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THE DIFFERENTIAL GAS TURBINE USING ELECTRIC TRANSMISSION

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ABSTRACT

This paper describes studies of simple gas turbine engines integrated with electrical transmission components. Recent developments in high-speed lightweight electrical machines and compact power electronics have enabled alternators and motors to be produced which can be coupled directly to the shaft of a gas turbine without an intermediate gearbox. For applications which require a wide range of power outputs, a single-shaft gas turbine with a high speed alternator can be run at constant speed while varying the current drawn from the alternator. This combines the flexibility of operation of a separate power turbine with the simplicity of a single-shaft engine. With this arrangement, in traction use high torques are obtained at low speed, while near-constant engine efficiency is sustained to about 50% of the design power. In the differential engine, the mechanical linkage between the compressor and the turbine is replaced with an electrical linkage. The turbine drives an alternator, and part of the alternator power is taken by a high-speed motor to drive the compressor. The excess alternator power forms the output of the engine. The compressor and turbine are now able to run at different speeds, and their operating points can be separately optimised at different engine conditions. For such an engine, studies show that high efficiency can be maintained to low power levels.

INTRODUCTION

Small gas turbine engines are widely used for power generation in cogeneration plant, total energy modules, standby generators and auxiliary power units. They are rather less frequently used in traction applications, but there remains a growing interest in this field. Despite a substantial body of design and development work, the gas turbine engine has never been considered as a serious contender to the internal combustion engine, or progressed into mass production (O'Brien 1980, Harmon 1982). The reasons for this involve a combination of factors, including a fuel economy which remains uncompetitive, a poor transient response, and high development costs to compete against an already mature product. However, the gas turbine engine has some important advantages which apply to both the power generation and traction applications. It is a compact, lightweight engine, very simple in design, with no major oscillatory masses. It has a multi-fuel capability, and increasingly important today, it is potentially a very low emission engine, because of its lean burn and continuous combustion characteristics.

In its simplest form, a gas turbine comprises a compressor, combustion chamber and turbine driving the compressor and providing output power (Fig. 1*a*). Usually a step-down gearbox is required because the speed of rotation of the engine shaft is much higher than that required for conventional electrical generators or for traction. Such an engine performs satisfactorily in applications that require constant speed running, but when a wide speed range is required, the power and the efficiency become very low at low speeds. This is because the compressor is constrained to run at a linear function of the output shaft speed, and at low speeds the pressure ratio developed is small. A multi-ratio gearbox may be used to alleviate this problem, but does not totally eliminate it, and does create additional complexity. For this reason, where variable speed operation is required, it is conventional to use a separate power turbine with a second shaft to provide the output power (Fig. 1*b*). The compressor and output shaft speeds are now decoupled, and the compressor speed and pressure ratio can be maintained at high levels even when the power turbine is stalled.

Recent developments in power transfer technology have now made it possible to combine the simplicity of the single-shaft engine with the operational flexibility of the two-shaft engine,

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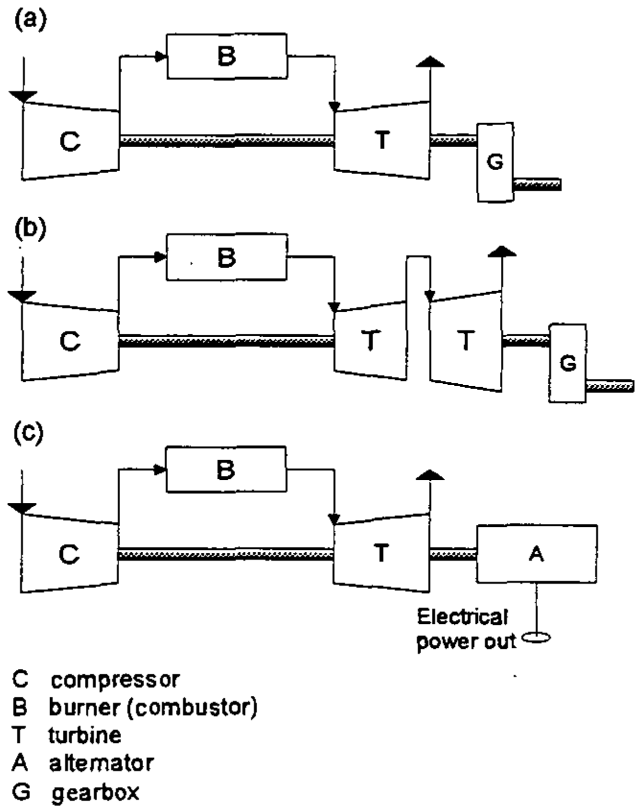

