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Problems Related to the Development of a Gas Turbine in the 10–30 hp Class

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After reviewing the inherent difficulties involved in the design of a shaft-power gas turbine in the 10–30 hp class, the design philosophy behind a gas turbine of 15 hp nominal power is explained. Among the numerous problems encountered during the development of this machine, vibrations in the high-speed rotor-bearing configuration proved to be the most troublesome. Parallel investigations on a high-speed computer and testing of various modifications of the original concept led to a successful solution on the base of a squeeze-film suspension for the rotor shaft. Practical experience has been collected using the turbine as cold starting equipment for heavy military vehicles and as an auxiliary power unit for executive jet aircraft.

Introduction

THE DEVELOPMENT of a practical gas turbine with an output of less than 50 hp has been attempted on several occasions. The Austin Motor Company tested for some time a 30-hp engine, designed according to a governmental specification [1].[1] The Curtiss Wright Corporation displayed during the 1959 Paris Air Show a very small gas turbine to drive a 7.5-kva alternator. This engine followed a patent specification, originating from Ritzi and Dreher [2]. But as far as the author became aware, these engines never left the prototype stage.

In more recent years, several gas turbines for jet engine starting have been developed, to mention but the engines of Microturbo, Solar, AEI, and Rotax. However, they are primarily rated for starting purposes where a short burst of roughly 100 hp is asked for. Under steady running conditions, they produce some 50 hp but, being rated for high torque at low speed, their weight-to-power ratio soon becomes prohibitive. On the other hand, a real interest for lightweight gas turbines to power auxiliaries in executive jet aircraft and in heavy military vehicles has grown up since 1960. It is the purpose of this paper to explain the design philosophy and to discuss some of the problems experienced with a gas turbine in the 10–30 hp class. This engine was developed as a private venture and came to small-scale production during 1965.

Inherent Design Difficulties

The air mass flow required to deliver 10–30 shp in a gas turbine with a pressure ratio limited to less than 6:1 and equipped with the best current heat resisting materials amounts to between 0.2 and 0.7 lb/sec. Taking for granted a well-chosen compromise between speed, blade height, blade and disk stressing, and air velocities relative to the blades, this low mass flow still permits blade heights and blade Reynolds numbers only, which are at least one order of magnitude smaller than usual practice. The increase in blade friction loss with falling Reynolds number [3–6] seems to exclude axial bladings, for the compressor in any case.

Radial stages are less prone to Reynolds number effects, as is well known, because they always possess considerable extended wetted surfaces where turbulent boundary layers find time to develop. Against that stands their higher flow path curvature with tendency to increased secondary flow. Further, their blades are much shorter than the blade spacings, which stresses the predominance of tip clearance effects. These are most pronounced for very small wheels where minimum clearances are limited by manufacturing tolerances and bearing clearances. Fig. 1 illustrates typical relative tip clearance for a 15-hp and a 250-hp gas turbine compressor wheel. This effect of tip clearance upon radial compressor performance is still open to question [7–9] but a variation from 2 to 10 percent seems to affect the stage efficiency as much as a hundredfold reduction of wheel Reynolds number [7, 8, 10]. Therefore, isentropic stage efficiencies, based on total head and temperature at inlet and outlet, of between 75 to 80 percent only, are anticipated for pressure ratios round 3:1 and mass flows of the aforementioned magnitude, for both radial compressors and turbines.

As the tip speed to cope with a chosen pressure ratio and enthalpy change is roughly a constant, the rotor speed varies inversely proportional to tip diameter. The smaller a wheel, the faster it has to spin. Unbalance forces are proportional to the square of the speed for a given eccentricity of the center of gravity and rotor mass. Hence a very small rotor must be balanced to relatively far narrower tolerances and concentricities than a 300-hp rotor, for instance. Fig. 2 shows the ratio of unbalance forces against rotor weight for various speeds and eccentricities. The need for balancing equipment adapted to achieve 10^{-4} mm eccentricities is obvious. Of course, such extreme values can only be achieved by finally balancing the fully assembled rotor, running in its own bearings. Another problem associated with the high speed of miniature gas turbines is the necessity for bearings of appropriate quality. Fortunately, gyros raise similar problems, and suitable ball bearings are readily available.

