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LOW-PRESSURE-RATIO, REGENERATIVE, BRAYTON-CYCLE ENGINES: THE NEXT GENERATION OF MARINE PRIME MOVERS?

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ISSN 0309-3948
Trans I Mar E (TM)
Vol. 99, Paper 1 (1986)

Low-pressure-ratio, Regenerative, Brayton-cycle Engines: The Next Generation of Marine Prime Movers?

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SYNOPSIS

Current gas-turbine engines for marine propulsion are aircraft-engine derivatives and operate on simple (CBE) cycles with comparatively low thermal efficiencies. In this study three other gas-turbine cycles were examined for their potential to provide greatly improved marine propulsion: the regenerated (CBEX) cycle, the intercooled regenerated (CICBEX) cycle and the direct-plus-inverted (DIC) cycle. From performance plots it was concluded that the CBEX cycle designed for its optimum pressure ratio (about 3:1) and with a high-effectiveness regenerator results in predicted thermal efficiencies in excess of 50%. Further performance gains can be achieved with the incorporation of improved ceramics, but the performance shown in this study could be produced with conventional technology and materials. The design-point and off-design-point performance of an engine operating on the CBEX cycle was calculated and compared with the performance of diesel engines over the complete power range, and the gas-turbine engine was found to be more efficient. Additional advantages and other aspects of the design are also discussed.

INTRODUCTION

Research and development of gas-turbine engines are dominated by the sophisticated requirements of military and commercial aircraft engines. The thermodynamics of the cycle require high temperature of energy addition (as do all heat-engine cycles) and therefore high turbine-inlet temperatures. This has resulted in substantial increases in turbine-inlet temperatures over the last few years (Fig. 1). Additionally for light engines (suitable for aircraft) the simple cycle (compressor plus burner plus expander, hence the designation CBE) must be used. For the full advantages of the higher temperatures to be realized with the cycle, the compressor pressure ratio must also be substantially increased. Modern jet engines have compressor pressure ratios between 20:1 and 40:1. Reasonably efficient compressors with a high pressure ratio require complex geometric arrangements.[1,2] A very large proportion of the huge expense necessary to develop new aircraft engines is attributed to the cost of producing acceptable high-pressure-ratio compressors.

The diesel engine is at present the most efficient prime mover. The suggestion that a better engine could be produced for such specialized duties as those demanded by marine propulsion requires a thorough and convincing explanation. A detailed justification for the approach is given in Refs 1 and 2, but a short summary is included here.

To a large extent, aircraft-engine developments have dominated much of the commercial gas-turbine field. Many industrial gas-turbine engines are simply jet engines in which the exhausts pass through large shaft-power turbines in place of the normal propelling nozzles.[3] The US Navy's principal gas-turbine propulsion engine, the LM 2500, is derived from the GE CF6 jet engine in this manner.

The maximum possible thermal efficiency of a heat engine is the thermodynamic Carnot limit (the Carnot coefficient), which is equal to $1 - 1/T'$, where T' is the ratio of the maximum to minimum cycle temperature. In gas-turbine engines T' is the ratio of the (absolute) turbine-inlet temperature to the (absolute) compressor-inlet temperature. For marine engines in the late 1980s, T' will be between its present value of about 5 and a future value of about 6, attainable through conventional blade-cooling techniques or through the use of ceramic turbine blades. The Carnot coefficient is therefore between 0.80 and 0.83.

There are two ways in which a Brayton cycle can approach this limit. First, by the incorporation of a heat exchanger the average temperature of energy (heat) addition is increased and the average temperature of energy (heat) rejection is decreased.[4] Secondly, by using a low-pressure cycle these effects are substantially enhanced. This approach increases the size and the weight of the turbomachinery, making it unsuitable for high-speed aircraft. For marine and other uses, the turbomachinery is still small in comparison with the size of

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David Gordon Wilson was born and educated in Warwickshire, served briefly in the merchant marine, and was awarded fellowships for graduate research at the University of Nottingham and for post-doctoral study at Harvard University and the Massachusetts Institute of Technology. He worked as a post-graduate apprentice, and later as a gas-turbine engineer, at Hawker-Siddeley Brush and on gas-turbine design at Boeing Airplane and Ruston & Hornsby, and he has taught mechanical engineering in Nigeria. For six years before joining the MIT faculty in 1966 he was technical director of Northern Research & Engineering Corp. in London and later in Cambridge, MA.

other engines, as will be seen below. The shaft speed is considerably reduced compared with high-pressure-ratio gas turbines, which is an advantage, and the thermal efficiency is considerably improved.

Reasonable component efficiencies were specified for the CBE, CBEX, CICBEX and DIC cycles and their potential to provide improved marine propulsion was compared. The low-pressure-ratio, highly regenerative CBEX cycle can attain thermal efficiencies in excess of 50%. The preliminary design of an engine operating on this cycle was carried out and the performance map of the engine over the complete speed–power range was calculated. The calculations show that this engine maintains a thermal-efficiency advantage in design-point and off-design-point performance compared with other prime movers over a considerable power range and rating. Other advantages of this engine include reduced weight and volume (increased payload), potentially lower maintenance costs, lower noise, cooler exhaust and less noxious emissions. Other aspects discussed include the choice of regenerator, the performance of compressors of different reaction, the effects of the pressure ratio on engine performance and the expected gains from future advances in ceramics (such as further improvements in thermal efficiency and reduced use of strategic materials).

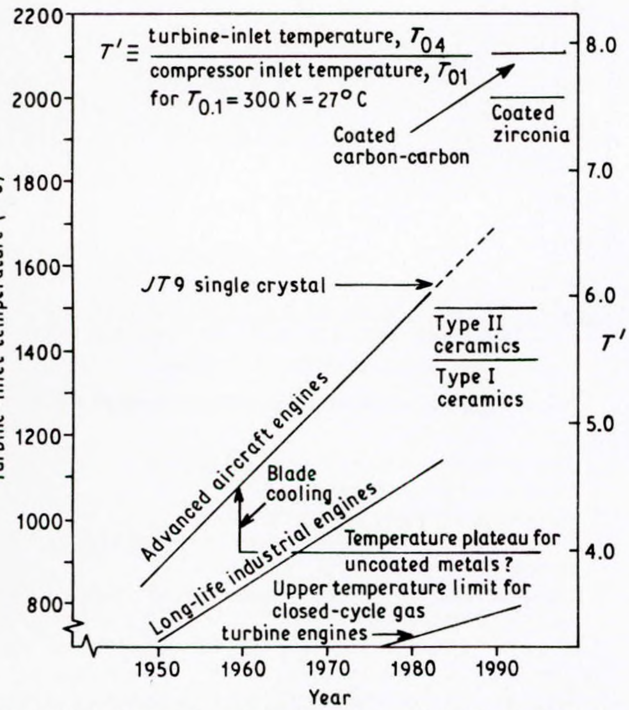

FIG. 1: Historical and projected increase of turbine-inlet temperature (from Ref. 1)

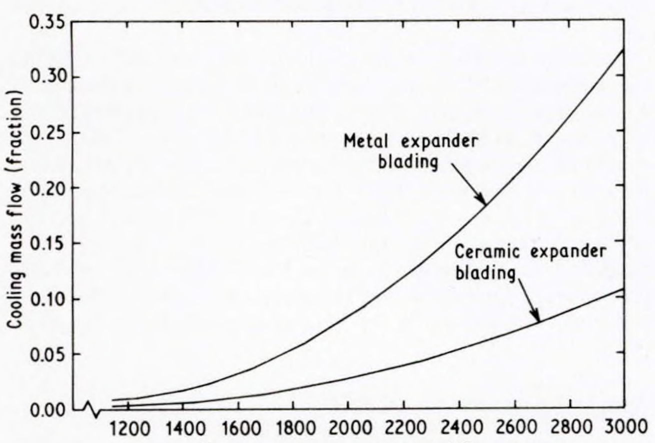

FIG. 2: Assumed cooling-air requirements as functions of turbine-inlet temperature

BACKGROUND

Despite the recent reductions in the absolute price of fuel, fuel costs remain a significant portion of the operating costs of commercial vessels. The long-term trends are for further increases in the price of fuel relative to other costs. The current study was undertaken to examine whether the principal engine used by fishermen today, the medium-speed diesel engine, could be surpassed, at least in fuel efficiency, by a turbine engine. Although the power levels considered are appropriate for relatively small vessels (about 1 MW), the conclusions regarding thermal efficiency are also applicable for larger and more powerful gas-turbine engines.

For comparison purposes, the power rating of the engine was specified at 1.1 MW (1500 hp), which was considered appropriate for the next generation of US fishing boats.[2] The calculated performance of the engine is compared with the performance of the medium-speed diesel engine normally used to power these vessels.

THERMODYNAMIC CYCLE STUDIES

Selection of an appropriate thermodynamic cycle is fundamental to the gas-turbine design process. This section summarizes the results of a preliminary comparison made among the simple Brayton cycle (CBE) and three proposed modifications that include adding a heat exchanger (X) or an intercooler (I) (CBEX, CICBEX) and combining a CBE with a variant known as an inverted cycle, the combination being called direct-plus-inverted cycle (DIC).[1] By making realistic approximations to component efficiencies and operating limits, overall estimates of design-point thermal efficiency and specific power have been obtained for each cycle, solely on the basis of thermodynamic considerations. These estimates have been used to compare the cycles and the potential gains they can offer for marine propulsion. This analysis served as groundwork for the preliminary design study of an engine operating on the CBEX cycle.

A computer code was used to calculate the performance characteristics for a wide variety of cycle configurations, given specified cycle parameters and component efficiencies. The strategy was to specify component-performance parameters representative of currently available hardware in the size and cost range of interest and to include upgraded performance estimates in areas where recent experimental evidence indicates possibilities for improvement over the next few years. To be specific, increases in the maximum permissible turbine-inlet temperatures of marine and industrial engines are expected with continued development of blade cooling and of ceramic components, and regenerator effectiveness can be made to exceed current levels if volume and weight constraints are relaxed. However, significantly improved aerodynamic efficiencies of small compressors and turbines cannot reasonably be expected in the near future, nor can we expect to see efficient high-pressure-ratio engines built at moderate cost.