Fig. 1 Schematic diagrams of (*a*) single-shaft engine with mechanical transmission (*b*) two-shaft engine with mechanical transmission (*c*) single-shaft engine with electrical transmission

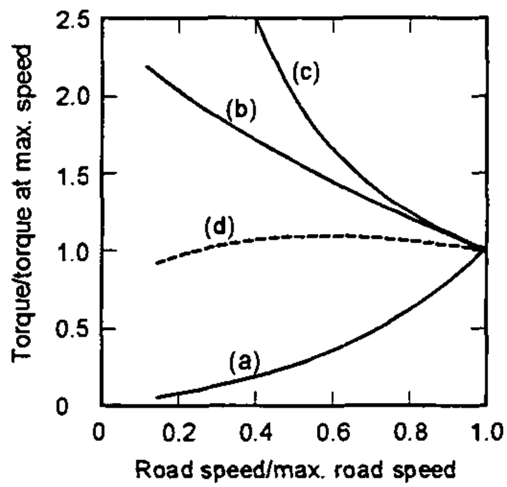

Fig. 2 Engine torque characteristics (adapted from Cohen, Rogers and Saravanamuttoo 1987). (a) single-shaft engine, mechanical drive (b) twin-shaft engine, mechanical drive (c) single-shaft engine, electrical drive (d) internal combustion engine

and also to eliminate the step-down gearbox, which is usually a heavy and expensive component. The key to this has been the development of alternators which can run at the same speed as the engine shaft, and can thus be directly coupled to it (Fig. 1*c*). Such an alternator can be driven at constant speed, and provides a constant voltage output. The power output is then determined simply by the current drawn. Clearly, the fuelling rate of the engine must be controlled precisely in response to the power demand, and for this an electronic engine management system is required.

The power available from an engine with a direct-coupled, electrical transmission is theoretically constant under all conditions, and in practice is limited only by the heat dissipation in the alternator and other transmission components. For traction applications, this gives a very favourable torque characteristic, as shown in Fig. 2. The internal combustion engine today has a roughly constant torque available over a wide speed range. The single shaft engine with mechanical transmission has a torque falling to zero with speed (and thus has no self-starting capability). By contrast, the two-shaft engine with mechanical transmission has a good torque back-up characteristic, but still below the constant power characteristic of the single shaft engine with electric transmission. This last provides an excellent potential for engine acceleration, and since the engine is running at constant speed, there is no delay introduced by the rotating inertia of the engine itself. Other advantages of constant speed running include a significant reduction in low-cycle fatigue of the turbine rotor, and better matching of rotating and stationary components, with better control over tip rotor clearances and other leakage paths.

THE SINGLE-SHAFT ENGINE WITH ELECTRIC TRANSMISSION

A preliminary design study was undertaken of a small, single-shaft engine with integrated high-speed alternator for portable power generation or traction applications. The basis of this is a simple cycle model for predicting design and off-design performance, such as is described, for example, by Cohen, Rogers and Saravanamuttoo (1987). Compressor and turbine stage characteristics are based on the authors' own rig test experience, the alternator characteristics are similarly based, and the combustion process is modeled as a simple heat addition with combustion efficiency and pressure loss. The fuel properties are those of motor gasoline. Similar assumptions were used in all of the various engines simulated, to ensure realistic comparisons.

