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Design Study of an Advanced Concept Simple Gas Turbine for Possible Use in Low Emission Automobiles

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United Aircraft Corporation studied the potential costs of various possible gas turbine engines which might be used to reduce automobile exhaust emissions. As part of that study, United Aircraft of Canada undertook the preliminary design and performance analysis of high-pressure-ratio nonregenerated (simple cycle) gas turbine engines. For the first time, high levels of single-stage component efficiency are available extending from a pressure ratio less than 4 up to 10 or 12 to 1. As a result, the study showed that the simple-cycle engine may provide satisfactory running costs with significantly lower manufacturing costs and NO_x emissions than a regenerated engine. In this paper some features of the preliminary design of both single-shaft and a free power turbine version of this engine are examined. The major component technology assumptions, in particular the high pressure ratio centrifugal compressor, employed for performance extrapolation are explained and compared with current technology. The potential low NO_x emissions of the simple-cycle gas turbine compared to regenerative or recuperative gas turbines is discussed. Finally, some of the problems which might be encountered in using this totally different power plant for the conventional automobile are identified.

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Design Study of an Advanced Concept Simple Gas Turbine for Possible Use in Low Emission Automobiles

H. C. EATOCK **M. D. STOTEN**

The use of gas turbines has frequently been identified as a potential solution to the increasing demand for a major reduction of automobile exhaust emissions (1).[1] The National Air Pollution Control Administration (NAPCA) of the Department of Health Education and Welfare[2] sponsored a manufacturing cost study of automobile gas turbines by United Aircraft Corporation (2, 3). The overall costs and feasibility of several types of gas turbine were compared with the current reciprocating engine. As part of the study, an advanced concept, high-pressure-ratio, simple-cycle (non-regenerated) engine was examined in considerable detail.

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

[2] Since replaced by the Office of Air Programs (OAP) of the Environmental Protection Agency (EPA).

In this paper, features of the preliminary design of both single-shaft and free-power-turbine versions of the advanced engine operating at 10 or 12:1 pressure ratio and peak turbine inlet temperatures of about 2000 F, are examined. The major assumptions employed for performance extrapolation are explained and shown to be generally consistent with current UACL technology. The potential for low NO_x emissions of the simple-cycle gas turbine as compared to regenerative/recuperative gas turbines is discussed. These advanced simple-cycle gas turbines are based on an extrapolation of UACL's perhaps unique radial compressor and turbine design capability (4-8). This capability has resulted from a continuous program of research and engine application starting in 1961. Much of this research has been jointly sponsored

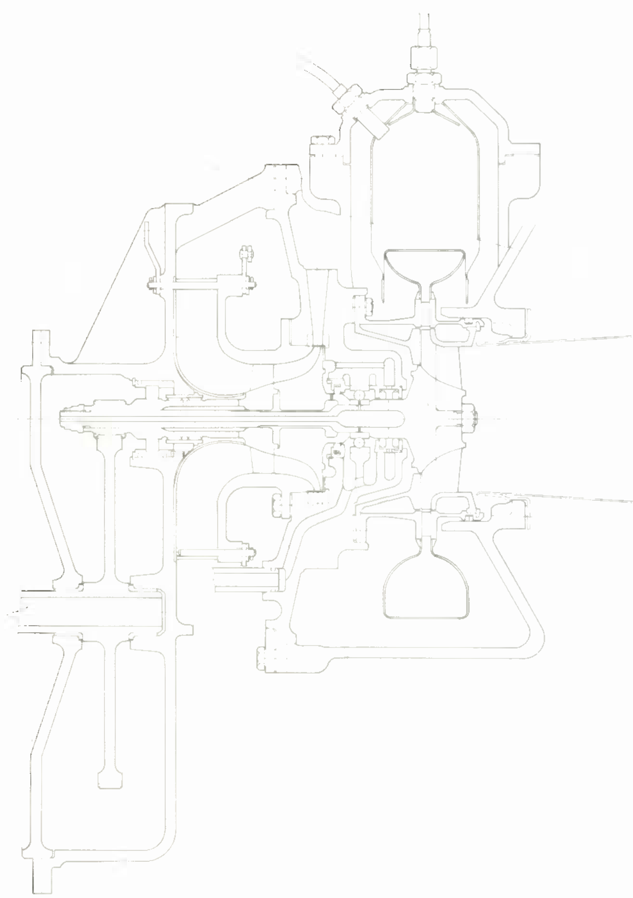

Fig.1 Layout: simple-cycle, single-shaft engine (SSS-12)

