surface velocity distributions as calculated by a potential flow analysis), but the vane shapes downstream of the throats were different. Test results indicated that the 20-vane nozzle was somewhat superior to the others at the design pressure ratio of 5:1.

Cooling air is ejected on both the suction and the pressure surfaces of each vane, 2% and 4% of primary flow rate, respectively. Vane channels were designed conservatively with no diffusion on the vane surfaces. The ejection of low-energy cooling air is therefore not expected to cause separation. Nozzle platforms are insulated from the gas by heat shields that extend into each vane channel, covering the outer half of the channel. This design feature proved necessary to reduce the magnitude of the radial temperature gradient in the platforms caused by excessive cooling of the outer diameters by the combustor liner air film. The nozzle vane cooling scheme is shown in detail in Fig. 8. Cooling air, emerging from the insert, impinges onto the vane inner wall and is split into two streams. One stream flows onto the vane suction surface, the other onto the pressure surface after passing over three rows of plate fins. This complex vane shape is the culmination of numerous preliminary design studies of simpler schemes. Figure 8 shows the calculated metal temperatures midway between platforms at the design point. A parabolic combustor outlet gas temperature distribution

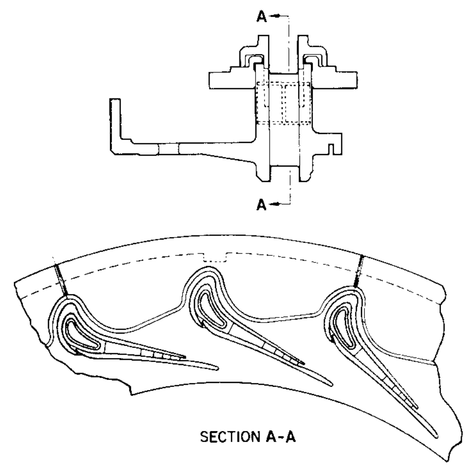

Fig. 7 Nozzle.

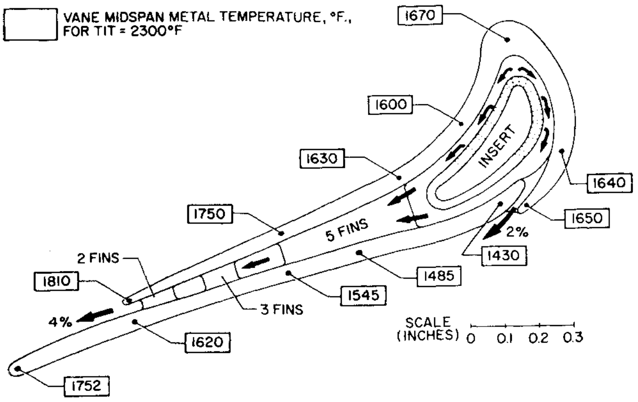

Fig. 8 Nozzle vane temperature distributions.

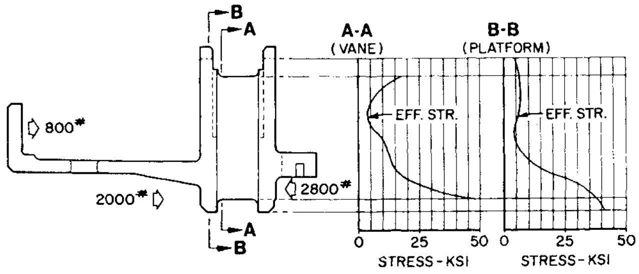

Fig. 10 Nozzle stress distributions.

was assumed, consisting of 2000°F at the platforms, a peak of 2375°F at the vane midspan, and an average temperature of 2300°F. A hot spot having a peak gas temperature of 2600°F will increase metal temperatures approximately 100° above those shown. The beneficial effect of the heat shields is shown in the average radial temperature profile in the platform, Fig. 9: metal temperature is nearly constant over the major portion of the platform, and a significant gradient exists only at its inner diameter.

Effective stress in the platform (Fig. 10) consists almost entirely of hoop stress, caused mainly by the radial temperature gradient at its inner diameter. The high effective stress at the vane trailing edge is primarily compressive axial stress, produced by a thermal gradient in the nozzle support cylinder, and by the gas loads acting on the backplate and the shroud.