Following an initial cycle optimisation study, the cycle pressure ratio was set at 5, which can be achieved with an aerodynamically efficient single stage centrifugal compressor, and is appropriate to the inclusion of regenerative or recuperative heat exchange. The turbine entry temperature was chosen to be 1200 K, which gives a long turbine life using

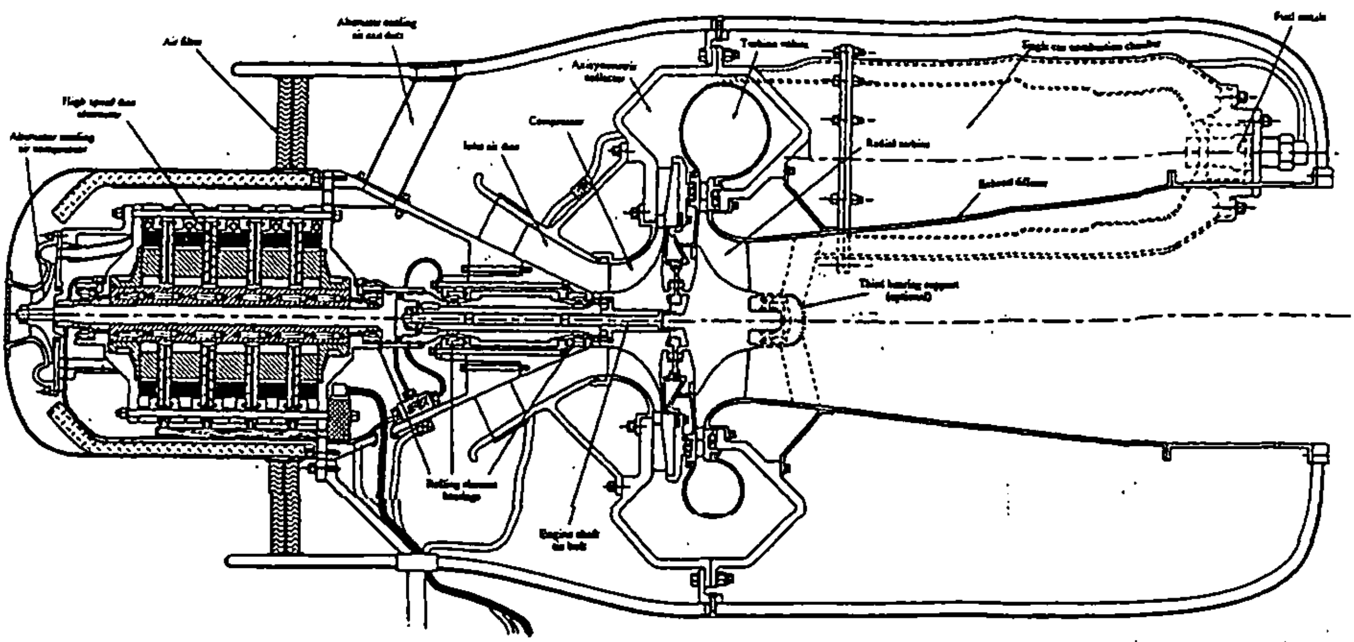

Fig. 3 The single-shaft engine with high-speed alternator

conventional metal alloys without requiring internal cooling. In fact, for these applications, the life requirements are such that it is possible to accept higher turbine entry temperatures, up to 1300 K, while developments in ceramic materials (Boyd and Kreiner 1988, Helms 1988) now hold out the strong possibility of increases to 1600 K.