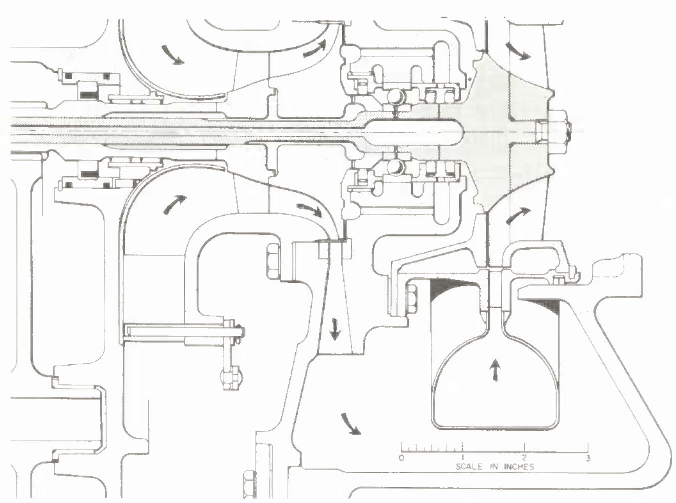

Fig.2 Gas path, SSS-12

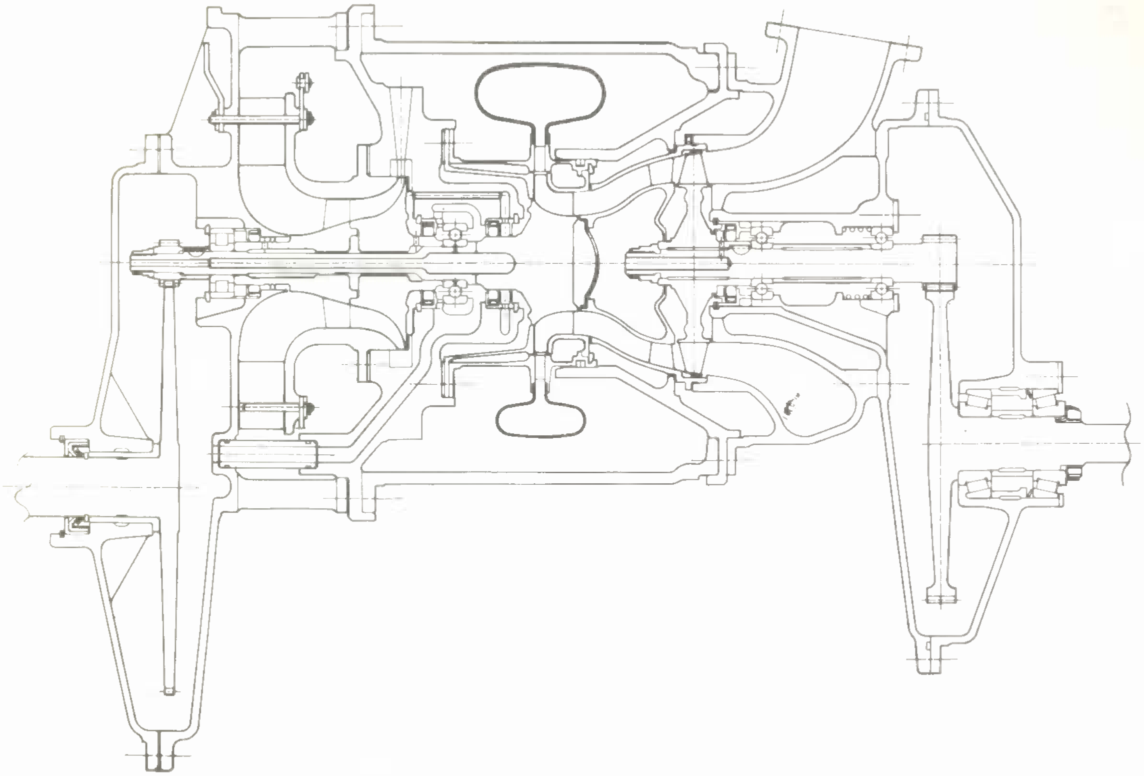

Fig.3 Layout: simple-cycle, free turbine engine (SFT-12)

by UACL and the Canadian Defence Research Board (DRB) in a manner similar to the Internal Research and Development (IR&D) programs of American Defense Contractors.

The manufacturing cost study showed, as expected, that the simple-cycle engine has substantially lower manufacturing costs than a regenerated engine. It also showed that running costs of the simple-cycle engine employing advanced components can be low enough that the overall cost of owning such an engine was the best of the gas turbines studied and was comparable to future reciprocating engines. This conclusion opens up the possibility of a different route in the development of the automobile gas turbine, one which compared to the regenerated or recuperated gas turbine offers lower first costs and reduced nitric oxide emissions.

The technology assumed for engine performance and cost predictions was extrapolated using the following guidelines: Manufacturing costs were estimated for the two rates of 100,000 and 1,000,000 units per year. Significant engine production was assumed to start in the 1975 to 1980 time frame. Component technology was predicted as the level demonstrable on test rigs by December 1975 largely on the basis of orderly development gains from on-going or currently planned programs.

ENGINE DESCRIPTION

The single-shaft engine and the gas generator of the two-shaft (free turbine) version share a common layout of the major components (Figs. 1, 2, and 3). Both have an annular, radial inflow intake delivering air to the same high-pressure ratio radial compressor and its pipe diffuser. From the diffuser, air passes into a plenum chamber which encloses a single can combustor and turbine entry scroll duct. The compressor is driven by an uncooled radial turbine which also provides the shaft power on the single-shaft engine. On the two-shaft engine, this is followed by an axial power turbine with an offset exhaust duct.

Main rotor shafts are mounted on two antifriction bearings straddling the compressor, and most of the static structure is made from iron castings.

PERFORMANCE

The key to the advanced concept simple-cycle engines is the availability of small-scale radial compressors and turbines which can operate at good efficiencies over pressure ratios from about 4:1 to 12:1. Operated near 12:1 and at 1900 or 2000 F turbine inlet temperature, high

Table 1 Standard-Day Engine Performance

ENGINE TYPE	SSS-12	SSS-12	SSS-10	SSS-10	SFT-12
POWER SETTING	RATED	IDLE*	RATED	IDLE*	RATED
Compressor Speed - rpm	130 K	94K	124.2 K	87 K	130 K
Power Turbine Speed - rpm	–	–	–	–	90 K
Output Drive Speed - rpm	13 K	9.4 K	6.2 K	4.3 K	13 K
Airflow - lbs/sec	1.06	0.47	1.16	0.55	1.06
Inlet Pressure Losses - %	0.8	0.15	1.0	0.22	0.9
Prewhirl Angle (IGV Setting)	0	30°	0	30°	0
Compressor Pressure Ratio	11.1	3.71	10.22	3.5	11.1
Compressor Efficiency, Total to Static - %	78.5	72.052	74.5	77.2	78.5
Combustor Static to Total Pressure Drop $\Delta P/_P$ - %	2	2.5	2.0	2.74	2
Combustor Efficiency - %	98	98	98	98	98
Turbine Inlet Temperature - °F	1960	880	2000	777	1950
Overall Turbine Pressure Ratio	10.1	3.46	8.96	3.22	10.1
Overall Turbine Efficiency, Total to Total - %	88.3	86.18	88.3	86.6	88.3
Exhaust Pressure Losses - %	6.32	4.35	9.6	5.22	6.3
Parasitic Power Losses - hp	3.8	1.61	3.9	1.558	3.8
Engine Gearbox Loss - hp	4.7	1.156	7.33	1.125	4.2
Leakage Airflow - %	1.5	1.5	1.5	1.5	1.5
Net Output - hp	150	0.0	151.8	0.0	150
Fuel Flow - lbs/hr	76.36	11.89	86.3	12.08	76.2
BSFC - lbs/hp/hr	0.51	-	0.57	-	0.51

* 'Idle' conditions are representative only as setting can be widely varied during development to optimize between fuel flow and response time.