Backplate and Shroud

Both the backplate and the shroud operate in an environment that tends to produce a large radial temperature gradient in these parts, causing large hoop stresses. A second, somewhat lesser stress-producing factor is the large pressure difference between the backplate and shroud cold and hot sides.

The design solution evolved for the backplate consists of selective cooling and the segmentation of the plate into two concentric rings, allowing each unrestrained radial growth. For the shroud, selective cooling is complemented by a piston that relieves the pressure load. Design point metal temperatures in these parts are shown in Fig. 11. Large local stress variations exist, but peak stresses have been held below the 300-hr stress rupture strength everywhere in these stationary components.

Fabrication Study and Casting Development Program

Concurrent with the turbine preliminary design at UACL, a fabrication study was conducted at FRDC. The objectives of this study were to select the most feasible method of fabricating the turbine nozzles and the turbine rotor, and then to evaluate the current state-of-the-(fabrication)-art relative to the design requirements.

The selection of the fabrication method to be used in making an air-cooled turbine part is strongly influenced by the material properties required and the cooling passage design. Both parts considered in this study were designed to operate with relatively high stress at metal temperatures between 1400°F and 1650°F. In this temperature range, IN 100 (PWA 658) exhibits more desirable properties than any other material that is currently available, wrought or cast. Although progress is being realized toward making IN 100 useful as a wrought alloy, it has so far been relegated primarily to cast parts. The tentative selection of IN 100 for its material properties resulted in greatly increased latitude in the design of the cooling passages, which would not be restricted to geometries that could be machined. Ultimately, this flexibility was required to achieve an adequate cooling design, and the preferred fabrication method was clearly investment casting.

During the fabrication study, three investment casting vendors submitted sample radial turbine rotors (with hollow blades), cast in IN 100; Fig. 12 shows one of the samples received. Some of the early sample rotors showed casting deficiencies such as failure to fill blade tips, blade core breakthrough, and porosity. Later castings were much improved in these respects, indicating that a geometrically sound rotor casting could be achieved with the normal development effort associated with a new design. However, results from the metallurgical tests (Table 1) told a different story with respect to the material properties. These tests included specimens taken from the rotor hub, the hub/blade junction, and the blade areas.

The average tensile strengths almost met the specification values, but because of a wide variation in tensile properties the minimum values fell below the norm. The measured creep-rupture life and ductility of the test rotors also showed

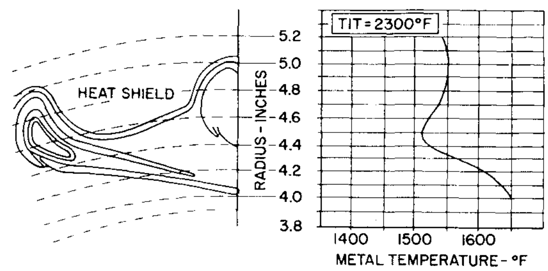

Fig. 9 Nozzle platform temperature distributions.

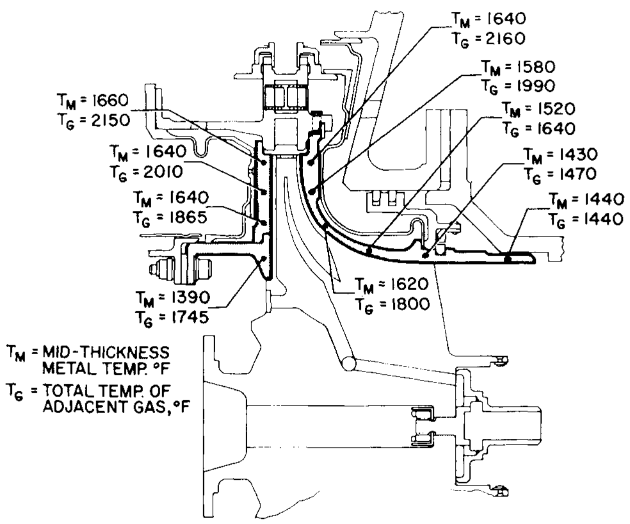

Fig. 11 Back plate and shroud temperature distributions.