In current advanced turbine engines the turbine-inlet temperature is higher than the materials can withstand. In order to reap the benefits of high turbine-inlet temperatures various methods of cooling the blades to the appropriate temperatures have been devised. The most common of these methods is to cool the blades with cooler air extracted from the compressor. This cooling-air flow reduces the potential expansion work that can be extracted from the turbine. Naturally there is a trade-off between the increases in turbine-inlet temperature and the amount of cooling-air flow that can be used to advantage. In the computer code the cooling-air flows for metallic turbine expanders were modelled[5] as a function of turbine-inlet temperature, as shown in Fig. 2. The corresponding requirements for expanders with ceramic blading were conservatively modelled as one-third of the requirements for metallic turbines at the same turbine-inlet temperature.

For reasons that will be fully explained below, it was also decided that a highly effective rotary ceramic regenerator would be used for regenerative cycles with pressure ratios below 6:1. A sketch of a rotary regenerator is shown in Fig. 3. Some of the working fluid leaks around the seals of these regenerators. Since the size of the seals increases with effectiveness, the mass leakages around the seals of the regenerator were modelled as a function of effectiveness, as shown in Fig. 4. (In this study a regenerator effectiveness of 0.975 was used and it was assumed that the corresponding leakage flows from the cold and hot sides of the regenerator were each 2.5% of the main flow.)

A more complete discussion of gas-turbine cycles can be found in one of the standard references.[1,4,6,7] The following discussion is intended to facilitate interpretation of the results that follow and to introduce some important definitions.

The net power produced by any of the following cycles is given as the difference between the power produced during expansion and the power absorbed during compression increased by one percent (the latter for auxiliary drives and other uses). Losses in the non-ideal cycles appear as entropy increases during the compression and expansion processes and as pressure drops in various other components such as ducts, burner, heat exchanger, intercooler etc., all of which can be seen to decrease the net power of the cycle.

Two quantities of key interest (used as ordinate and abscissa in the following performance plots) are the thermal efficiency, defined as the power output of the cycle divided by the rate of energy addition during the combustion process, and the specific power, defined as the power output of the cycle normalized by the product of the mass-flow rate, specific-heat capacity and stagnation temperature at inlet. The thermal efficiency is an approximate measure of the fuel efficiency of the engine. Other losses such as bearing and disc friction are not included in this definition and will reduce the brake fuel efficiency by perhaps 1–5%. The specific power is a measure of the power produced per unit mass flow and can be regarded as an approximate measure of relative engine volume and weight.

In the performance plots of Ref. 2, cycle parameters were specified that were functions of pressure ratio, temperature and other variables. In this study as uniform parameters as possible were specified for all cycles. Although the performance parameters are modified by small amounts, the trends are repeated and the conclusions of Ref. 2 and of this paper are the same. In producing the performance plots included in this paper the following parameters were specified.

1. Total temperature at compressor inlet 300 K.
2. Coolant temperature at intercooler or waste-heat-recovery-boiler inlet 300 K.
3. Compressor total-to-total polytropic efficiencies of 0.90.
4. Expander total-to-total polytropic efficiencies of 0.90.
5. Intercooler effectiveness of 0.90.
6. Heat-exchanger effectiveness of 0.975 (see below).
7. Burner efficiency of 0.996.
8. Cooling-mass-flow fraction from compressor delivery as shown in Fig. 2.
9. Fraction of flow leaking from cold side of regenerator is 0.025.
10. Fraction of flow leaking from hot side of regenerator is 0.025.
11. Fraction of flow leaking from compressor delivery to atmosphere is 0.02.
12. Sum of pressure losses for CBE are 7% of the compressor pressure ratio.
13. Sum of pressure losses for CBEX cycle are 12% of the compressor pressure ratio.

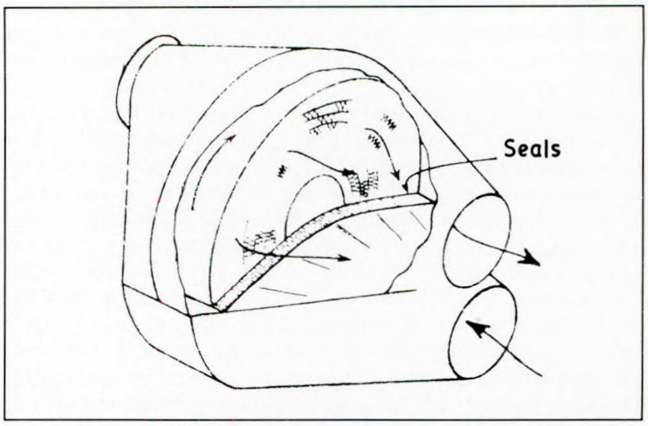

FIG. 3: Typical rotary regenerator (from Ref. 1)

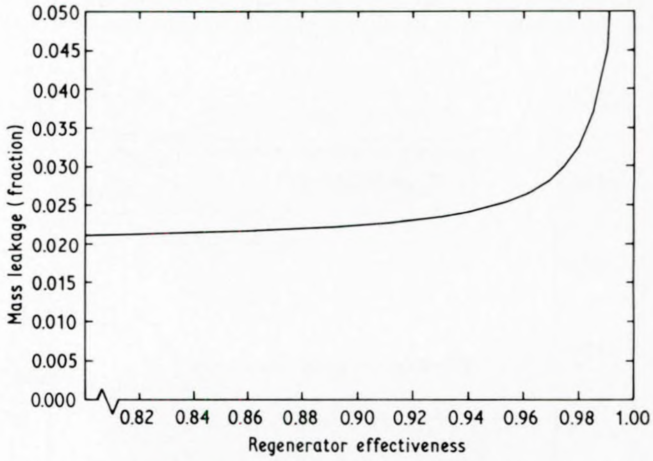

FIG. 4: Assumed variation of regenerator leakage flows

14. Sum of pressure losses for CICBEX cycle are 14% of the compressor pressure ratio.
15. Temperature ratios T' of 4, 5, 6 and 7.
16. Pressure ratios r of 4 to 100 in increments of 4 for the CBE cycles.
17. Pressure ratios of 2 to 16 in increments of 1 for the CBEX and CICBEX cycles.

The cycle parameters chosen for the regenerators have recently been verified by experiments. For example, it is reported[8] that the measured effectiveness of rotary ceramic regenerators was 93.3% to 98.7%, that the measured pressure drops were from 2.5% to 5.4% and that the measured leakages were 3.4% with NiO/3O CaF_2 wearface and 4.5% with cooled seals.

Performance plots were produced for the CBE, CBEX, CICBEX and DIC cycles using the high-cooling and the low-cooling flow fractions in Fig. 2. The effect of the cooling flow is to reduce the available energy at the turbine inlet. This drop in availability is modelled in the computer program by penalizing the cycles in two ways. First, the turbine-inlet temperature is reduced by the energy balance between the main flow and the cooling flow. Secondly, the resulting increase in entropy is modelled as a pressure drop at the turbine inlet (see the temperature against entropy diagrams that follow). This pressure drop is included in the sum of pressure losses for each cycle.

Simple Brayton cycle (CBE)

All the cycles that will be discussed in this section are derived from the simple (CBE) cycle. The path of the working fluid through the components and the temperature against entropy diagram are shown in Fig. 5. In its ideal form the CBE cycle comprises isentropic compression, followed by heat addition at constant pressure, followed by isentropic expansion to ambient static pressure.

The thermodynamic availability of a working fluid at any point in a cycle is a measure of the ability of the working fluid to deliver work by coming to thermodynamic equilibrium with the environment.[9] In the simple cycle the thermodynamic availability of the hot turbine exhaust is wasted. Increasing the pressure ratio of the cycle (for fixed turbine-inlet temperature) reduces the exhaust temperature, thereby increasing the thermal efficiency.

The thermal efficiency of a simple cycle continues to increase with pressure ratio until the benefit of reduced exhaust temperature is balanced by increased compressor power consumption, at which point an optimum pressure ratio is reached.[1] The optimum pressure ratio turns out to be quite high, ie greater than 20:1, as illustrated by the CBE cycle performance plots shown in Fig. 6 (where the pressure ratio increment is 4).

Low-pressure-ratio regenerated cycle (CBEX)

The CBEX cycle is a modification of the CBE cycle. In this cycle the thermodynamic availability of the turbine exhaust temperature is 'transferred' from the turbine exhaust to the compressor exit via a heat exchanger. The path of the working fluid and the temperature against entropy diagram for this cycle are shown in Fig. 7. The addition of the heat exchanger results in much lower optimum pressure ratios for the regenerative cycle compared with the simple cycle. A CBEX cycle incorporating 'perfect' components can be shown to have an optimum pressure ratio of unity; if a highly effective (roughly 95% or greater) heat exchanger is used, and assuming typical temperature ratios and component efficiencies, optimum pressure ratios in the range from 2:1 to 6:1 are found. (The effectiveness of the heat exchanger is defined as the actual energy transferred between the two fluids divided by the maximum energy that could have been transferred between the two fluids without the expenditure of additional work.)

Maximum effectiveness for gas-turbine heat exchangers has risen rapidly,[2] as shown in Fig. 8. The highest figure is just over 0.95 for in the Allison GT 404 engine. This engine uses twin ceramic discs of moderate size.[10] The effectiveness of such regenerators could be increased to 0.975 by doubling the thickness of the ceramic discs. A limiting pressure ratio of 6:1 is generally applied to rotary regenerators.[11] Since the expected optimum pressure ratio for the CBEX cycle was well below 6:1, the regenerator effectiveness was specified at 0.975. Another reason for the specification of a ceramic regenerator is that the maximum possible inlet temperature for a metallic heat exchanger is currently below 1000 K. Since the temperature drop in the low-pressure-ratio expanders considered may be as low as 250 K, metallic heat exchangers could be used only with either high-pressure-ratio expanders or with low-temperature-ratio cycles, leading to lower efficiency in both cases. The performance plots obtained with the above cycle parameters are shown in Fig. 9 (where the pressure ratio increment is 1). As expected, the optimum pressure ratios are around 3:1.