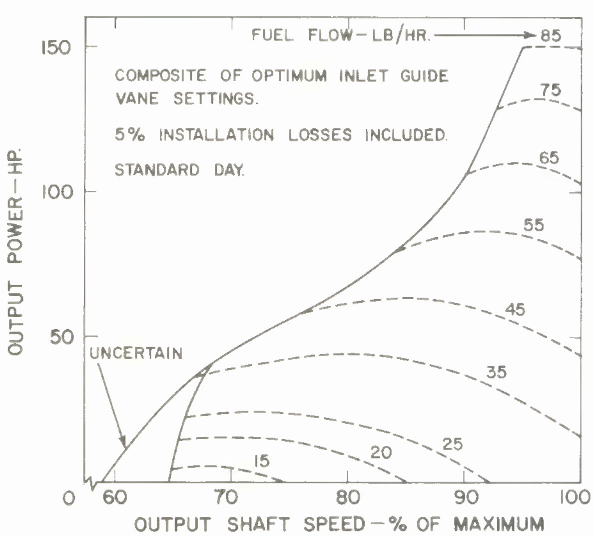

Fig.4 Engine performance estimates, SSS-10

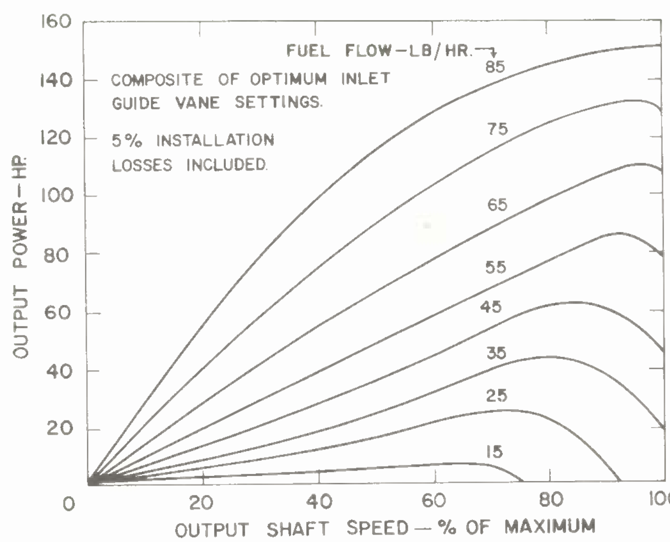

Fig.5 Engine performance estimates, SFT-10

power, combined with very good fuel consumption, is obtained. When operated at 4 or 5:1 pressure ratio for low power and idling, fuel consumptions are obtained which are respectable, although not so good as to be expected from either reciprocating engines or regenerated/recuperated gas turbine engines. The advanced radial components also operate at relatively high specific speeds which is necessary for maximum efficiency. This leads to high rotating speeds and very small size with re-

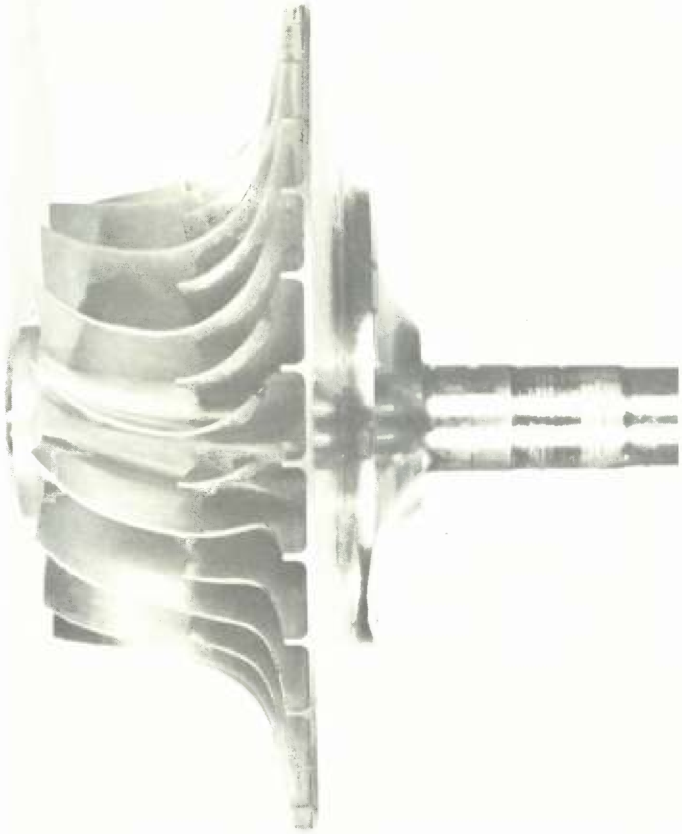

Fig.6 Centrifugal impeller — experimental

Fig.7 Centrifugal impeller — production

sultant low weight, and modest requirements for expensive superalloys.

Table 1 is a compilation of standard-day performance at rated power and at idle of both the SSS 12 and SFT 12 versions of the advanced high-pressure-ratio engine. The rated power point shown for SSS 12 and SFT 12 is aerodynamically derated so that the engines are flat rated at 150 hp to about 85-deg ambient temperature.

A version of the basic design was revised to include both a modest derating of compressor pressure ratio for greater emphasis on low power operation and significantly greater installation losses and penalties. This more conservative engine is called SSS 10 and is also tabulated. Power, speed, and fuel flow curves for SSS 10 and SFT 10 are shown in Figs. 4 and 5, the SFT 10 figures being simply scaled from SFT 12 values.

COMPONENTS

Radial Air Intake and Inlet Guide Vanes

The intake is a full annulus radial inflow design, with the structural loads carried across it by a cast truss. The full radial-inflow intake provides good flow area for the necessary effective but low-loss intake filtration. It also provides very low intake pressure drop coupled with excellent freedom from aerodynamic flow distortion problems. Variable inlet guide vanes have been included, although it is not certain their use would be mandatory. They do provide a powerful development tool for adjusting and optimizing engine operation, however. For example, they can be used to cater for compressor surge line kinks or dips to aid in obtaining good acceleration and in setting engine idle for a suitable compromise between fuel consumption and initial acceleration. Being cold and mounted between plane radial surfaces, these vanes would not suffer from the cost, mechanical unreliability, and performance penalties typical of variable hot geometry, such as power turbine nozzles. They were considered to be manufactured either as individual plastic moldings or as cuttings from an aluminum extrusion. Vane icing as a common problem for aircraft gas turbines, but is unlikely here because of intake filtration, low air velocities over the vanes, and general heating of the intake structure.

Compressor

This is the most advanced component in these engines and is the key to the whole concept of advanced but simple engines. A single centrifugal compressor able to operate in very small scale at high specific speed ($N_s \geqslant 90$)[3] and good efficiency over a pressure ratio range from about 3 or 4:1 to 12:1 offers great promise of obtaining an engine which is economical to manufacture and relatively small, light, simple, and rugged.