Table 1 Summary of metallurgical test results

Property	Range of data	Average value	PWA 658 minimum specification
Yield strength, psi	0–118,000	93,000	95,000
Ultimate strength, psi	100,000–132,000	122,000	115,000
Tensile elongation, %	0–11	4	5
Creep-rupture life at 1400°F, hr	0–36	14	23
Elongation prior to rupture at 1400 °F, %	0–3.3	1.2	2

considerable scatter, and in addition, the average values were below the material specification. These metallurgical deficiencies are believed to be caused by undesirable grain formation, which is difficult to control because of the different cooling rates of the thin blades and thick hub section. Since the turbine rotor has been designed with the presumption that the minimum properties everywhere met the material specifications, these results indicated that a rotor casting development program would be necessary to achieve the material properties required for design point operation. Such an effort was added to the program, and is currently in progress.

TRW Metals Division, Minerva, Ohio, was selected to produce the IN 100 castings and to conduct the rotor casting development program. In this development effort, two different approaches are being undertaken simultaneously: centrifugal casting and bicasting. In the bicast (sometimes called bimetal) approach, the rotor blades are case individually, and then a hub of the same material is cast onto the blades. Material properties are expected to be improved, since the blades and hub can be poured under different conditions to promote ideal grain growth in both sections. The potential benefits that might be derived from a successful bicast program are considerable, but the attendant risks are also high. To provide for a backup fabrication technique, the more conventional centrifugal casting development effort will proceed in parallel with the bicasting effort. A destructive spin test of rotors produced by both methods will be used in conjunction with metallographic inspections to evaluate the relative merits of the two techniques, and to select the fabrication method that will be used for the test rotors.

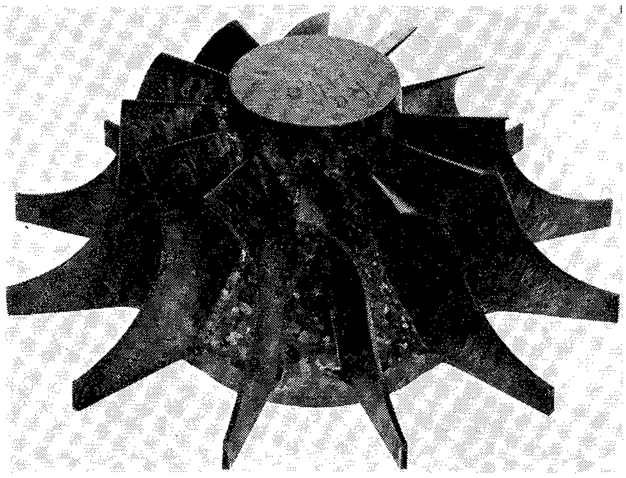

Fig. 12 Sample turbine rotor casting.

Test Program

The radial turbine design just described is part of a larger effort in which the turbine will be fabricated and evaluated experimentally in a realistic environment. In this case, realistic environment means testing at the design pressure level, pressure ratio, and temperature. Conceivably, valid