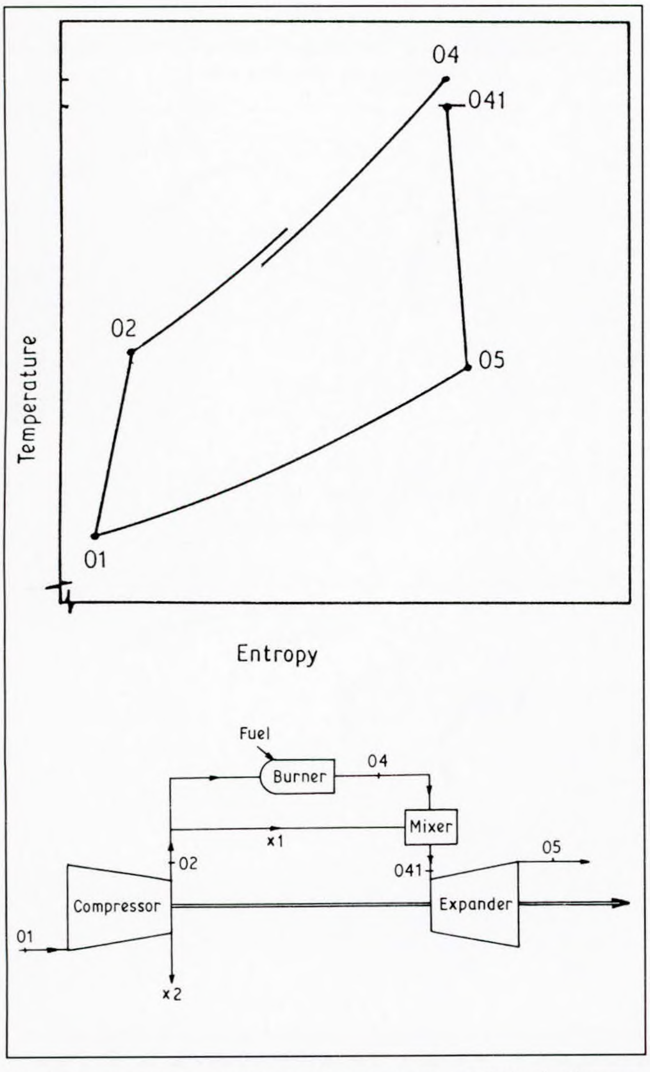

FIG. 5: Plot of temperature against entropy and block diagram for simple cycle

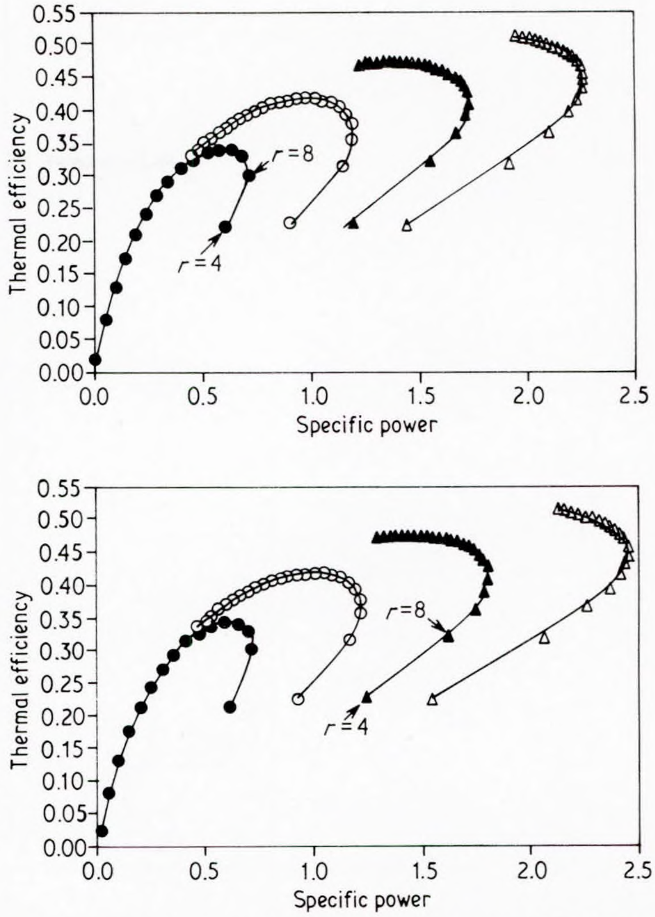

FIG. 6: Plot of simple cycle thermal efficiency against specific power for large (upper) and small (lower) cooling-mass-flow fraction. ●, $T' = 4$; ○, $T' = 5$; ▲, $T' = 6$; △, $T' = 7$

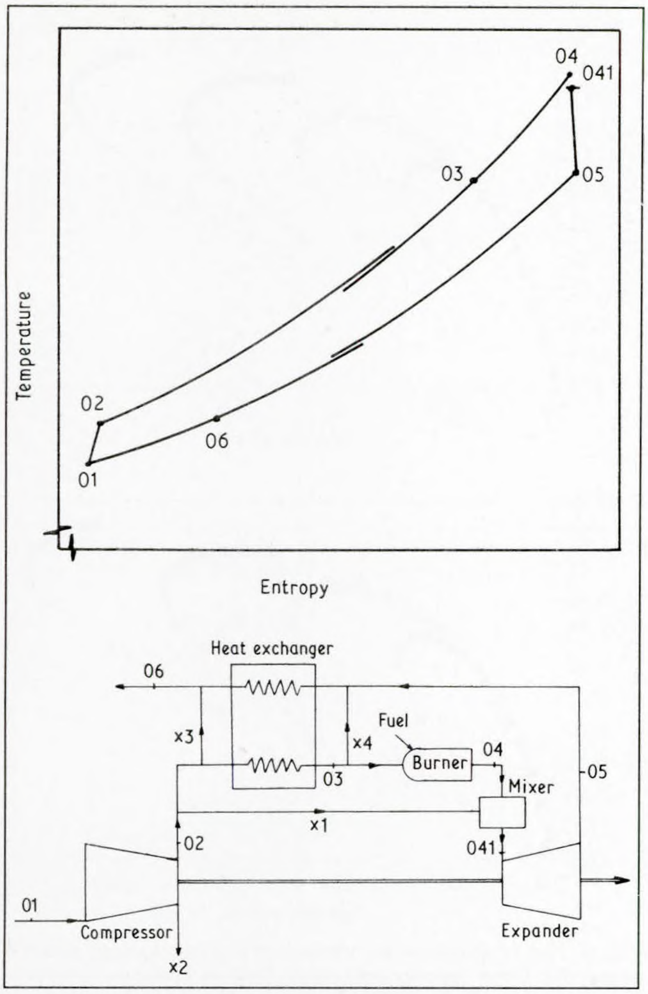

FIG. 7: Plot of temperature against entropy and block diagram for heat-exchanger cycle

Intercooled regenerated cycle (CICBEX)

The thermal efficiency of the CBEX cycle can potentially be improved still further by compressing in two or more low-pressure-ratio compression groups and cooling to near ambient temperature between groups (CICBEX). This practice of intercooling between compressors takes advantage of the fact that the compression work required for a specified pressure ratio is directly proportional to the inlet temperature.

The path of the working fluid and the temperature against entropy diagram for this cycle are shown in Fig. 10. For the CICBEX cycle, 90% intercooler effectiveness was specified. This performance should be readily attainable given the abundance of sea water for use as a cooling medium. The performance plots obtained with the above cycle parameters are shown in Fig. 11 (where the pressure ratio increment is 1).

Direct-plus-inverted cycle (DIC)

In the direct-plus-inverted cycle additional work is obtained by extending the expansion process of the simple cycle to subatmospheric pressure, cooling the gas to near ambient temperature and then recompressing to atmospheric pressure. The thermal efficiency of a simple cycle can be improved by the addition of this subatmospheric 'inverted' cycle, provided of course that the additional work of over-expansion exceeds the work required to recompress back to atmospheric pressure. The path of the working fluid and the temperature against entropy diagram for this cycle are shown in Fig. 12.

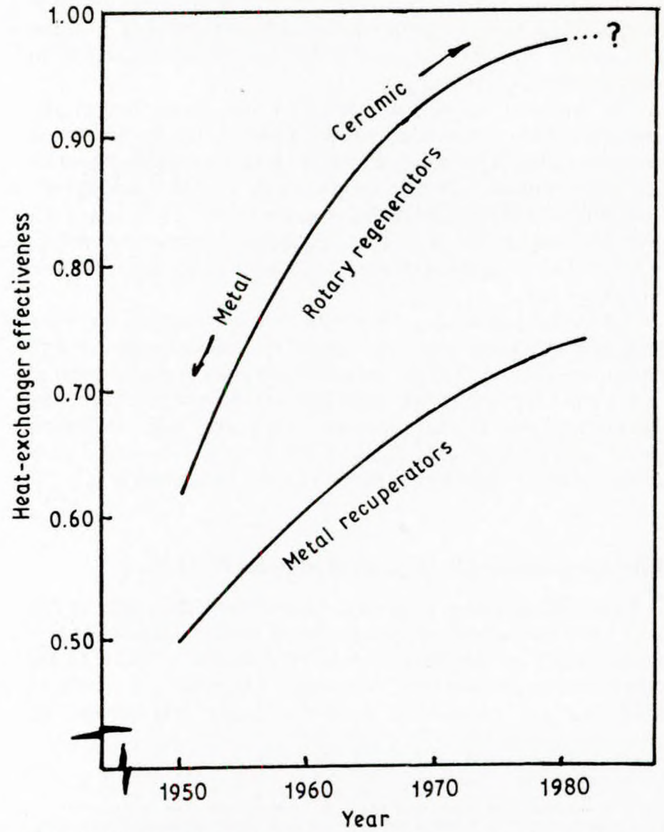

FIG. 8: Historical and projected increase of maximum regenerator effectiveness

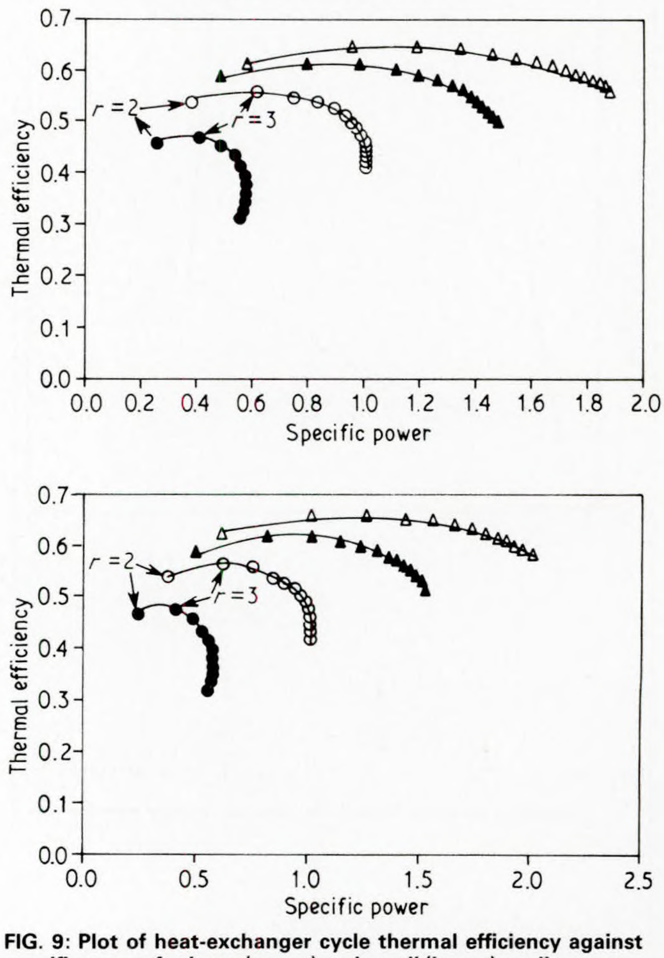

FIG. 9: Plot of heat-exchanger cycle thermal efficiency against specific power for large (upper) and small (lower) cooling-mass-flow fraction. ●, T' = 4; ○, T' = 5; ▲, T' = 6; △, T' = 7