Actual speed ranges for both d-c and a-c generators for aircraft equipment, which seem to be the only ones having attractively low weight, are set between 3000 and 8000 rpm. 400 cps–24,000 rpm a-c alternators would be most interesting, but they are still in a development stage. Hence gear ratios of more than

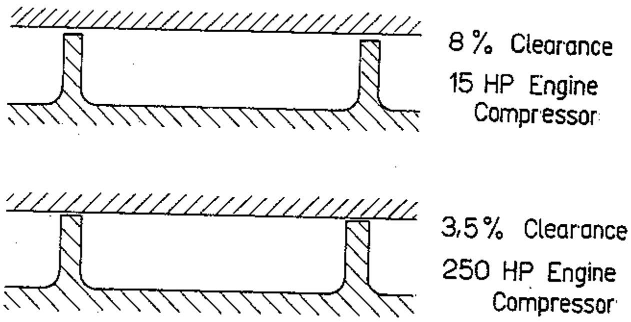

Fig. 1 Comparison of typical relative blade tip clearances for 15 hp and 250 hp gas turbine compressor wheel of similar geometry

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

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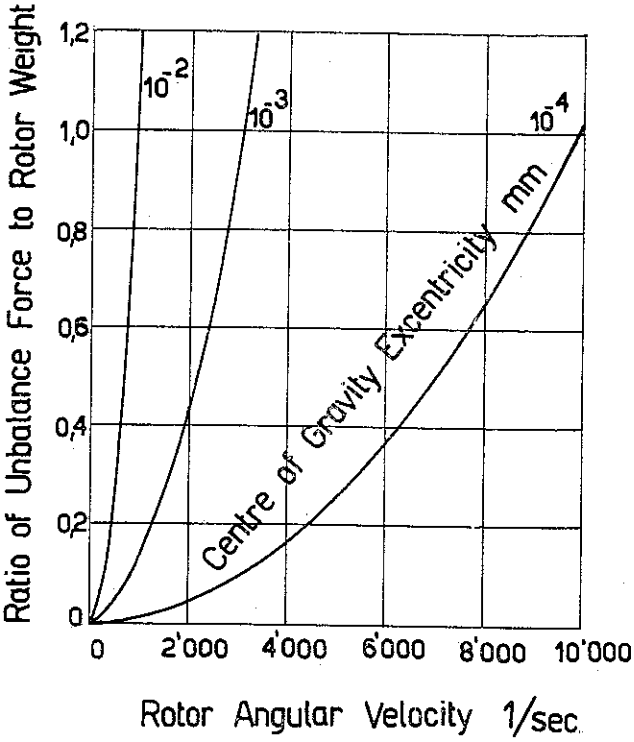

Fig. 2 Ratio of unbalance forces to rotor weight as a function of rotor angular velocity for various eccentricities of center of gravity For derivation of formula, see Appendix

10:1 between the high-speed rotor and the output drive of the anticipated gas turbine are usually needed

The auxiliaries indispensible to any gas turbine are the fuel and lubrication systems, the engine governor, the starter, and the ignition equipment. Care has to be exercised to avoid the often-practiced principle, in which the gearbox and the related auxiliaries are made up like a Christmas tree using separate shafts and bearings for every item, thus leading to power losses and superfluous weight. Unfortunately, available auxiliaries are so bulky and heavy, not to speak of cost, that they would spoil every attempt to produce a small, lightweight power gas turbine. Consequently, they have to be developed specifically to be integrated into the engine. Obviously, their overhaul life and their reliability have to be as good as any high-temperature part of the engine.

Design Concept of Gas Turbine Type GT 15

Having reviewed the inherent design difficulties of all gas turbines of low power, and accepting a limited overall efficiency, every emphasis was placed on the concept of a compact, light machine. Extensive cycle calculations on a high-speed computer indicated that an efficiency of 12 percent is the most which can be attained.