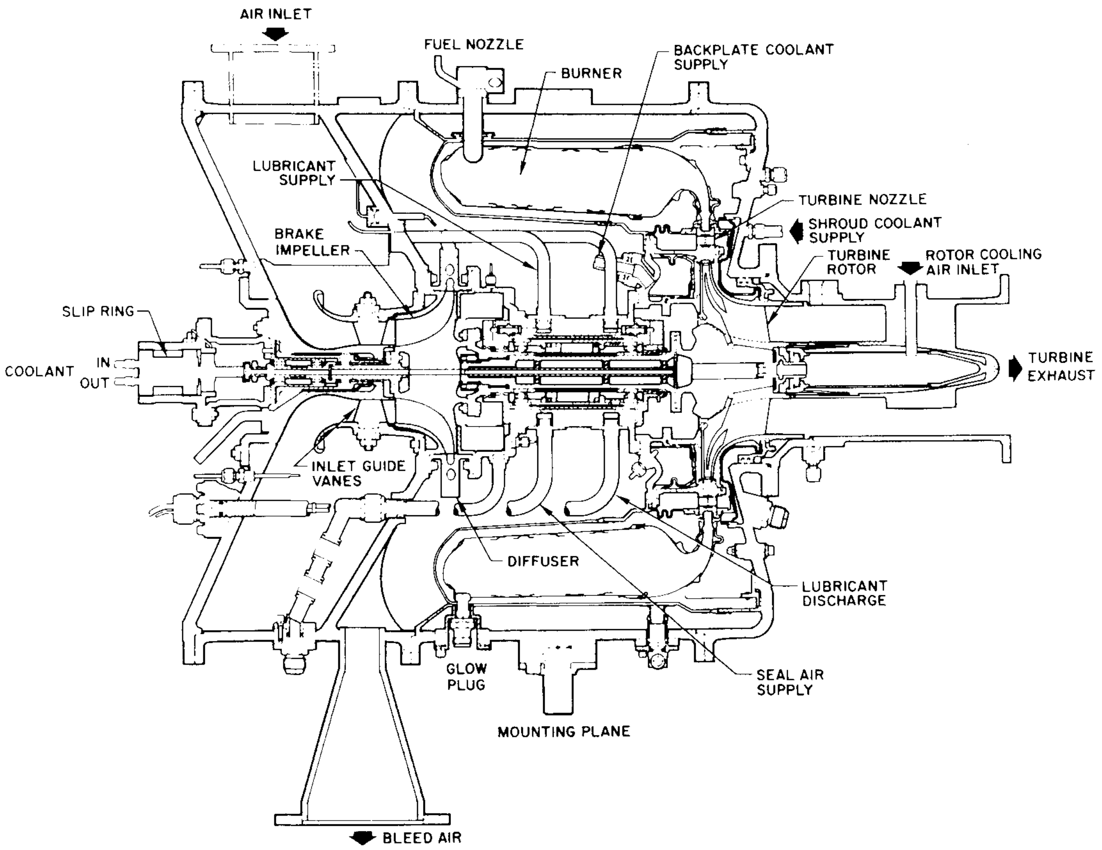

Fig. 13 Hot test rig.

performance data could be obtained at reduced pressure levels, still matching the design TIT and stage-pressure ratio. However, the convective heat-transfer coefficients would be significantly different, and one of the primary objectives of the program (cooling feasibility) would not be conclusively demonstrated.

The test rig designed by UACL to duplicate the radial turbine operating environment is shown in Fig. 13. This is a supercharged gas generator, using compressor bleed air taken from a facility gas turbine engine to provide the pressurized inlet air at 90 psia. This feature of the test rig, in conjunction with the bleed air control, makes available a wide range of turbine operating conditions. The compressor is a 3:1 pressure ratio (nominal) single-stage centrifugal design, with adjustable inlet guide vanes and two different sets of pipe diffusers to prevent compressor surge over the range of operation. The combustor is an adaption of the UACL PT6 burner. In the bearing compartment, oil is used to cool the section, lubricate the bearings, and film-damp resonant frequencies at the compressor bearing.

The test program consists of combustor evaluation followed by turbine testing. The objective of the combustor evaluation is to determine combustion efficiency at the design temperature rise, and to achieve an acceptable temperature profile. Turbine testing will consist of a total of 50-hr run time, divided between reduced temperature testing (35 hr, 1600°F TIT), design point testing (5 hr, 2300°F TIT), and cyclic testing (10 hr). The objective of the reduced temperature testing is to establish turbine performance over the widest possible range of operating conditions, and to determine the effectiveness of airfoil cooling designs. In the design point tests, the feasibility of operating a highly loaded radial turbine at high inlet temperatures will be demonstrated. The cyclic tests will vary the turbine operating conditions between design point (2300°F TIT) and half speed (1500°F TIT). These 5-min cycles should reveal any severe low-cycle fatigue problems. The performance data generated by this test program will enable engine cycle analysts to perform meaningful tradeoff studies between radial and axial turbines.

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