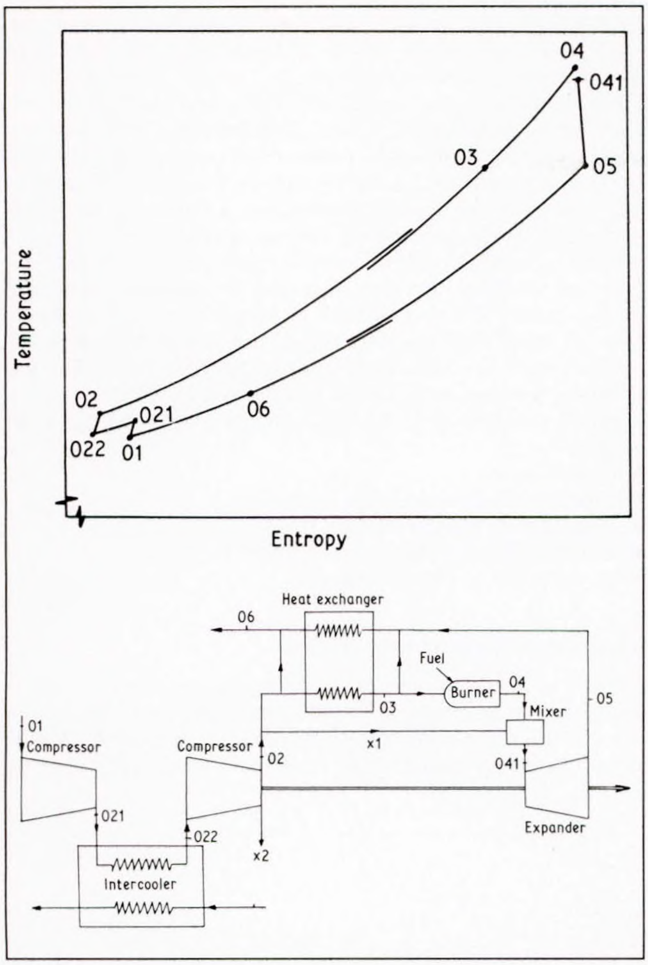

FIG. 10: Plot of temperature against entropy and block diagram for intercooled cycle

Evaluating direct-plus-inverted cycle performance is less straightforward, and the designer has wide freedom in specifying inverted-cycle components, depending on the intended application and economic constraints. This investigation is restricted to DIC cycles with low-cooling-mass-flow fractions (see Fig. 2). An optimum inverted-cycle pressure ratio (for maximum thermal efficiency) can be found corresponding to a specified set of simple-cycle conditions. For a non-ideal cycle the evaluation of this optimum pressure ratio must be accomplished numerically. At a fixed temperature ratio this optimum pressure ratio is found to decrease monotonically with increasing simple-cycle pressure ratio, as shown in Fig. 13. The resulting DIC cycle performance plot is shown in Fig. 14 (where the pressure ratio increment is 1).

Comparison of the proposed cycles

In Table I the cycles have been ranked according to thermal efficiency, specific power and pressure ratio as functions of temperature ratio. The most efficient cycle is the low-cooling-mass-flow fraction CICBEX cycle, followed by the high-cooling-mass-flow fraction CICBEX cycle, the low-cooling-mass-flow fraction CBEX cycle and the high-cooling-mass-flow fraction CBEX cycle. The DIC and CBE cycles exhibit the lowest efficiency. The DIC cycle is not substantially ahead of the CBE cycle except at rather high temperature ratios, which raises serious questions as to the economic attractiveness of the scheme for new engine development. The direct-plus-inverted cycle nonetheless retains its appeal as an add-on to existing simple-cycle engines. Thus from the standpoint of thermal efficiency, the regenerated cycles look very attractive, whereas the DIC and CBE cycles are not very interesting.

The advisability of intercooling now becomes an issue. The use of intercooling is seen to enhance thermal efficiency, provided regenerator performance and compressor polytropic efficiency are unimpaired. The increased optimum pressure ratios indicated in Table I are bound to degrade regenerator performance. The much smaller blade lengths in the high-pressure compressor casing after an intercooler have relatively larger blade clearances, which will result in reduced efficiency. Additionally, intercooling increases the complexity and cost of the cycle. It is unlikely that an engine manufacturer would be persuaded to tackle all these cycle additions at once. It was decided that, until the case of intercooled-regenerative cycles was proven, effort should concentrate on the relatively simpler CBEX cycle.

Preliminary analysis of the CBEX cycle can be used to quantify the very substantial gains in thermal efficiency over that of the simple cycle at pressure ratios appropriate for high-efficiency engines. Other major advantages not made evident by preliminary cycle analysis are the reduced cost of manufacture associated with a low-pressure-ratio engine and the opportunity to design for low blade stress, which favours the reliable operation of ceramic turbines. Compromises take the form of reduced specific power and the need to use a heat exchanger, both of which contribute to increased engine weight and volume compared with simple cycles. These size and weight penalties render the proposed cycles inappropriate for aircraft, which is the reason for the reluctance of the major gas-turbine engine manufacturers to commit private funds to the investigation of these cycles. However, even with these size and weight penalties, the resulting gas-turbine engines would be considerably smaller than the equivalent diesel engines.

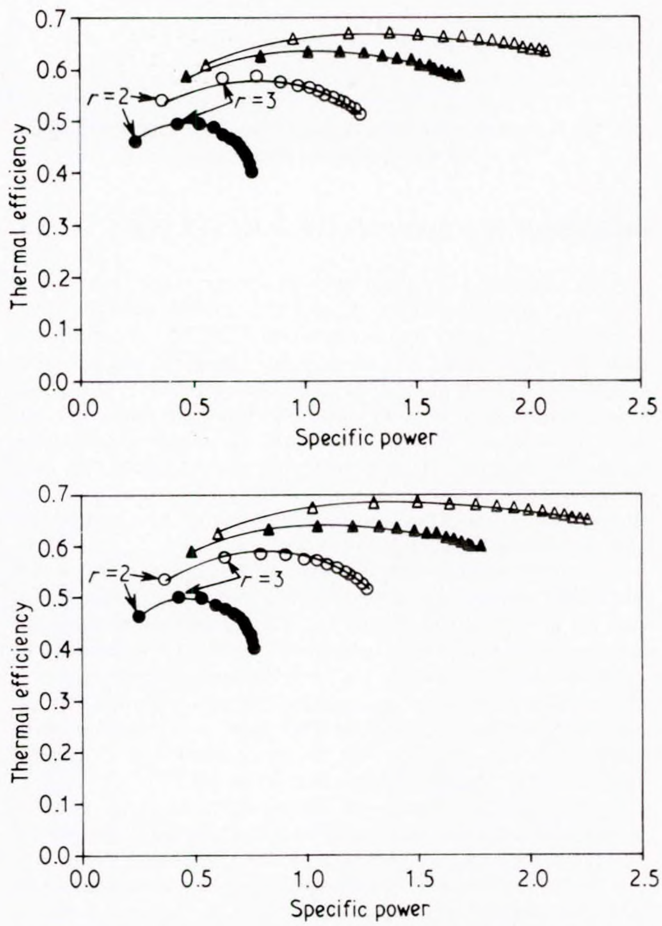

FIG. 11: Plot of intercooled cycle thermal efficiency against specific power for large (upper) and small (lower) cooling-mass-flow fraction. ●, $T' = 4$; ○, $T' = 5$; ▲, $T' = 6$; △, $T' = 7$

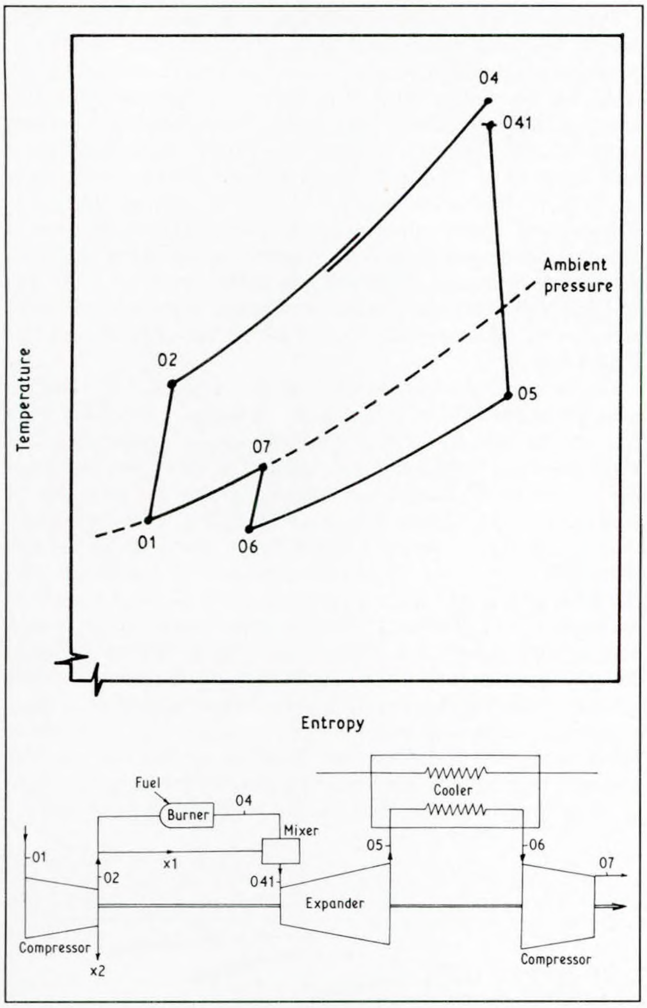

FIG. 12: Plot of temperature against entropy and block diagram for direct-plus-inverted cycle

PRELIMINARY DESIGN OF A BASELINE ENGINE

A preliminary design was undertaken for two reasons. The first was to provide fishermen, engine manufacturers or others interested in engines operating on the CBEX cycle and rated around 1 MW with a set of example specifications covering overall size, speed, number of stages and so forth. While a final design might vary in some respects, the differences are not likely to be major. The second reason was to provide a specific engine for analysis, particularly with respect to its part-load performance and the comparison of its performance with the performance of a diesel engine. This engine was named the LPR (for low pressure ratio) engine.