The compressors used to project the engines here are based on research programs jointly funded by the Canadian Defence Research Board and UACL. This work started in 1961 and has continued very actively since then. Current work is divided between two programs, one of which studies small-scale centrifugal compressors with referred airflows of 1.5 to 2.5 lb/sec and maximum speeds of 80 to 94 K rpm. The second deals with larger centrifugal or combined axial-centrifugal compressors typically with referred airflows up to 15 lb/sec and maximum speeds of 37.5 K rpm. Early results from these programs have been fairly widely reported (4-6). The technology has been used and proven on engines from 1963 on. This includes the existing PT6-A20, A27, A34, L73, T3, T400, and the JT15D-1. It also underlies the engines under development (PT6 A41, JT15D-4) as well as projected future engines. The technology has received independent confirmation by the ATEGG-II program conducted for USAF by P&WA at East Hartford (6) and by the ST9 engine (advanced turboshaft engine of 1500 shp) currently under development by P&WA's Florida Research and Development Center (FRDC) for the U. S. Army.

The core of this technology is threefold:

1 Impeller analysis and aerodynamic design is conducted using a number of sophisticated programs which provide three-dimensional compressible-flow information including secondary flows and shroud friction losses. Boundary layers on all four passage walls are computed. The resultant impellers show rather complex passage shapes (as in Figs. 6 and 7) which are very far from being arbitrary. These impellers typically achieve their design point performance without modification or development.

2 The aerodynamic design, stressing, and manufacture of the impeller with its complex passage shapes is fully integrated. Aided by a joint DRB-UACL program of mechanical research which has been underway since 1963, extensive programs using finite element techniques have been developed. Both membrane and bending behavior of the complex blade shape is simulated for elastic stress and dynamic analyses by three-dimensional triangular plate elements. Linearly-varying-strain ring and plate elements are used in the elastic and plastic stress analysis of the hub. Both blade and hub analyses consider thermal, centrifugal, and external loads. The basic aerodynamic design and manufacturing requirements are input to these stress programs which then compute the blade shape definition for minimum stress levels. The output is then applied to further computer programs which define the cams to the control four axes of a special Hayes machine on which the impellers are flank milled. The cams themselves are made on numerically controlled machines.

3 The third feature of these advanced-technology compressors is the use of the unique pipe diffuser which was invented at UACL and developed under the DRB-UACL research program. The pipe diffuser, Fig. 8, is particularly worthwhile for pressure ratios of 5 and above. It provides naturally a swept leading edge to handle transonic or supersonic flow, this edge also functioning as a vortex generator for off-design performance promoting stability over a wider flow range. The impeller exit flow angles become progressively more tangential from the center toward either shroud or back-face wall. This angle variation tends to be automatically matched by the metal angle of the ridge or "breakwater" which is also at a lower diameter and more tangential angle near either wall. The throat diameter of the individual pipes can be readily machined, permitting low-cost control of the circumferential variation of throat area from pipe to pipe as well as the total area. For high-pressure components, this factor alone is of great importance as the difference between an optimum match at 11 and 12 pressure ratio is typically only $5^1/2$ percent in area.

Fig. 9 shows the compressor map projected in August 1970 for rig demonstration in December 1975. A zero prewhirl design with variable inlet guide vanes tested in 15-lb/sec size hardware was the basis for this prediction. Test results were obtained with progressive inlet throttling so that at design speed the inlet pressure was only about 40 percent of ambient and the Reynolds' number was equivalent to a compressor of only 2.1 lb/sec flow. These results were adjusted for predicted Reynolds' number and hardware scale effects and for anticipated modest efficiency gains expected from the planned development programs. The efficiencies shown are about two points above the then current test results.

Fig. 10 shows the compressor map projected in October 1970 for December 1975 demonstration of a 10:1 P.R. development of the same large com-

[3] Specific speed $N_s = N\sqrt{Q}/H^{0.75}$, where N is rpm, Q is volume flow at inlet total density, cu ft/sec and $H = J\,C_p\Delta T_1$ is isentropic "head rise" in feet, frequently called adiabatic head rise.