The choice of the type of combustion chamber influences the layout of a turbine probably more than all other considerations. The annular type dispenses with scrolls and transfer ducting and leads to a fully symmetrical design resulting in balanced thermal expansion and stress The space available between compressor and turbine wheel is copious, and the thermal loading of the flame tube can be limited to conservative values with benefits on its maintenance and life. The distribution of the very small fuel flow to a multitude of injectors around the circumference of the combustor promotes inevitable matching difficulties between them. In addition, there are manufacturing problems with nozzle orifice diameters of less than 0 01 in The obvious answer is rotating atomization which has been well known for 50 years in oil burning, and which has been pioneered on gas turbines by Turbomeca An additional advantage of rotating atomization lies in the fact that a low-pressure feed of the fuel into the spinning device is fully sufficient. To throw the fuel from the high-speed rotor shaft into the appropriate region of the combustion chamber, the shaft has to be uncovered over a certain length This entails either a completely overhung turbine wheel or a bearing at its exhaust end. In the latter case, the outer structure of the combustor and the turbine nozzles would have to support this bearing by means of spokes protruding through the exhaust pipe Cooling of this bearing would be imperative

An overhung turbine wheel was preferred, with ball bearings on either side of the compressor wheel They are lubricated with fuel This allows for positive cooling of the inner bearing, at the same time keeping churning losses relatively low.

The gearbox can be adapted for output speeds between 6000 and 24,000 rpm by selection of appropriate gear ratios One single layshaft can supply the necessary reduction ratio. A double gear pump and a tachogenerator ride on the layshaft. The primary of these pumps feeds fuel from the tank into a distributor, where a relief valve keeps the pressure constant, irrespective of fuel flow The various lubricating jets are supplied out of the distributor. A further flux from the distributor, after being regulated in a solenoid-operated valve, enters the high-speed shaft from where it is injected into the combustion chamber. The secondary pump scavenges the gearbox and feeds surplus fuel back to the tank. Thus the fuel acts both as fuel and as lubricant Heat generated in the high-speed bearings and the gears is carried back to the tank serving as a heat sink Intended as an auxiliary power unit (APU), the gas turbine will always be linked with a large prime mover with corresponding tankage. An individual cooling system will not be necessary

The solenoid-operated fuel valve receives its governing current from an electronic engine governor with an idle and a full-speed

Fig. 3 Showing comparative size of gas turbine type GT-15. Turbine section with exhaust pipe on the left, gearbox on the right Dry weight of engine, fully equipped, 22 lb. Rated power 15 hp at sea level, 25 C ambient temperature

setting Any deviation from the speed setting resulting from a load change is sensed by the tachogenerator and corrected accordingly. The engine is protected against overspeed by an independent section of the governor and against overtemperature by a thermostat switch in the turbine exhaust, both of which act on a second solenoid valve in the feed line of the primary pump.

Starting of the engine can be performed by hand with a rope-type starter, alternatively by starter windings in the driven d-c generator, or by a cartridge which impinges its hot gas jet on a small turbine wheel on the output shaft. Ignition during starting is provided by a HE-ignition system of suitable size.

Development History of GT-15 Engine

Design studies started in the spring of 1961 in response to inquiries for an APU for an executive jet aircraft. The first prototype was ready for testing during May, 1962. Development during the next two years progressed slowly but steadily. Numerous difficulties had to be overcome, which will be described in some detail. During 1964, most of these problems were solved satisfactorily and endurance running with several prototypes became possible, paralleled by field testing in various applications.

The design concept of the GT-15 engine is based on practical experience with several gas turbines going back as far as 1947, when Saurer Ltd. built and tested a purely experimental engine, mainly to become familiar with a new technology. A second lightweight engine, already using the fuel as lubricant, originally fitted with a single can combustor and rebuilt for a truly annular type with multiple injectors, led to the forerunner of the GT-15. This forerunner of 50 hp nominal power incorporated rotating atomization for the fuel and an overhung turbine wheel. It gave such encouraging results that the design, manufacture, and testing of the GT-15 (see Figs. 3 and 4) were decided.

Tests with the first prototypes soon proved that it was impossible to run the engine from idle speed, set at 55,000 rpm, safely up to full speed. Heavy rubbing of the high-speed shaft against the seals adjacent to the bearings inevitably led to seizure. At first, this evidence of shaft bending vibration was not at all understood. Calculations of the ball-bearing stiffness (Fig. 5) showed much lower values than initially assumed during the design calculations of the critical shaft speeds. The behavior of the rotor-bearing system, allowing for variable stiffness, was subsequently programmed for the high-speed computer. The methods used are similar to those described by Sevcik [11]. They showed a wide spread of critical bending speeds, especially for the second order, depending on bearing and support stiffness (Fig. 6). The calculations further showed roughly 100 times stronger deflection of the shaft when running through the second order than when running through the first or the third orders. Probably biased by the design calculations, the rubbing traces on the shaft had initially been supposed to be due to the first bending critical mode, while they now proved to result from the second mode. To complicate the analysis of the evidence, this second order was split into several resonance peaks.