A drawing of the engine-alternator assembly is shown in Fig. 3. This was originally designed for an output power of 50 kW, but in practice it can be scaled without difficulty to suit power requirements in the 20–100 kW range. Even higher and lower power demands than this can be accommodated without major changes to the principal design features. At the nominal output, the shaft speed is 100 000 rev/min. The design shows a simple construction. The compressor is a single stage centrifugal compressor with channel diffuser, and a prototype unit has demonstrated 79.5% total to static efficiency and a wide flow range on test. The combustion chamber shown is a single, cylindrical can type, but an annular combustor may be adopted for a prototype engine. The turbine is a single-stage, radial inflow unit. This was preferred to the more conventional (for gas turbines) axial stages because at least two stages would be required at this pressure ratio. On test, a prototype radial turbine achieved 83% total to static efficiency with non-optimised nozzle vanes (Pullen et al 1992), and with a vane redesign, another 3% improvement is predicted.

The alternator is a multi-stage disk design described by Pullen (1991) and Fenocchi (1991). The rotors each consist of an aluminium frame containing six, high strength, rare-earth magnets. These are retained by an outer ring of wound carbon fibre, which provides the strength necessary to withstand the centrifugal load in operation. The stators are designed to minimise eddy current losses, and the final alternator will be partially evacuated to reduce windage loss. The bare engine and alternator assembly, stripped of accessories, weighs 20 kg for 50 kW power output.

The concept engine also incorporates a heat exchanger, although this is not shown in Fig. 3. Several heat exchanger designs have been studied. At one end of the scale a rotary regenerative heat exchanger is predicted to achieve an effectiveness of 0.95, but there are concerns here for the effect of carry-over of hot exhaust gases into the compressor delivery air, and of the wear and life of seals on the regenerator disk. There are also serious weight implications, since this heat exchanger would add 30 kg to a 50 kW engine. Various stationary recuperator designs have also been studied, and it was found that an effectiveness of 0.5 was achieved with double pass designs using plain or louvred fins, which can be fitted inside the engine envelope of Fig. 3 around the turbine diffuser, and weigh only 5 kg.

The use of a heat exchanger brings considerable benefits to the gas turbine performance, as the thermal efficiencies in Fig. 4 show. These are plotted for the design pressure ratio and the measured component efficiencies, and show the improvements of both higher turbine entry temperature and heat exchanger effectiveness.

The specific fuel consumption of the engine with a heat exchanger effectiveness of 0.5 is plotted in Fig. 5. This graph shows that at 100% speed and power levels above 50% of design, the fuel consumption is very nearly constant. At low power it rises in spite of the benefits conferred by the electric transmission. This is because the mass flow rate decreases and the compressor and turbine, despite constant speed running,

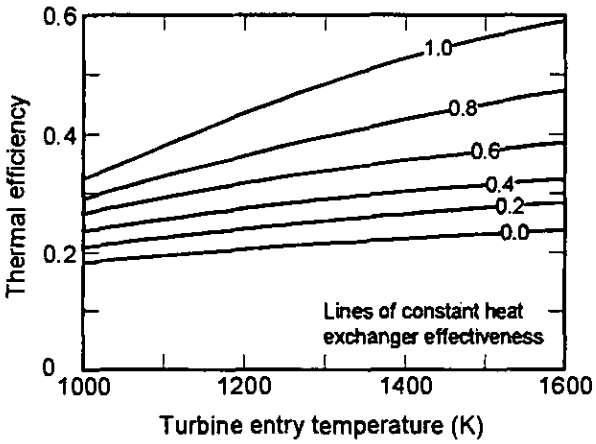

Fig. 4 Thermal efficiency of single-shaft engine with high-speed alternator

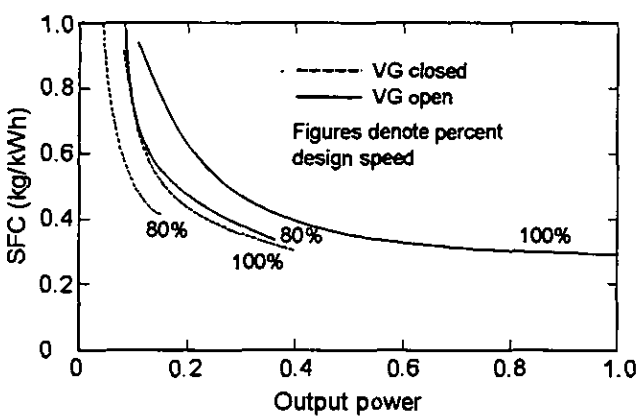

Fig. 5 Specific fuel consumption of single-shaft engine with high-speed alternator