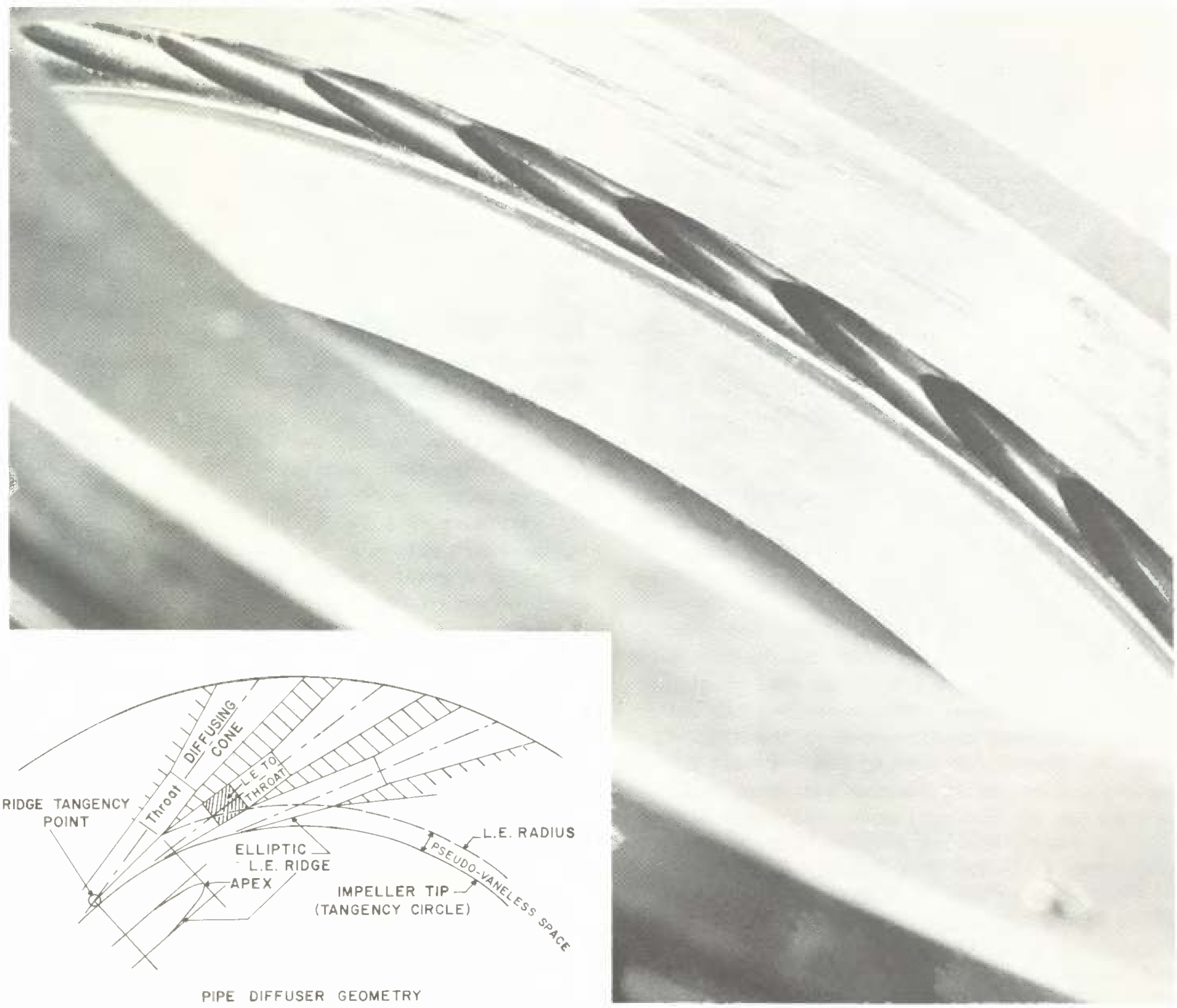

Fig.8 Pipe diffuser leading edges

pressor. This compressor was used in the more conservative SSS 10 engine. Development gains on both large and small rigs from November 1970 to August 1971 were particularly rapid and show that the projected 1975 compressor performance was probably conservative. These results for representative pressure ratios of 4, 7, and 11 are tabulated as follows:

Pressure ratio - - - - - -	4	7	11
Corresponding maximum hp - Standard day, zero prewhirl) - - - - - - - - - -	40	70	170
Projected 1975 (Fig. 9), MAX. η_{T-S} - - - - - - - - -	0.722	0.730	0.793
Large rig test results, Re No corrected, Nov. 1970, MAX η_{T-S} - - - - -	0.750	0.770	0.722
Small rig test results, Mar. 1971, MAX. η_{T-S} - -	0.738	0.757	0.738
Small rig test results, Aug. 1971, MAX. η_{T-S} - -	0.77	0.78	0.75

The results from the small rig are particularly encouraging. They were obtained at referred flows of 2 to 2.5 lb/sec, but with intake throttling to 1.2 to 1.4 lb/sec, so that Reynolds' numbers are below the values for the engines. Since the test hardware is no more than 50 percent bigger than the engine size, scale effects are fairly

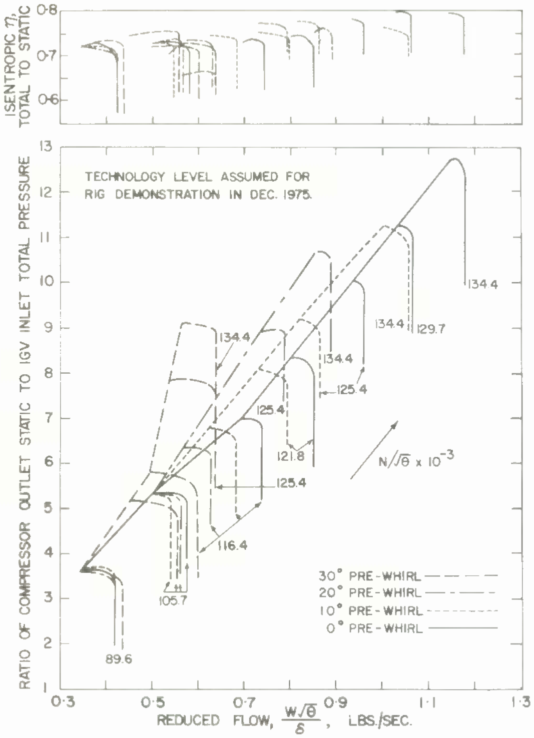

Fig.9 12:1 compressor map, Dec. 1975

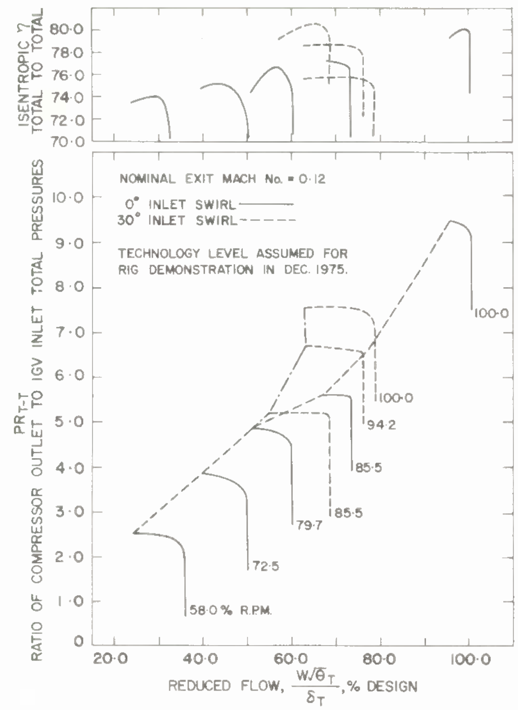

Fig.10 10:1 compressor map, Dec. 1975

closely approximated. The lower efficiency at 11:1 pressure ratio would reduce peak power or else the amount of flat rating in the engine and, hence, reduce the ambient temperature up to which the rated power can be maintained. This effect can be offset by sizing the engine a little larger with consequent weight and cost increments. The gain in efficiency at pressure ratios of 4 and 7 is very encouraging and indicates significant reductions in fuel consumption at the very important low and intermediate power demands, although this would be somewhat offset if the engine was resized for increased peak power.

The compressor rotor has been split into a separate inducer followed by a centrifugal impeller. This allows for better control and easier manufacture of the inducer blades which could be made in high tensile steel for resistance to erosion and foreign object damage. The impeller must have a tip speed approaching 2100 fps, and the resulting centrifugal stresses in a steel part would far exceed safe limits. The much better strength/density ratio of titanium allows the impeller to have satisfactory low-cycle fatigue life, and since it only weighs about 13 oz when finished, the material costs for titanium are quite modest.

Manufacturing this impeller at low cost in large quantity presents a considerable challenge. If feasible, titanium casting could present an attractive solution. At this point in time unfortunately, predicting reliable castings for stress levels, which approach 80,000 psi at 600 F, appears to be forecasting a breakthrough rather than simple development. Titanium castings were, therefore, ruled out.