Currently turbine-inlet temperatures are over 1200 K for uncooled turbines and up to 1800 K for cooled turbines (from Fig. 1). For the LPR engine we limited the combustor-exit temperature to 1555 K, which for an ambient temperature of 300 K makes T' about 5.2. This combustor-exit temperature is typical of current naval gas-turbine engines with metal blading[12] and therefore allows the LPR engine to be designed with metal or ceramic blading. As discussed above we specified a rotary ceramic regenerator of effectiveness 0.975.

The turbine rotor blades are the most critical components in gas turbines because they must withstand the impingement of the high-temperature combustion gases at very high velocities. The higher the temperature the gases can be allowed to reach, the higher will be the cycle efficiency and the higher the engine power output. An enormous research effort in many countries has gone into improved metallurgy, effective air and even water cooling, ceramic coatings of metal blades and, in the past few years, the use of ceramic and other non-metallic materials from which vanes, blades, discs and combustor liners can be made. Some small turbines (aircraft auxiliary power units) are now being produced with ceramic 'hot parts' (although not, so far as is known, with ceramic rotors) but some research engines are running with ceramic rotors. A major effort is also underway in several countries to produce ceramic turbochargers.[13] In view of the rewards in higher efficiencies and lower production costs, it seems very likely that success will not be far off for both the turbocharger and the turbine applications.

From inspection of the above performance figures we deduced that, for thermal efficiency in excess of 50%, the design of the LPR engine must be aimed at two goals. First, high T' values (which necessitate the use of cooled metal turbine blades or uncooled ceramic blades). Secondly, the optimum pressure ratio for the selected value of T'.

Based on the above we chose the following for the LPR engine.

- CBEX cycle.
- Compressor pressure ratio of about 3:1.
- Regenerator effectiveness of 0.975.
- Combustor-exit temperature of 1555 K.
- Rated power 1.1 MW (1500 hp).

Compressor preliminary design

The low pressure ratio of the LPR engine allows considerable freedom in design. A single-stage centrifugal compressor would have a peak polytropic efficiency of about 0.87. Multi-stage axial compressors would be more efficient for this power

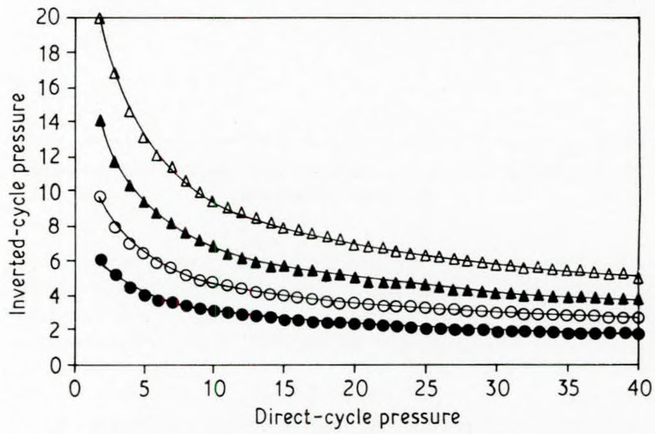

FIG. 13: Plot of optimum pressure ratios for inverted cycles as functions of direct cycle pressure ratios. ●, T' = 4; ○, T' = 5; ▲, T' = 6; △, T' = 7

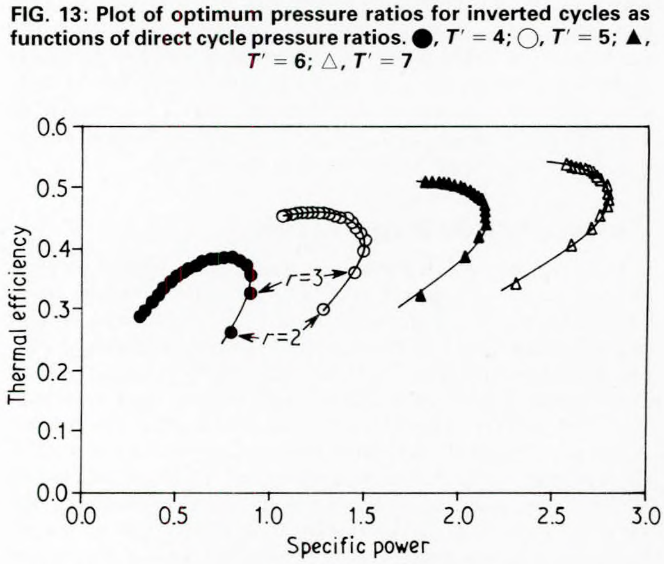

FIG. 14: Plot of direct-plus-inverted cycle thermal efficiency against specific power. ●, T' = 4; ○, T' = 5; ▲, T' = 6; △, T' = 7

Table I: Summary of performance of different cycles

	$T' = 4$	$T' = 5$	$T' = 6$	$T' = 7$
		Pressure ratio		
CBE large	20	20*	20*	20*
CBE small	20	20*	20*	20*
DIC	14	20*	20*	20*
CBEX large	3	3	3	4
CBEX small	3	3	3	4
CICBEX large	3	4	4	4
CICBEX small	3	4	5	5
		Thermal efficiency		
CBE large	0.3381	0.3915	0.4087	0.4122
CBE small	0.3395	0.3922	0.4094	0.4134
DIC	0.3834	0.4524	0.4886	0.5065
CBEX large	0.4654	0.5553	0.6105	0.6409
CBEX small	0.4711	0.5610	0.6156	0.6568
CICBEX large	0.4944	0.5837	0.6294	0.6657
CICBEX small	0.4965	0.5823	0.6361	0.6799
		Specific power		
CBE large	0.5845	1.1701	1.7355	2.2384
CBE small	0.5909	1.1919	1.8058	2.4129
DIC	0.7687	1.3522	2.0747	2.7504
CBEX large	0.4138	0.6171	0.8013	1.1853
CBEX small	0.4177	0.6305	0.8357	1.2817
CICBEX large	0.4294	0.7868	1.0129	1.2054
CICBEX small	0.4335	0.8009	1.2091	1.4973

A maximum pressure ratio of 20 was specified because the costs and difficulties of compressor development are very high at higher pressure ratios. Cycles for which the optimum pressure ratio for maximum thermal efficiency is greater than 20 are marked with an asterisk (*), and data for a pressure ratio of 20 is presented. For cycles for which the optimum pressure ratio for maximum thermal efficiency is less than 20, data for the optimum pressure ratio is presented.

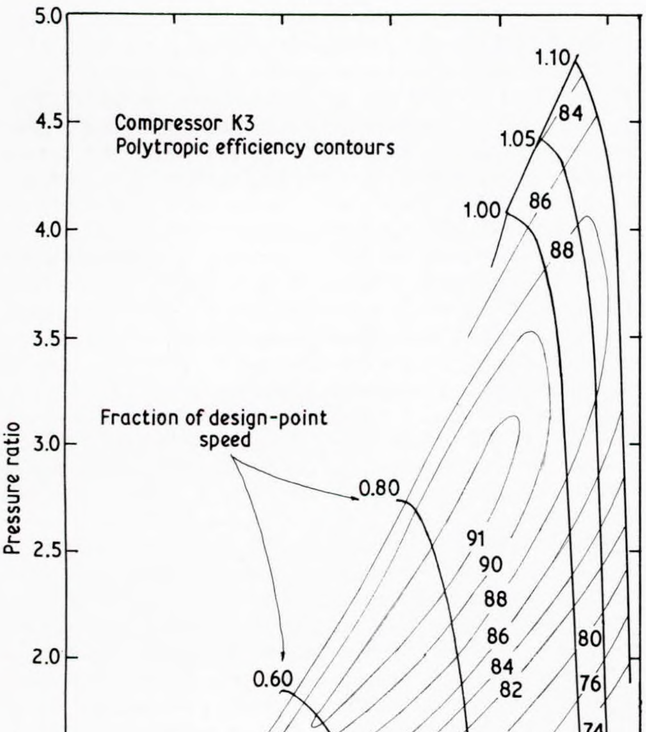

FIG. 15: Predicted compressor characteristics with contours of polytropic efficiency

range. Preliminary design of these compressors is performed by choosing vector diagrams relating the inlet and outlet flow directions and the blade velocities through the stages. Characteristics of vector diagrams are the work coefficient, flow coefficient and reaction.[1] In the design process one must also choose the blade speed. The design-point and off-design-point performances of three compressors for the LPR engine have been studied.[14]

1. A 50%-reaction compressor with a mean blade speed of 350 m/s.
2. A 50%-reaction compressor with a mean blade speed of 275 m/s.
3. A 100%-reaction compressor with a mean blade speed of 200 m/s.

To reach a pressure ratio of 3:1 the first compressor required four stages, while the second and third compressors required six stages (in effect the second and third compressors trade lower blade speed for number of stages). Our calculations indicated that the third compressor was possibly 1% less efficient than the other compressors at the design point but was more efficient at off-design points. The conclusion is that high-reaction compressors are more advantageous than 50% reaction compressors at off-design-point operation. Similar conclusions were reached in Ref. 15, although for different reasons. Since marine propulsion requires that engines operate frequently at off-design points we chose to use the third compressor in our preliminary design.

The performance map of the third compressor was used in the calculations of engine performance that follow and is shown in Fig. 15. This calculated performance map is validated by the similarity of the results to tested compressor characteristics, for instance those of a Ruston and Hornsby compressor with a 5:1 design-point pressure ratio shown on page 316 of Ref. 1. This compressor would have a rotor-blade-tip diameter of 300 mm (11.8 in) and an overall length for the compressor of under 500 mm (20 in) including the diffuser. The shaft speed would be about 16 700 rev/min.