A turbine was fitted with a dummy rotor of the same dynamic properties and stiffness as the real rotor and equipped with both clearance and vibration pickups. Consequently, the shaft orbit and the extent of the deflection between the bearings and at the

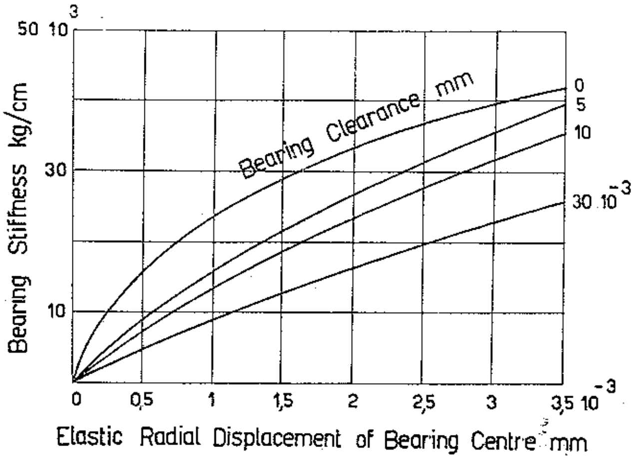

Fig. 5 Ball bearing stiffness perpendicular to bearing axis, as a function of elastic radial displacement of center of bearing and of initial bearing clearance. (Bearing type Fafnir MM 9103 K—MBR, size 17 × 35 × 10 mm with nine balls of $^7/_{32}$ in. dia)

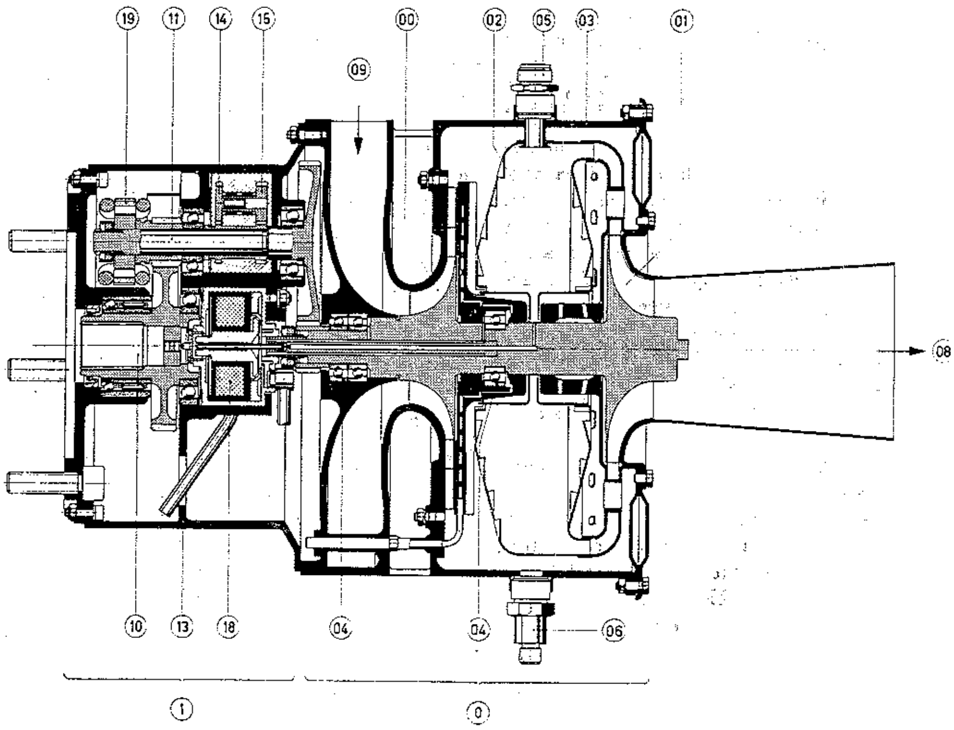

Fig. 4 Schematic sectional drawing of GT-15 engine. (0) Turbine section, (1) gearbox section, (00) radial compressor, (01) radial turbine, (02) combustion chamber, (03) turbine casing, (04) bearings, (05) igniter plug, (06) drain valve, (08) exhaust pipe, (09) compressor intake, (10) output gear, (11) layshaft, (13) gearbox casing, (14) primary pump, (15) secondary pump, (18) regulating valve, (19) tachogenerator