The impeller vanes are arranged so that each flank can be generated by a single pass of a milling cutter in a special four-axis milling machine. This process has been in use for several years and is now well developed. By assuming the use of a multi-spindle machine, total floor-to-floor times of just under 40 min. with less than 30 min. labor content were estimated.

A further possibility is the GATORIZING$_{TM}$ forging process. This process, which is being developed in Florida by United Aircraft, allows suitably prepared blanks to be pressed into intricate forms with a high level of accuracy. With some redesign, it should be possible to form the impeller vanes to their finished shape in one step, without material wastage. At the moment, this process is some way off for volume production of impellers, but it has excellent potential for low-cost parts' manufacture.

The diffuser passages are conical holes drilled directly in the solid iron casting. To provide an erosion-resistant leading edge at the passage intersections, a stainless-steel ring has been provided, cast in place. This solid heavy

Fig.11 Radial turbine nozzle

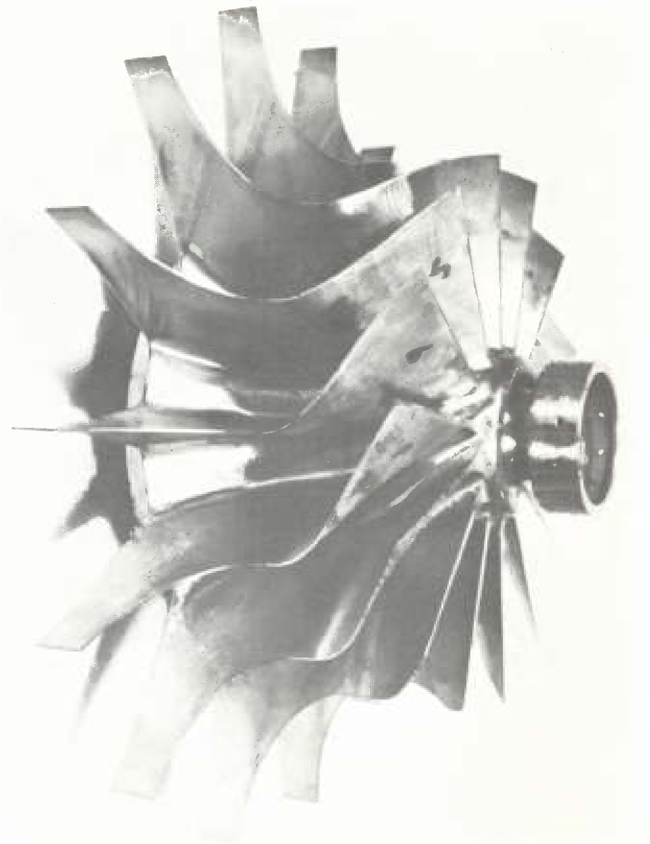

Fig.12 Radial turbine rotor

structure provides naturally for impeller burst containment.

Main Engine Case

The main engine case encloses the flametube and turbine and contains the compressor delivery air. Heat-resisting cast iron will be necessary, both to retain some strength at the higher temperatures and to avoid scaling which could block small air passages or cause erosion damage in the turbine. This same casing would also provide the bulk of the turbine burst containment. This has not been specifically included in the engine design, but local strengthening of the casing should provide containment capability at a small cost with some weight penalty.

Combustor

A single can design was chosen for low potential exhaust emissions, low cost, and inherent reliability due to its axi-symmetry and single-point fuel injection. A very similar combustor is one of three currently being developed for low NO_x emissions for EPA's Office of Air Programs. A conventional value of 98 percent combustion efficiency was assumed for performance, although the actual efficiency should be essentially 100 percent. The difference can be considered a conservative allowance for radiant heat losses up to the turbine exit. The combustor pressure drop is 2 percent static to total (3 percent total to total). This is approximately twice the value on current PT6 engines, but is considered realistic because of the low emission requirements. Construction is fairly conventional superalloy sheet metal because anything else requires an extrapolation of current technology. Again, the small airflow and high pressure results in very small components, so the well-tried approach is not very expensive. Of course, if ceramic components become realistic, then it may well be possible to reduce costs substantially.

Radial Turbine

This component was chosen both for the turbine of the single-shaft engine and for the gas generator turbine of the free power turbine engine. It provides very high work capability, low cost, and relative insensitivity to problems associated with small scale. Its sensitivity to clearance effects on the inlet blades is only about one seventh that of a comparable axial turbine when clearance is expressed as a percentage of inlet flow path width. Peak efficiency near the design point was estimated as 88.5 percent total and 80 percent total to static. These efficiencies are based on test results obtained in 1966 on a joint DRB-UACL research program (Figs.

11 and 12). In this program, the design point of 88.7 percent efficiency at 5.9 pressure ratio was achieved on the first test, and a peak efficiency of 90 percent total to total was obtained at 7:1, the highest pressure ratio tested (Fig. 13). These tests were run on 10.8-in.-dia wheels, but at reduced pressures and temperatures so that the test Reynolds numbers were 1.86 times that for the SSS 12 engine. The design and analysis received further confirmation as design point efficiencies were met or exceeded on a cooled radial turbine program conducted for the U.S. Army (7, 8). Aerodynamic loading on these turbines is kept to modest levels by using a relatively high outlet absolute Mach number, as can be seen by the large difference between total-to-total and total-to-static efficiencies. The unobstructed exhaust on the single-shaft engine allows for efficient diffusion of this high velocity flow.

The turbine nozzle ring is an uncooled investment casting in NX 188 nickel base alloy. This alloy requires coating for oxidation protection, but has relatively good strength at very high temperatures. The maximum inlet temperature of 2000 F is similar to the peak temperatures which have been used in racing car engines (9, 10). In these engines, the uncooled turbine nozzles made of WI 52 cobalt base alloy stand up surprisingly well and give considerable confidence that uncooled hardware is practical at these temperatures in the automotive engine. If actual running experience showed that the nozzle life was poor, it might well be better to lower the peak temperatures a small amount, rather than accept the cost and reliability penalties of cooling these very small nozzles. Again, progress with ceramics technology may permit their use for low-cost nozzles which have good high-temperature capability.

Operating cycles for the engine in 3000 hr of typical automobile use have been analyzed and can be summarized as equivalent to 100 hr at maximum blade metal temperature and 10,000 start-run-stop cycles. These life values are dependent on the type of transmission used and the engine running line, as this affects the variation of turbine temperature with power and the duration of transient temperatures. This type of high-temperature, high-pressure ratio radial turbine requires a combination of properties in the blades and hub which are only obtainable from forged superalloy materials, such as Udimet 700. Current investment cast materials have been investigated and would require a large reduction in either maximum temperature or speed in order to give satisfactory life. An evaluation of the turbine has shown satisfactory blade creep life and burst speed margin at 2000 F peak turbine inlet temperature, using Udimet 700.