The above blade speeds are considerably lower than the current maximum blade speeds of about 400 m/s used in the industry. The low blade speed reduces foreign-object damage and steady-state blade stresses and increases the blade length, thus reducing relative clearance. This alone could well overcome any other efficiency disadvantage of the high-reaction design. The low blade stresses open up another attractive area of design freedom: the possibility of using reinforced polymer resins for the blades and vanes, possibly in low-cost moulding. Three materials identified[16] as having outstanding high-temperature fatigue and creep resistance are polyphenylene sulphide (PPS), polyetheretherketone (PEEK) and polyethersulphone (PES), reinforced with glass, Kevlar or carbon fibres. They should be excellent in a marine environment. Once the production equipment has been made, the manufacturing cost of such compressors should be relatively low.

Turbine preliminary design

Assuming that the LPR engine was designed with one axial-flow turbine directly coupled to the compressor and the output shaft, three stages (six rows of blades) with an outside diameter of about 450 mm (17.7 in) would be required. The shaft speed would be about 16 700 rev/min, giving very low turbine-blade stresses compared with conventional designs, and therefore providing favourable conditions for the application of non-metallic blades. A two-stage epicyclic reduction would probably be used if the engine was coupled to a controllable/reversible-pitch propeller.

For the calculation of design-point and off-design-point performance of the LPR engine it was assumed that the expander had the performance map shown in Fig. 16. This turbine performance was extracted from that shown on page 128 of Ref. 17, where the design-point conditions were very similar to those of the LPR engine.

Conceptual design of rotary regenerators

The turbine exhaust would pass into a ceramic regenerator. The usual arrangement for the small engines so far equipped with this type of heat exchanger is to use two ceramic discs, one on each side of the turbine. If this scheme were used for the LPR engine, the discs would be 1.75 m (69 in) in diameter and 136 mm (5.4 in) thick. At the present stage of production technology, a disc of this size would be manufactured by building up from smaller sections.[18] Each disc pair would be independently driven by a fractional-horsepower electric motor through a standard gear reduction and rim drive. The exhaust gases would leave the opposite faces of the discs and be ducted up the stack, perhaps giving up further heat to a waste-heat boiler.

Calculated performance of the LPR engine

The detailed design-point and off-design-point performance of the LPR engine was calculated with the aid of the computer code NEPCOMP (Navy engine performance computer program), otherwise called NEPII.[19] NEPCOMP can be used with or without component characteristic maps and consists of modules that represent engine components (compressor, regenerator, burner, turbine, duct) interconnected by flow-station numbers or other components (shaft, load). Calculations begin at the engine inlet and flow properties are computed at consecutive flow stations. A converged solution occurs when both equilibrium mass flow and horsepower balance are satisfied.

One of the inherent advantages of using NEPCOMP is that the off-design-point performance of the LPR engine can be calculated while using the actual performance map of each component. Naturally, the results are only as good as the model of the engine that is input in the code. The model of the LPR engine used in NEPCOMP is shown in Fig. 17.

The output of NEPCOMP was translated into a series of normalized plots that illustrate the performance of the LPR engine. T' contours are shown in Fig. 18 and thermal-efficiency contours in Fig. 19, from which it can be seen that the predicted thermal efficiency of the LPR engine (comparing design and off-design points) is higher than the thermal efficiency of all prime movers in use today.

Overall configuration

The overall arrangement of the engine, including a rotating-matrix regenerator, is shown approximately to scale in Fig. 20. The compressed air leaving the compressor would be ducted to sectors on the discs, pass through the matrix and be combined in the casing of a single combustor supplying the turbine. The combination of ducting, heat exchanger and combustor would probably be located above the turbomachinery line to allow easy access for servicing.

COMPARISON OF THE LPR ENGINE AND A DIESEL ENGINE

Performance comparison

It was decided to present the performance of the LPR engine and the diesel engine in the form of thermal efficiency, which is non-dimensional and independent of the heating value of the fuel. The usual specific-fuel-consumption (sfc) curves can be obtained from the equation:

$$\text{sfc} = 2545/(\eta_{th}\ \text{HVF})\ \text{(in units of lb/shp h)}$$

where η_{th} is the thermal efficiency of the cycle and HVF is the heating value of the fuel (in units of Btu/lb). Also

$$\text{sfc (units of lb/shp h)} \times 0.6083 = \text{sfc (units of kg/kW h)}.$$

The performance of the LPR engine was obtained using an HVF of 18 300 Btu/lb, which is a typical value for diesel fuel oil.

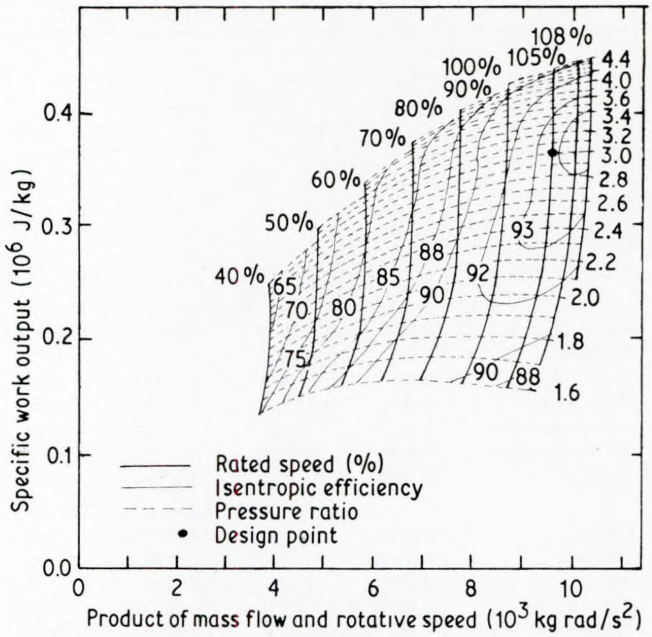

FIG. 16: Turbine characteristics with contours of isentropic efficiency

In a fashion similar to that described above for the LPR engine, the performance curves of three more regenerative gas-turbine engines of the same power output but of higher pressure ratios were also calculated.

1. Engine LPR1, with a design-point pressure ratio of 3.14:1.
2. Engine LPR2, with a design-point pressure ratio of 6.23:1.
3. Engine LPR3, with a design-point pressure ratio of 8.28:1.
4. Engine LPR4, with a design-point pressure ratio of 10.80:1.

The design-point and off-design-point performance of these gas-turbine engines was compared with that of a typical medium-speed diesel engine which in the current engine market powers fishing boats of the same size. The maximum thermal efficiency of this diesel engine is 35.7%.

Marine vessels are usually powered by fixed-pitch propellers (FPP) or controllable-pitch propellers (CPP). The performance of each engine was examined for three cases: coupled to an FPP, coupled to a CPP and coupled to a constant-speed drive. An example of a constant-speed drive is the case in which the engine drives an AC generator.

For the FPP case a cubic curve relating shaft power to propeller speed was specified. This would result in a linear relation between vessel speed (in knots) and propeller speed (in rev/min) (see page 93 of Ref. 20). For the CPP it was specified that it would receive the required power at the respective engine's optimum speed for maximum thermal efficiency at this power. For the constant-speed drive it was specified that the propelling device would receive the required power at design speed. Although this is a crude approach, it is a sufficiently accurate representation for comparison purposes.

The performances of the four gas-turbine engines and that of a current medium-speed diesel engine, each coupled to an FPP, a CPP and a constant-speed drive, are shown in Fig. 21. At this point it should be noted that at considerably higher power ratings slow-speed diesel engines can also attain design-point thermal efficiencies in excess of 50%. The performance of an LPR engine rated at the corresponding higher power ratings should be similar to the performance of the LPR engine shown here, or perhaps slightly improved because of the expected increase in component efficiencies (because the effect of clearances is reduced).

Comparison of the performance of the engines illustrates the thermal-efficiency advantage of the gas-turbine engines over a current diesel engine of similar power. The most efficient engine in all cases is engine LPR1, which at the design point operates at the optimum pressure ratio for the cycle (3:1). The thermal efficiency of the LPR engines is decreasing with increasing pressure ratio at design points because as the pressure ratio increases we move away from the optimum cycle conditions indicated in Fig. 9. Figure 21 shows that this is also observed at off-design-point operation of the engines.

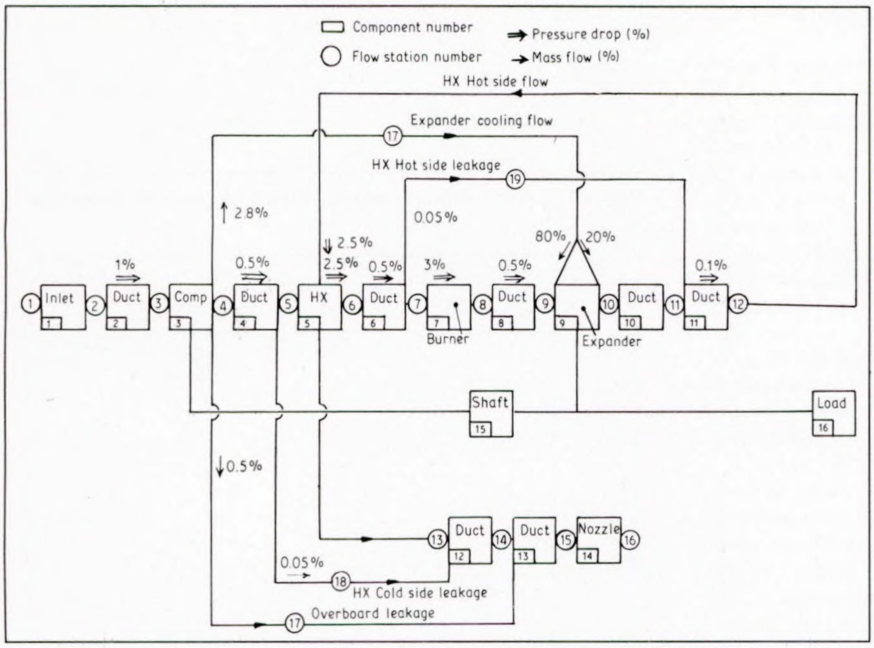

FIG. 17: Model of LPR engine used in NEPII calculations

Comparison of some other aspects of the two engines

The LPR engine appears to have the following advantages over diesel engines for commercial or military marine propulsion.

The LPR engine is smaller and more efficient than the diesel engine. Therefore for a given operation it requires less fuel than the diesel. This means that there is more weight and volume available in the vessel for allocation to payload. This results, in general, in more efficient operation. Alternatively for the same payload the LPR engine would permit the design of smaller ships that would have better arrangements and would require less propulsive power, thus economizing on fuel use twice.