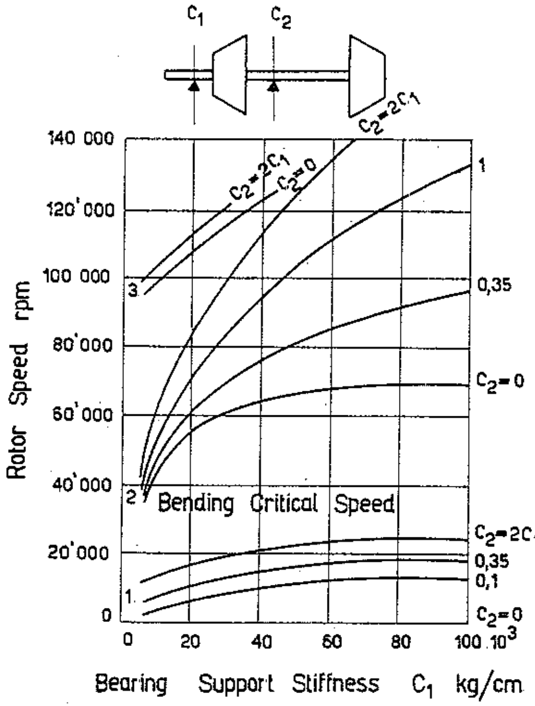

Fig. 6 First, second, and third critical bending speed as a function of bearing and support stiffnesses C_1 and C_2 for GT-15 rotor configuration (abscissa C_1, parameter C_2)

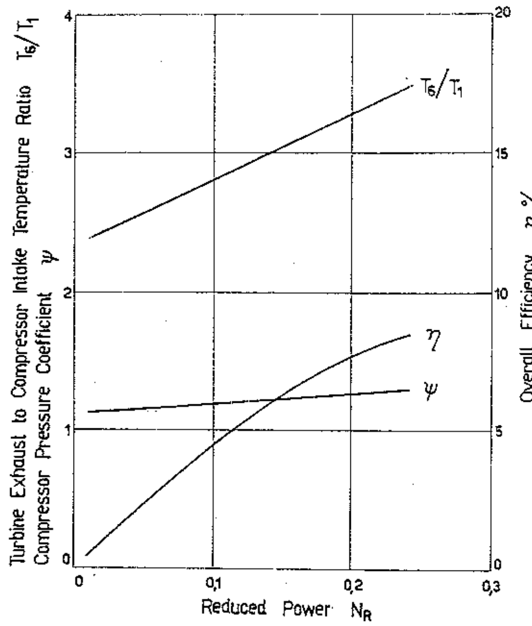

Fig. 7 Compressor pressure coefficient, ratio of turbine exhaust temperature to compressor intake temperature and overall efficiency of GT-15 gas turbine against reduced power. For definitions of coefficients, see Appendix

turbine end could be measured throughout the speed range for various bearings and bearing support modifications. The pickups on the dummy turbine permitted the identification of the vibrational modes of the real engine on the basis of the corresponding vibration signals. Supporting the inner bearing in a built-in squeeze film damped suspension, balancing the rotor according to the shape it takes in its second bending critical mode, and stiffening the bearing at the intake end of the compressor, proved during the course of tests to be the solution of the problem. Thanks to these measures, the high-speed shaft generates very little vibration throughout the full range of speed and load conditions.

Initially, the feed lubricant to the inner bearing came from two diametrically opposed holes in the turbine shaft. After several hours running, the shaft developed a slight curvature perpendicular to the plane of the holes. Increasing unbalance arose with corresponding rubbing of the bearing seals and shaft seizure. Three equally spaced holes cured this trouble. A similar difficulty was produced by the fuel return-flow hole in the bearing support. This hole had been pierced radially through the bearing support structure. After prolonged running, the bearing support hub lost its concentricity with the bearing at the compressor intake side and again led to the shaft seals rubbing and seizure. A redesigned bearing support with a spiral-wound return flow channel proved to remain stable under all running conditions.