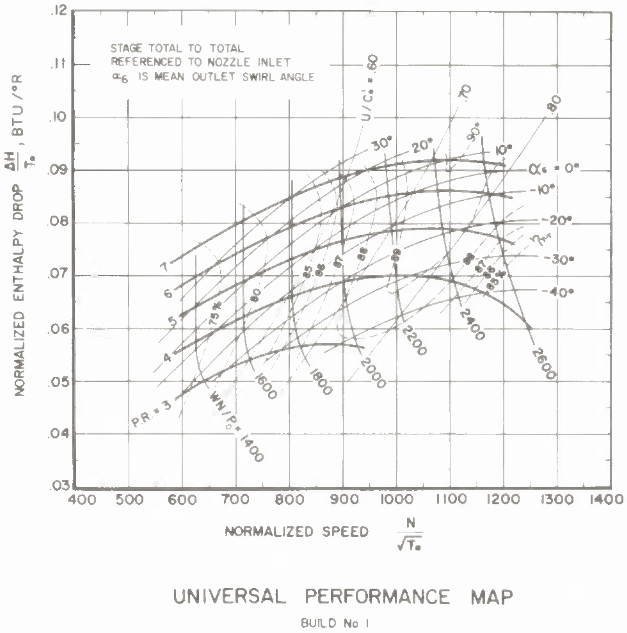

Fig.13 Radial turbine test results

As the turbine cannot be cast, it is necessary to find ways of making this part from a forging. This has been made easier by separating the turbine into two halves, the highly stressed "Star" which has straight radial blades and the exducer which has curved blades and can be made as a casting in coated IN 100. Two methods of producing the star were considered:

Electrochemical Machining. The small size of these turbines allows ECM of the blades from a solid forging in acceptably short cycle times. This manufacturing process requires no development in order to produce acceptable parts.

GATORIZING$_{TM}$. This forging process is applicable to superalloys, as well as titanium. With the possible exception of the thin blade tips, it should be possible to form the turbine star close to its finished shape. This process is coming into use on somewhat less complex parts than radial turbines, but it seems reasonable to expect that it can be developed to produce this part with the necessary material properties. Quantity production costs could also be attractive because of the low material wastage.

The radial turbine shroud is supported entirely by axi-symmetric structure. This frees the turbine/shroud clearance from thermal distortion effects to the maximum extent possible.

The lower pressure ratio across the gas generator turbine on the two-shaft engine results in a higher relative gas temperature and, hence, higher metal temperatures. This means that the peak turbine inlet temperature must be about 100 F lower on the free turbine engine for equal rotor life, or 1900 F. Some further compromise of the aerodynamic design to reduce metal temperatures would allow a higher inlet temperature on this version, if necessary.

Radial turbines are prone to wear due to hard carbon or grit which tends to be bounced between blades and nozzles until pulverized. Care with inlet filtration to prevent sand ingestion and with the combustor to prevent the formation of hard carbon will thus be required. Turbine efficiency is the single most important parameter for engine performance, and additional testing to verify the estimated values at the proper scale and pressure ratio would be very valuable.

Power Turbine (Free Turbine Engine)

The axial power turbine on the high-pressure ratio small engine presents a very difficult design problem. An extensive optimization between disk stress levels, cycle efficiency, and costs is necessary for a complete design. Time and budget limitations did not allow this; consequently, the design reported here must be considered representative only.

The power turbine as shown is fatigue limited at the bore if made from one piece of forged Udimet 700. Blade stresses are satisfactory as the temperature is too low for creep problems. The possibility of casting an integral wheel has been investigated in order to reduce costs. A solid, unbored disk is required, and the fatigue life of the disk would then depend on the hub material properties that could be obtained from a casting and material development program. If the properties of cast material could not be developed to a sufficient standard, then forged material could be used with the blades produced by electrochemical machining, although this would imply some cost penalty.

Bearings

Anti-friction and plain bearings were both considered for this study. Anti-friction bearings have been demonstrated to have low losses at the speeds and D.N. values required for these engines, and life estimates were adequate. It is expected that automotive production quantities would bring bearing costs down to an acceptable level. Plain bearings, while undoubtedly cheaper, were not specified because of the higher losses anticipated. If they can be shown to have satisfactory losses, then they could easily be incorporated.

Shaft Dynamics

A critical speed analysis was made of the rotor and bearing arrangement of the single-shaft engine. This showed that the design is satisfactory, although further optimization of the exact bearing positions would be desirable to improve the margin between the operating speed range and certain critical speeds.

Reduction Gearbox

The reduction gear, which has been drawn and costed for the single-shaft engine, consists of a single-stage spur of 10:1 reduction. This was drawn before the transmission requirements were understood, and it is more likely that a two-stage spur gear set would be needed with an output speed of 4000 to 6000 rpm. The pitch line velocity for the first-stage pinion would be unchanged at 30,000 fpm in either case. This is a high value, but is within modern practice.

The free turbine engine requires an accessory box on the gas generator of similar complexity and gear tooth quality to the single-shaft reduction box. In addition, it has a power-reduction gearbox turning at somewhat lower input shaft speeds.

INSTALLATION PROBLEMS

Compared to the low-pressure ratio regenerated engines, which have been demonstrated up to now in automobiles, the simple-cycle engine has lower airflow, which should ease intake and exhaust ducting problems. These ducting problems are much more severe for any gas turbine than for a reciprocating engine with, at most, one quarter the flow.

As nearly all the waste heat is carried away in the gas turbine exhaust flow, this can present a problem, particularly on the simple-cycle engine at idle where a somewhat greater quantity of heat is carried away by a smaller airflow than on the regenerated engine. Some means of dispersing this heat over a large area would seem to be necessary.

The noise from gas turbine installations is usually favorably compared to reciprocating engine installations because of its smoother and less pulsating character. The necessary intake filtration for this engine will provide a high degree of compressor noise attenuation, and careful exhaust diffusion and ducting to minimize exhaust losses will tend to minimize exhaust noise. However, careful design and development of the intake and exhaust will certainly be needed to meet the anticipated standards for noise.

Although the basic engine size is quite small, careful attention would have to be paid to accessory size and location, to the intake air system and the exhaust system to obtain an optimized engine installation. The current conventional automobile layout has evolved over at least a 70-year period, and it appears highly unlikely that the same vehicular layout will prove optimum for any radically different engine. It may well be very desirable to fit these engines into nearly conventional chassis, however, for initial development and evaluation and perhaps also for initial production.