gradually move away from their points of maximum efficiency, and because the parasitic pressure losses in the combustion chamber and heat exchanger assume a greater importance. The problems of compressor and turbine operating points have been tackled in two ways (see Baines et al 1992). The first of these is variable geometry (VG) devices, which may be incorporated effectively to re-match the compressor and turbine at low power conditions. Such devices include compressor preswirl vanes, variable diffuser vanes, and variable turbine nozzles. Studies showed that for maximum effect variable geometry should be applied to both the turbine and the compressor, and that these devices should be activated together. The benefits of this are shown in Fig. 5, where it is assumed that with the VG devices in the fully closed position, the effective flow areas are reduced by 50% for both the compressor and the turbine.

However, variable geometry devices add considerable complexity to the engine design and operation, and to the cost of

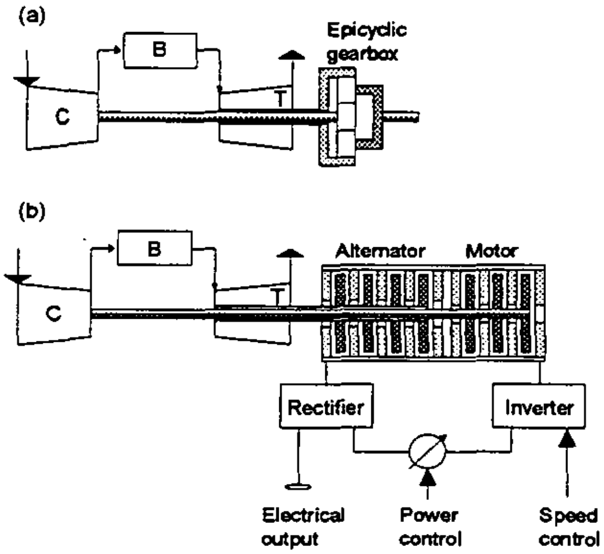

Fig. 6 Differential gas turbine engines (*a*) mechanical differential gear (*b*) electric differential

the engine. Another possibility is to allow the engine to switch to a lower speed at low power levels, and equivalent 80% speed curves are also shown in Fig. 5. The benefit of low-speed running is a saving in fuel consumption of the order of 25% at low power, but this also runs counter to the constant speed philosophy here proposed, and the benefits in low power fuel economy must be set against the drawback of reduced vehicle acceleration at the points where the engine speed changes.

THE DIFFERENTIAL ENGINE

The use of a high-speed electric transmission also opens up another possibility, which is to replace not only the *external* mechanical transmission from the engine with electrical components, but also the *internal* transmission of power within the engine itself. In its simplest form, the shaft from the turbine driving the compressor is replaced by an electric motor which is driven from the alternator output. This then allows the speeds of the turbine and compressor to be decoupled, and introduces an additional degree of freedom in the matching of them. This is the differential engine concept.

The differential engine is not a completely new idea. Various proposals have been made in the past for engines with a mechanical linkage, using an epicyclic gearbox (Hutchinson 1956, Anon 1958). Such an arrangement is shown in Fig. 6*a*, and can provide large output torque at low or zero output shaft speed while maintaining high turbine efficiency. At zero shaft speed the planet gears act as idlers, and the turbine drives the compressor at its maximum speed. As the turbine speed is increased, the planetary centres and the output shaft rotate, but