Diesel engines cannot be started and run up to full power from a cold condition as they require a period of time, increasing with size, to warm up. The LPR engine could be started in minutes. The particular diesel engine used in this study is about 3.5 m long by 1.7 m wide by 2.1 m high and weighs dry about 8 tonnes. In comparison the LPR engine would have a six-stage compressor of about 0.30 m in tip diameter and 0.50 m long (including the diffuser) and a three-stage expander with outside diameter of about 0.45 m. The rotary regenerator would have two discs 0.4 m thick and 1.75 m in diameter. (Recent advances in rotary-generator technology may permit a single disc

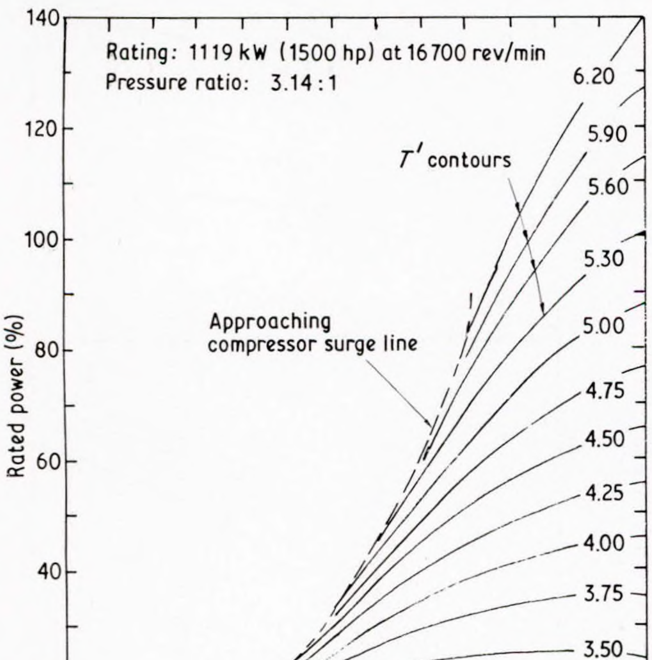

FIG. 18: Plot of turbine-inlet temperature (expressed as T') as function of power and speed

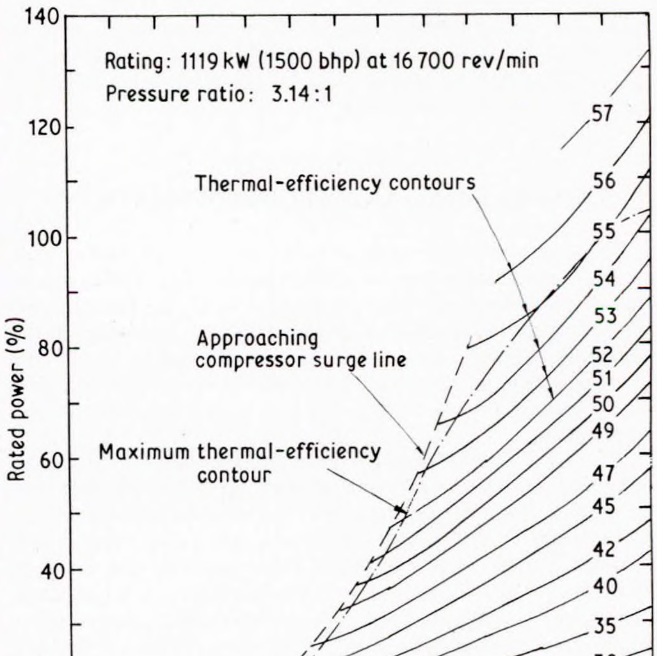

FIG. 19: Plot of thermal efficiency of the LPR engine as function of rated power and speed

of larger dimensions.) The weight of the LPR engine would be between 1 and 2 tonnes.

The LPR engine would be quieter and would give less vibration than the diesel engine, resulting in improved living conditions. The exhaust from the LPR engine would be less noxious than that of the diesel and because the engine is smaller it allows for greater flexibility in arrangements. Since the rejected energy from gas-turbine engines is concentrated in the exhaust the LPR engine is more compatible than the diesel to energy-recovery schemes. If the temperature of the exhaust was reduced by such methods the infrared signature of the vessel would also be considerably reduced.

The diesel engine requires smaller inlet and exhaust ducts than the LPR engine (but the difference was not large enough in the case of the baseline boat that was chosen for comparison purposes in Ref. 2 to affect the arrangements). Gas-turbine engines are sensitive to salt ingestion and therefore the air inlet must be protected from spray. Diesel engines in general operate at lower speeds than gas turbines. Special attention is required on the controls of the LPR engines to prevent overspeeding on sudden removal of the propeller load. Such load removals may occur if part of the propeller emerges out of the water in heavy pitching. Some aspects of how the use of a gas-turbine-engine prime mover affects the propulsion of marine vessels and the special controls required are discussed in Ref. 21.

The marine environment is severe, and there is consequently some risk associated with installing the first LPR engine in a boat because the engine will have been tried but will not have been proven by long use in any application or environment before, while the diesel engine has already been installed and successfully operated in many boats. Another undetermined factor is the reliability and maintenance requirements of the LPR engine. With a fully developed gas-turbine engine the crew would be required to do little more than simple maintenance such as lubricating oil, air- and fuel-filter replacement and possibly compressor cleaning by periodic spraying with distilled water or rice injection.

Finally, the capital cost of the LPR engine is not known. However, it is estimated that for mass production it will be comparable to or less than the capital cost of the diesel engine (Fig. 1.12 of Ref. 1), especially if the engine is designed with a reinforced-plastic compressor.

CONCLUSIONS AND RECOMMENDATIONS

The recommended prime mover is a low-pressure-ratio, highly regenerative Brayton-cycle (gas-turbine) LPR engine. The performance of this engine has been calculated at the design point and at off-design points. At the design point the thermal efficiency is about 55%. At off-design points the thermal efficiency remains higher than that of a corresponding diesel engine, the most efficient prime mover available for this application.

The LPR engine would have many advantages over the diesel engine. Some of these advantages are lower fuel consumption, lower weight, less space (and in consequence of the above increased payload), less noise and easily recoverable exhaust energy. However, the LPR engine has not yet been built, and it therefore has two disadvantages: lack of tested hardware (which would prove the above claims) and unknown maintenance requirements and reliability, which are impossible to predict.

While all new machinery is customarily introduced with promises of very low maintenance requirements, promises not always borne out in practice, the gas turbine in several duties, including marine service in the US and UK navies with highly rated aircraft-derivative units, has indeed required exceptionally low maintenance.[22] In naval duty it is generally the practice to exchange whole engines when anything greater than minor maintenance is needed. The small size and low weight of

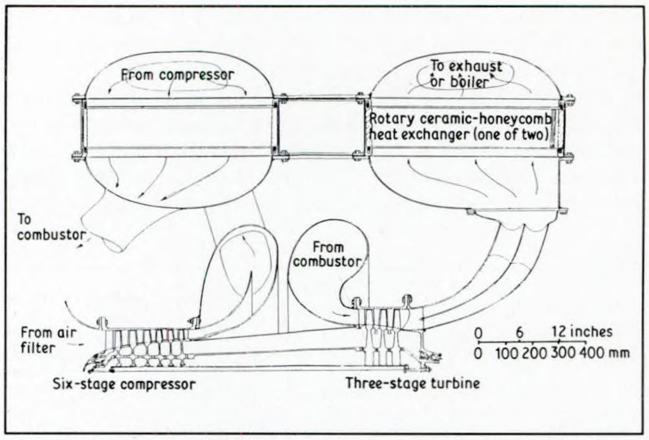

FIG. 20: Cross-sectional sketch of the LPR gas-turbine engine

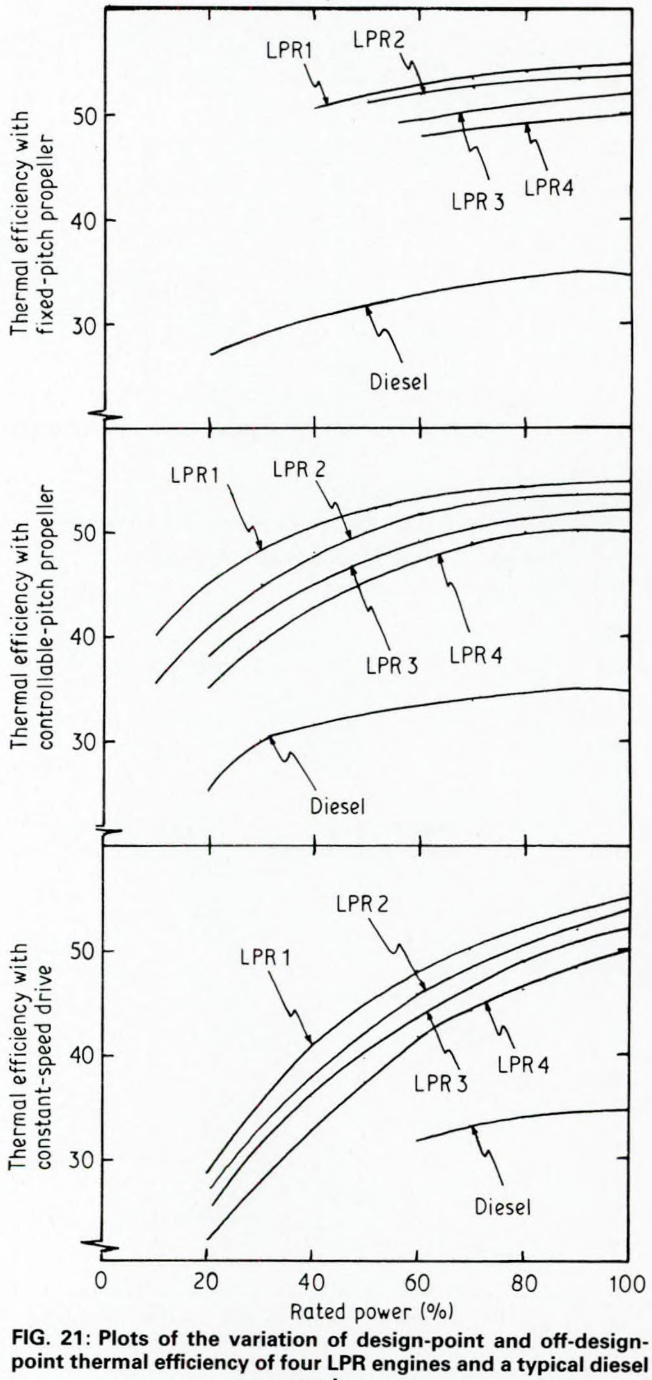

FIG. 21: Plots of the variation of design-point and off-design-point thermal efficiency of four LPR engines and a typical diesel engine

turbine units make them fairly easy to remove and replace, even during naval warfare.[23]

In the early 1960s the Royal Navy tried out the Rolls-Royce RM-60, a complex intercooled three-shaft engine, but did not pursue the concept further. Recently the US Navy has selected three companies (AiResearch, Allison Gas Turbines and Rolls-Royce) to undertake the conceptual design of a new naval intercooled engine.[24,25] Thus the time appears to be ripe for a reconsideration of the next generation of marine engines.