POTENTIAL FOR LOW EMISSIONS

The relatively steady burning process, availability of excess air, and absence of cold walls makes it much easier to design gas turbines for low exhaust emission than reciprocating engines. Unburned hydrocarbons (UHC) and carbon monoxide (CO) are typically very low at high power settings in any gas turbine. Aircraft engines, however, are frequently operated fuel rich in the primary zone for good altitude relight and general stability characteristics. The smaller of these engines will tend to use simple spray nozzles which give somewhat coarse atomization, hence slow burning, at low fuel flows. Under idle and low power conditions, the CO and UHC may then be higher than desired for ground vehicles, but this is not a fundamental problem; rather it requires additional development and particularly better low-flow fuel injection, perhaps using dual-orifice spray nozzles, airblast, prevaporization, or a pre-mixed system. Emissions of the oxides of nitrogen, NO and NO_2, called NO_x, are the most difficult exhaust pollutants to control and reduce to desired levels for gas turbines in ground vehicles (1 and its references). Many theoretical studies (11-13) have shown that flame temperature is, by far, the most significant factor affecting NO_x formation. Unpublished kinetic reaction calculations by Pratt & Whitney Aircraft indicate that primary zone residence time is perhaps one twentieth as significant a factor as flame temperature and pressure level is only about one thirty-third as powerful a factor (assumed combustor inlet of 800 F and 15 atm, 0.8 equivalence ratio, 0.002 sec after completion of combustion). Uniformity of primary zone, amount of recirculation, and primary zone fuel-air ratio will all influence the mean effective values of flame temperature and residence time. Peak flame temperature will vary almost linearly with combustor inlet temperature which, therefore, can be presumed a very important index of NO_x formation. This has been repeatedly documented in recent papers (13-17). Here the simple-cycle engine has a very major potential advantage over regenerated or recuperated gas turbines, particularly at low power where the bulk of the fuel is consumed in automobile applications. The regenerated or recuperated engine, using power transfer and/or variable power turbine nozzles set for low fuel consumption, will typically by operating at about 1300 F combustor inlet temperature at low power (13, 14) compared to about 475 or 500 F for the advanced simple-cycle engine. Tests on one regenerated gas turbine engine [Fig. 11 in reference (13) and Fig. 14 in reference (14)] showed that an 800 F reduction in flame temperature yielded a sixfold reduction in NO_x. Data on certain non-regenerated engines and experimental combustors show substantially larger reductions [Fig. 10 in reference (15) and Fig. 12 in reference (16)]. The data from these reports has been nondimensionalized and plotted in Fig. 14. The lower slope of the regenerated combustors presumably is due to their leaner air-fuel ratio. Even if one-third higher fuel flow is assumed for the simple engine at low power, an estimate of five times reduction in tendency to NO_x production appears conservative.

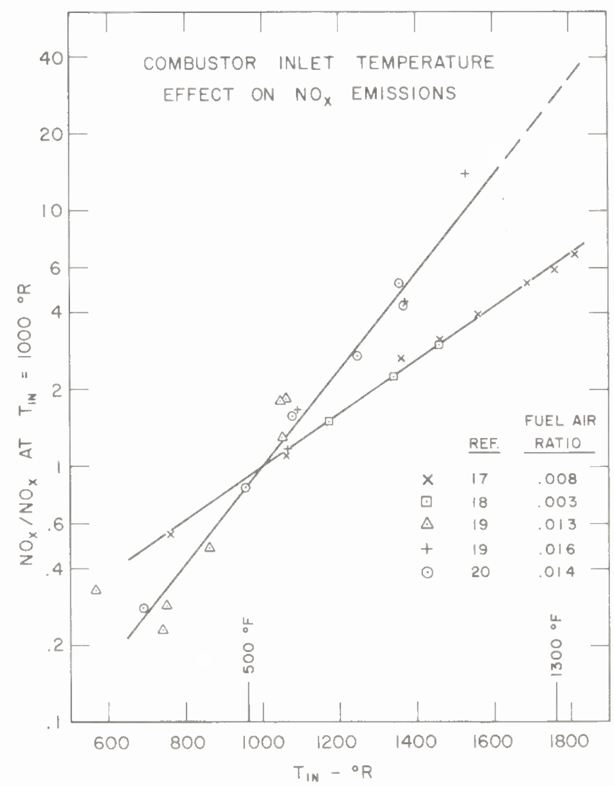

Fig.14 Effect of combustor inlet temperature on NO_x emissions

Very recently, this comparison has been studied in some detail (18). Fuel consumption for the simple-cycle engine modeled was twice that for the regenerative engine. This regenerated engine was also assumed to have a low-power combustor inlet temperature of only 1000 F. These assumptions very significantly worsen the simple-cycle engines comparative NO_x emissions which were still computed to be only one-third or less or a grams/mile basis.

TRANSMISSION

An attempt has been made in this paper to show the advantages of the single-shaft engine, but these are meaningless without a suitable transmission. The study considered an eight-speed mechanical transmission with a controlled slipping clutch as a potential single-shaft engine transmission using essentially available components which give good efficiency. Subsequent studies have indicated than an infinitely variable transmission, even with lower efficiency, can be attractive because it allows the engine to operate along an optimum running line. An infinitely variable transmission would also be easier to integrate with the engine to give acceptable handling (19). The final choice between single-shaft and free turbine engines must be made considering the overall effectiveness of the installed engine.

DEVELOPMENT PROGRAM

A considerable amount of development would be necessary in both design and manufacturing techniques in order to have the other engine components available to complement the advanced compressor. In addition to this, any completely new power unit for production automobiles must be developed as a system. Although beyond the scope of this paper, such a program must include development and integration of the engine, transmission, drive train, and accessories. Packaging for installation in a suitable production vehicle would form another major program.

CONCLUSIONS

1 An advanced simple-cycle engine was identified as a prime candidate for low-emission, low-cost gas turbine automobiles (2, 3).

2 The key, and most advanced, component of that engine is an ultra-small, high-speed centrifugal compressor. While the performance projected for that compressor may appear unbelievably good, it is shown to be, in fact, consistent with, and quite close to, current UACL levels of technology.

3 Other components and problems requiring solution include: (a) transmission, (b) exhaust systems, and (c) power turbine (of the free turbine version of the engine).

4 Low combustor inlet temperature promises very large reductions in the basic tendency to produce NO_x which is the most difficult gas turbine exhaust pollutant to control.

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