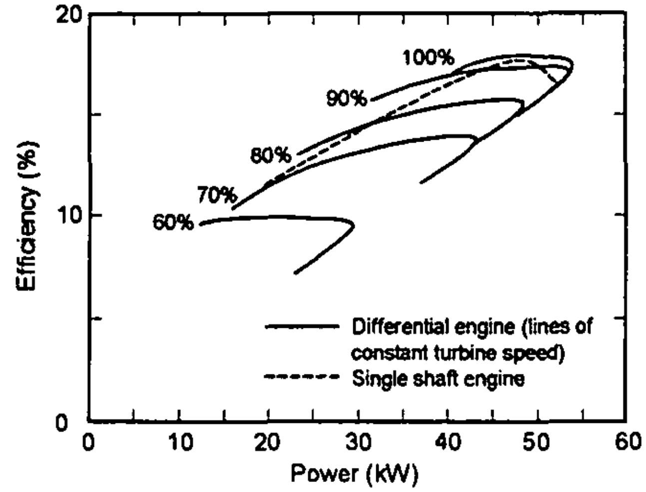

Fig. 7 Thermal efficiency of the differential gas turbine

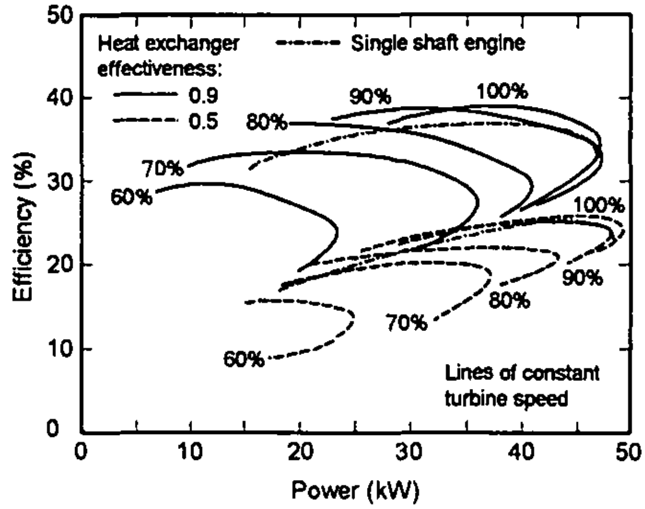

Fig. 8 Thermal efficiency of the differential gas turbine engine with heat exchange

at a lower speed than the turbine. Thus, the compressor can be maintained at constant speed for all output shaft speeds.

The electric differential engine provides a similar result, but is able to dispense with the epicyclic gear and provides an electrical output directly. It is shown in schematic form in Fig. 6*b*. The output of the alternator is a.c., at a frequency which depends on the speed of the turbine. The current is then rectified and a controlled current is drawn from it to drive the compressor. In order to do so, the d.c. is inverted to a frequency whose value will determine the speed of the motor which drives the compressor. Thus, the compressor speed and power input can be independently controlled by suitable electronic engine management, and the turbine speed and output power are determined by the requirements of matching the compressor air flow rate and pressure ratio, and by the engine fuelling rate. The alternator and motor are shown in this figure as an integrated unit. In practice it is possible to separate them, have the motor at the 'front' of the engine and avoid the complication of coaxial shafts

The performance of the differential engine is plotted as thermal efficiency as a function of power output for lines of constant turbine speed in Fig. 7. The operating line of the single-shaft engine is also displayed. For the purposes of this comparison, neither engine incorporated a heat exchanger, and so the absolute levels of efficiency are low. It can be seen that for each turbine speed, there is a clearly defined maximum power output, and at this point the engine is also operating at or very near its maximum efficiency. The high efficiency is then held to power levels much lower than the maximum. By comparison with the single-shaft engine, the maximum efficiency is very similar, because the same component characteristics are used in calculating the performance of both engines, but the curves for the differential engine are generally flatter and give better efficiencies at low powers.

Along each constant turbine speed line shown in Fig. 7, the compressor speed varies, so that the compressor operating line runs roughly parallel to the surge line and follows the contours of peak compressor efficiency. This is reflected in the high efficiencies of the engine cycle. The alternative strategy of running the engine at constant compressor speed will cause the compressor and engine efficiencies to fall more quickly at low powers, and the high power point will ultimately be limited by compressor surge.