Cycle calculations executed before the detailed design were based on component test results of larger units. It was therefore important to determine actual component behavior as soon as the hardware became available. The compressor characteristic was established by motoring an engine without combustion chamber or turbine but fitted with a suitable intake and exhaust duct for the compressor. The ducting was equipped with pressure and temperature pickups and a throttle valve. Whereas pressure coefficients proved to be slightly inferior to design values, the power coefficients versus flow coefficients followed the same dependence as on the larger compressors. Gearbox and high-speed bearing losses were also determined by motoring a fully equipped gearbox with a dummy high-speed shaft. Measurements on the complete turbine comprised compressor mass flow, intake temperature and pressure, compressor discharge pressure, output torque, layshaft speed, fuel consumption, turbine exhaust temperature, and back pressure. By a trial and error method and suitable introduction of the compressor characteristic and the mechanical losses, these measurements permitted the computation of the turbine characteristic and the combustion efficiency. Typical examples of compressor pressure coefficient, temperature ratio between intake and exhaust temperature, and overall efficiency, all plotted against reduced power, are illustrated in Fig. 7. The turbine characteristic which was deduced thanks to the aforementioned method fell close to the design assumptions. Improvement of the compressor characteristic and reduction of the mechanical losses is however still desirable.

In order to keep the fuel consumption and the exhaust temperature of the turbine at full load as low as possible and, on the other hand, to run the group at a safe distance from compressor surge during load transients and acceleration from idle to full speed, the turbine nozzle width was chosen on the basis of tests with several nozzle widths. Further, the electronic engine governor was fitted with an acceleration-limiting device which operates only on running up from idle to full speed. The range of pressure coefficients obtained during a variation from no load to full load is shown in Fig. 7.

Field Test Experience

In a first series of evaluation tests as an auxiliary power unit for heavy military vehicles, where it drives an a-c generator and its exhaust heat is used to preheat the cooling system of the main engine, cold starting tests with a variety of fuels have been performed down to −40 C (−40 F). The starting was done by hand or by cartridge. The fuels used were JP1, JP4, and a winter diesel fuel of German manufacture. JP1 and JP4 brought no problems but the winter diesel fuel, with a pour point of −60 C, could not be sufficiently atomized at −40 C to initiate the ignition. Comparative tests on a blended fuel with the same viscosity of 20 cst at room temperature as that of the winter diesel fuel at −40 C showed again that it was impossible to start the engine by hand, whereas the cartridge starter performed satisfactorily. The cartridge starter drives the turbine almost up to idling speed within about 3 sec. In the case of hand-starting, the turbine has to light up at a very low speed to assist the operator in stepping it up to self-supporting speed in about 12 sec. It became clear that poor atomization at such a

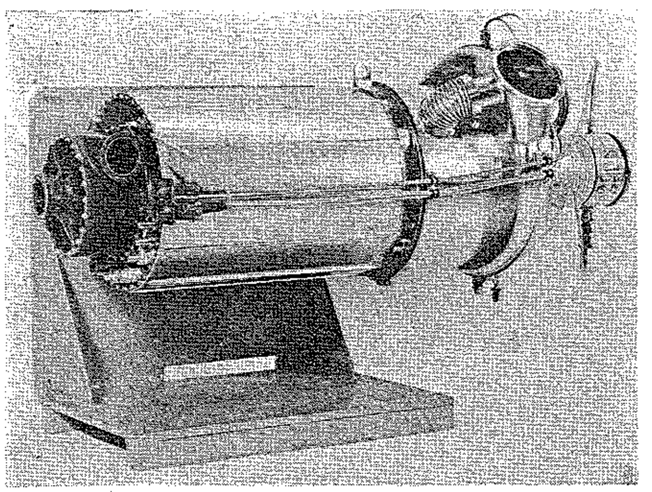

Fig. 8 General view of auxiliary power pack. Gas turbine GT-15 at far right end, d-c generator in center and blower G-15 at left end, all fully, shrouded. Connections for fuel supply, d-c power, unit governing, and fire prevention concentrated on bulkhead at blower end. Overall length 43 5 in.; overall weight 125 lb. Generator GE2CM782

low speed prevented positive ignition. A small auxiliary fuel injector, mounted on the combustion chamber parallel to the HE-igniter plug and fed by a hand-operated pump, gave satisfactory ignition and allowed hand-starting down to −40 C even with this viscous type of fuel