This comparison is repeated for engines with heat exchangers in Fig. 8. Curves are shown for two values of heat exchanger effectiveness, 0.5 and 0.9, corresponding roughly to a simple, compact recuperator and a rotary regenerator respectively. The latter in particular shows that very competitive levels of thermal efficiency can be achieved, and again the trend is for the differential engine to have quite flat efficiency characteristics over a wide power band.

Operation at constant turbine speed thus has considerable benefits in performance, is unlikely to incur problems of compressor stability, and also gives a constant alternator output voltage which may be advantageous in some applications. However, other operating strategies are possible, and were investigated. The differential engine characteristics at constant turbine entry temperature are shown in Fig. 9, and these have a similar form to the constant turbine speed curves. The effect of increasing the turbine entry temperature is to increase the power output quite markedly, but its effect on thermal efficiency is much more limited. An increase in temperature from 1200 K to 1600 K, for example, increases the power output by 65%, but the efficiency by only 3–4 percentage points. This is a consequence of the fact that the pressure ratio is held constant. In fact, a study of the single-shaft engine-alternator showed that significant gains in efficiency could be achieved by improvements in heat exchanger effectiveness and turbine and

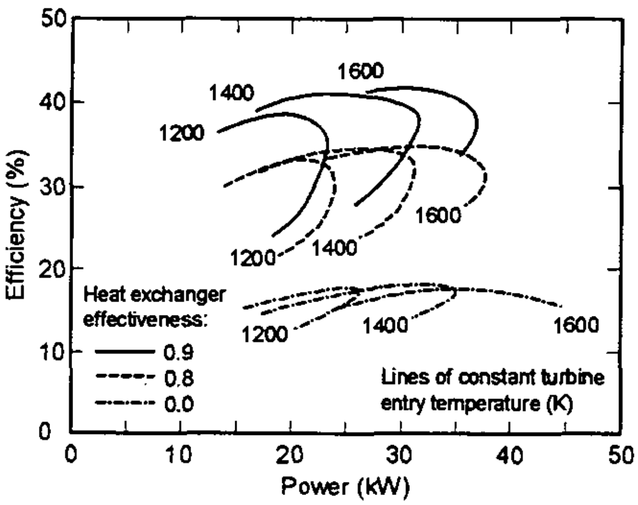

Fig. 9 Differential gas turbine operation at constant turbine entry temperature

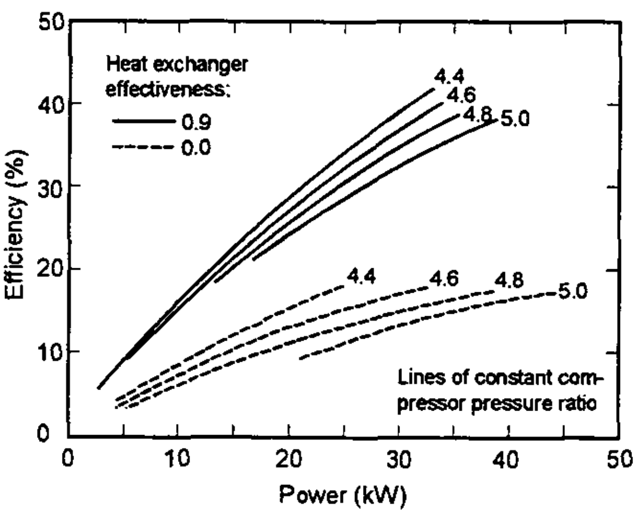

Fig. 10 Differential gas turbine operation at constant pressure ratio

compressor efficiency, rather than by increases in turbine entry temperature (Baines et al 1992), and these results apply equally to the differential engine.