During the development of the primary objective to produce an APU for executive jet aircraft, the requirement arose to generate not only d-c power but to be able to run the climatization system of the aircraft before operation of the main engines and to boost the restricted power of the GT-15 to start the main engines. Using the same gearbox and compressor as on the gas turbine, a blower was built which can deliver pressurized air Mass flow and pressure ratio are adjusted according to the climatization system by means of suitable gear ratios and compressor wheel diameters, always, of course, within the power envelope of the GT-15. The d-c generator is provided with shaft ends on either side, so that the gas turbine and the blower can both be mounted on the generator. Thus a very compact unit became available which was fully shrouded and ventilated according to FAA rules, Fig. 8 By proper parallel wiring of the d-c generator and the aircraft battery and appropriate governing of the system, two types of jet engines in the 3000-lb-thrust class could successfully be started

Conclusions

Two principal conclusions may be drawn from the development history of the GT-15 engine: The cost involved in working the prototypes up to production finish, compared on a weight basis and for the same prototype numbers used, is very similar to the development expenditures for gas turbines in aircraft propulsion [12, 13] This seems to be fortuitous, bearing in mind the relative simplicity of the GT-15 compared with the design of a modern jet or turboprop engine. But even with its background on hand, because of its size, the development of the GT-15 engine had to be worked up from the very bottom for every detail, including all manufacturing methods.

Extrapolations of overall efficiencies and weight-to-power ratios on the basis of already existing power plants, a frequent theme of survey papers, seem to be risky (Fig. 9) Any improvement of technology helps to evolve better design concepts, by which the assumptions for such extrapolation are changed.

Acknowledgment

The author is indebted to Adolph Saurer Ltd , whose management made possible the development of the GT-15 engine and who gave permission to present this paper. To bring this engine to production finish required the concentrated effort of a small, closely knit team, to which full recognition is here given.

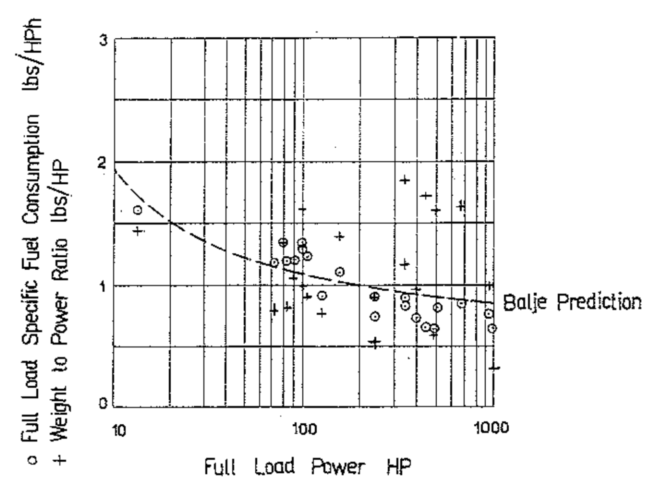

Fig. 9 Full-load specific fuel consumption and weight-to-power ratio for modern single-shaft gas turbines without heat exchange, data compiled from 1965 Gas Turbine Catalog, compared with predicted specific fuel consumption by Balje [14] (curve for turbine inlet temperature 816 C)

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APPENDIX

Unbalance forces F are proportional to rotor mass W/g, center of gravity eccentricity e, and to the square of the rotor angular velocity ω; hence

$$\frac{F}{W} = \frac{e\omega^2}{g}$$

Compressor pressure coefficient is defined as isentropic compression enthalpy change divided by the stagnation enthalpy of the rotor tip speed, all in coherent dimensions

$$\psi = 2\frac{\Delta i_s}{U^2}$$

Reduced power is defined as the power divided by the compressor intake enthalpy and the compressor weight flow, all in coherent dimensions

$$N_R = \frac{N}{\dot{G}c_pT_1}$$

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