Another operating strategy which was investigated was to run the engine at constant compressor pressure ratio. Here it was found that a wide range of turbine entry temperature was required to cover the complete power band at constant pressure ratio. It can be seen (Fig. 10) that the efficiency is not maintained at a high level for each pressure ratio, and at low power, the thermal efficiency is very low, even with heat

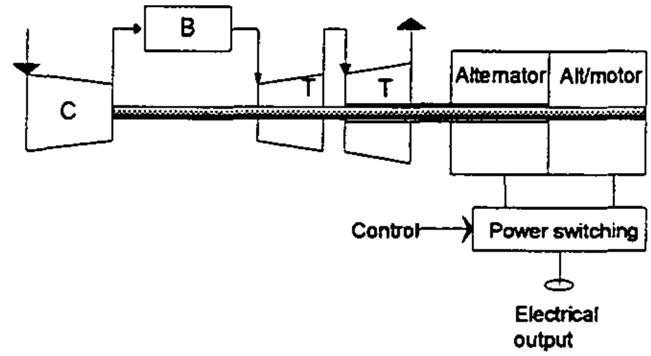

Fig. 11 Schematic diagram of differential gas turbine with separate power turbine

exchange. At these conditions the majority of the turbine power output is being expended in sustaining a high pressure ratio across the compressor, and consequently the fuel demand is high. Clearly constant turbine entry temperature or constant speed operation is to be preferred, for under these conditions the compressor pressure ratio is not constrained to be unnecessarily high at low power levels.

MORE COMPLEX ENGINES

More complex engine cycles using the differential concept are possible. Hutchinson (1956) describes a two-shaft engine, in which the power turbine drives the ring of an epicyclic gear, the compressor-turbine shaft is connected to the sun gear, and the power output is taken from the planet gear. This engine was shown to operate over a wider output shaft speed range along a typical load line than the conventional two-shaft engine, although the low-speed torque is somewhat lower. Translating this arrangement into electrical components gives rise to a practical difficulty, because at different engine conditions, there may be power transfer to or from the compressor-turbine shaft. Any electrical machine connected to this shaft will, therefore, be required to act both as a generator and as a motor (Fig. 11). While there is no reason in principle why this should not be done, it has not yet been examined in detail or demonstrated, using the alternator technology described in this paper. There are potential benefits from this scheme, and it will be further investigated in the future.

It is also possible to combine gas turbine and internal combustion engines to form compound engine schemes, also using high-speed alternators and motors for power transfer in place of mechanical gearing. The preliminary results of some investigations in this direction suggest that very high ratings are possible, and this project is continuing (Panting 1993).

CONCLUSIONS

The development of high-speed electric alternators and associated low-cost power electronic devices has increased the scope for simple gas turbine engines, particularly in applications where variable power is a requirement. The conventional single-shaft engine is at a considerable disadvantage in this respect, with a very poor torque curve and a very low thermal efficiency at low power.

A high-speed alternator direct coupled to a single-shaft engine allows the engine to be run at constant speed, while the power is varied by means of the current drawn from the alternator. In traction applications the resulting power is constant for all driving speeds, resulting in very high torque. The thermal efficiency of such an engine shows a considerable improvement at low power conditions. Ultimately, however, the engine efficiency is brought down as the compressor pressure ratio and the turbine entry temperature fall. Some recovery of performance is possible using variable geometry devices or allowing the engine to switch to a lower speed, but these detract from the simplicity of the engine and the operational benefits obtained from constant-speed running.

The differential engine is a further development of the concept of integrating gas turbine and high-speed electrical power converters. By eliminating the mechanical coupling between the compressor and turbine, additional flexibility of operation is obtained, and in particular, the operating point of each component can be separately optimised. As a result, very flat efficiency characteristics are obtained, and the efficiency remains within 2 percentage points of the maximum to below 50% power. Various engine operating strategies were explored, and it was found that operation at constant turbine speed or constant turbine entry temperature gave the best results. Other modes of operation were limited by a falling compressor efficiency, and problems of compressor stability.

More complex differential gas turbine engines, using two shafts and a separate power turbine, can be conceived, together with compound gas turbine/internal combustion engine systems, all based on the new high-speed electrical machines. In some cases, however, these rely on as yet untried features of such machines. These will be the subject of future investigations.

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