The low-pressure-ratio highly regenerative gas turbine has particular advantages for marine use. The low blade speeds required would enable non-metallic materials to be used with advantage, although the virtues of the cycle are not dependent on the use of non-metals. The design-point fuel consumption should be exceptionally good, and part-load consumption should be better than that of any current competitor. Engines of this type could be produced today (indeed, it could be said that the industry is moving cautiously toward this type of design) but developments in non-metallic materials, particularly in ceramics and ceramic-shielded graphite, would, if initial good reports of the resistance of ceramic coatings to sulphidation attack are further confirmed, make the engine even more attractive for marine use.

The concept of the low-pressure-ratio, highly regenerative cycle is not limited to the power rating of the LPR engine. Similar engines can be designed at different power levels, although the arrangement of the rotary regenerators may become more complex in larger power ratings.

Further development of these engines to the point of designing and building a prototype engine for testing is required, as is the design of a two-shaft engine to allow for greater load/speed flexibility and easier starting. Uses of the LPR engine may also prove advantageous to land-based installations where fuel efficiency is important.

A university group funded by public money to work in the area of gas-turbine propulsion cannot do much more than stimulate change and point out advantages and disadvantages of different technologies, because prototype-engine-development costs are enormous and usually not covered by research funds. We hope that an engine manufacturer will study this apparently attractive engine and produce some version of it for marine propulsion and other uses.

ACKNOWLEDGEMENTS

This research was part of a larger project entitled 'High-efficiency Brayton-cycle engines for marine propulsion' sponsored by the MIT Sea Grant College Program under grant number NA-81AA-D-00069 from the office of Sea Grant, National Oceanic and Atmospheric Administration, US Department of Commerce. The Sea Grant officers principally concerned with the program were Arthur B. Clifton and Clifford A. Goudey. John W. Gilbert helped us with the choice of the 'hypothetical vessel'. We express our sincere thanks to all the above.

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Discussion

Dr B. M. BURNSIDE (Heriot-Watt University): First I should like to congratulate the authors on an interesting paper. As they say, manufacturers of large slow-speed marine diesel engines quote overall thermal efficiencies of about 50%. Presumably this figure is attained using high-grade fuels and the use of lower quality oils would result in lower efficiency.

Could the authors please say what grade of fuel is used in the engines they quote at 35.7% thermal efficiency in their paper. Is it desirable to increase the refinement of fuel required by changing to gas turbine drives or should the trend be towards using lower quality oils in diesel engine drives?

The authors' views on using organic Rankine cycle (ORC) diesel exhaust and cooling water waste-heat recovery units to increase the efficiency of existing diesel drives would be interesting also. A study by Morgan and Davis[1] showed that the thermal efficiency of commercial diesel engines of that date could be raised to about 47% in this way.

1. D. T. Morgan and J. D. Davis, 'High efficiency decentralized electrical power generation utilizing diesel engines coupled with ORC engines operating on diesel reject heat'. Thermo-Electron Corp. Report No. NSF-RA-N-74-287 (Nov. 1974).

C. W. FREEMAN (Rolls-Royce plc): It is generally accepted that the gas turbine is the most suitable propulsion engine for large naval vessels requiring high installed power. Simple-cycle units are now well-established in service and the next generation are under consideration. A 30% improvement in fuel consumption is being sought and an intercooled/recuperated cycle is the most promising successor.

This paper makes a theoretical assessment as to whether the gas turbine is equally attractive for fishing boats requiring only 1 MW power units. From a comprehensive performance parameter study, the selected engine includes an exhaust heat exchanger and operates with design conditions of 3:1 pressure ratio and 1555 K temperature.

The authors call on industry to review this engine selection. It is only by making a detailed analysis of the cost of ownership, including reliability, that the optimum arrangement can be finally determined. Interestingly, the naval intercooled/recuperated engine proposal is now in this phase of assessment and the most thermally efficient cycle is unlikely to be the most cost-effective engine.

With regard to the 1 MW engine proposals, the following initial comments are offered:

1. For a small engine, a radial turbine and compressor concept would probably be the most cost effective.
2. It is agreed that increased cost and complexity make intercooling less attractive for the smaller engine.
3. Selecting a 3:1 pressure ratio gives optimum thermal efficiency but is well off the optimum specific power level of 8:1. A high specific power would not only favourably influence the size of the gas turbine and hence its first cost but also the regenerator and installation ducting size. In competition with the diesel, the gas turbine is particularly vulnerable on its complete installation volume.
4. 1555 K with a 3:1 pressure ratio produces a regenerator gas-entry temperature around 1250 K, which certainly rules out using metallic heat exchangers and ducting. Selecting a higher pressure ratio would lower the regenerator entry temperature and reduce the level of advanced technology.
5. It is likely that selection of a regenerator with 0.975 effectiveness would become unattractive when weight, volume and first cost are taken into account. A level nearer 0.85 is more practical. This would bring the pressure ratio for optimum thermal efficiency to around 6:1, which is close to the value for optimum specific power.
6. It is possible to achieve relatively flat gas turbine part-power thermal efficiency curves which are competitive with the diesel. However, using typical component characteristics it has been possible to obtain curves similar to those reported only by the introduction of variable geometry.

The above comments are given constructively on a very professional paper. The suggestions reflect a practical and possibly conservative industrial viewpoint. In summary, if a 1 MW engine was to be launched today, it would probably be of radial design with exhaust heat exchanger, 1400 K and 6:1 pressure ratio.

H. WATSON (Watson Engineering Consultants Ltd): This paper is a clear account of a useful study. My fears are that the 50% thermal efficiency which is forecast for CBEX will not be reached, so that a lower figure would be realistic to compare with the 35% level for diesels. These fears are based on:

1. A tendency to breakdown in the equivalent development stages of land-based gas turbines.
2. A further worsening of reliability if cruder oils are burned and thus a liking for expensive fuel.
3. Proportionally more fall-off of efficiency at light loads than with diesels, although here the authors' figures seem to suggest not too bad a performance for CBEX away from the design point.

Any development will show improvement with time, but the use of high temperatures and ceramics suggests some ultimate write-off of efficiency against outage and maintenance. This is certainly so if the comparison is made with steam plant. The title implies the cycle is being judged for marine plant as a whole, not just for the small 1.1 MW plant for fishing boats on which it centres. Thus the comparison for the upper end of the power range should be with the steam cycle.

The feature which gives me most concern is the heat exchanger. This tends to dominate the arrangement, both dimensionally and in the strong dependance on it to achieve high cycle efficiencies. Also it could well be the critical factor in reliability, being a large, high-temperature device subject to gyroscopic action from sea motion and yet critically depending on the effectiveness of seals, presumably with fine clearances. Also, as the CBEX grows to more generally useful powers, would the rotary heat exchanger become even more of a problem? It seems to me that it is the heat exchanger which should receive the main emphasis of development effort.

CBEX appears to be fairly compact compared with the diesel quoted. Taking the data given on page 11 and Fig. 20, a rough estimate based on engine envelope gives 4 kW/ft^3 for CBEX as against 2.5 kW/ft^3 for the diesel.

The title poses the question 'the next generation of marine prime movers?' It all depends of course on the power output. For the small sizes of around 1 MW there could well be a place for CBEX given that the difficulties mentioned are overcome. The popular middle range engine is surely some diesel derivative. However, for the top range, say 20 MW and much higher, I believe the choice may well be a practical combination of steam and gas turbines in a combined cycle and, in the long term, burning a coal-derived fuel. Thermal efficiencies would not be at the 50% level but would be substantially higher than for the steam cycle alone. They would also be competitive against the diesel, more so because of the cheaper fuel source.

I think that it is not too early to explore this prospect again.

Prof. D. E. WINTERBONE (University of Manchester Institute of Science and Technology): The authors have presented an interesting analysis of the possibility of replacing the low- and medium-speed diesel engines as a prime mover in marine applications. The engine proposed is quite similar to gas turbines that have been considered for use in road traction. In the late 1960s there was a major lobby in the truck industry to introduce low-pressure-ratio regenerated gas turbines as the vehicle power plant. This seemed to become necessary because of the need for ever increasing powers within a relatively small volume. A large amount of development work was undertaken by many of the large automotive companies, including Leyland (in the UK) and Ford and General Motors (both in the USA). All of these engines incorporated regenerators to achieve a satisfactory thermal efficiency, to enable them to compete with the diesel engine.

The difference between the automotive gas turbine and the one proposed here is that the former was a two-shaft device, comprising a gas generator and power turbine. This had the benefit of decoupling the turbine driving the wheels from the gas generator unit, and also enabled variable guide vanes to be fitted between the gas generator and the power turbine. The variable geometry was included to improve the part-load efficiency of the engine.

A large number of these engines were constructed and they were fitted into various mobile test beds, including road tankers, buses, and even railway locomotives. While they achieved a good performance at full load, I think it is true to say that they never lived up to their expectations at part-load. The other problem that was encountered was the life of the regenerator, and the locomotive engines were run without regenerators for part of their life, simply to overcome the problems associated with them.

It will be interesting to see whether the turbine proposed by the authors encounters similar drawbacks in manufacture and endurance, but perhaps materials have moved on sufficiently since the mid-1970s to remove these difficulties.

I wonder if the authors have considered the possibility of using a two-shaft gas turbine with variable geometry as plant for their ships. It might remove the necessity for expensive high-speed gearing and give some benefits in flexibility of operation, enabling astern drive to be obtained by moving the turbine nozzles. I realize that in the case of a controllable-pitch propeller this would be achieved by putting the propeller pitch into the astern position. The thermodynamic advantage that could be obtained would be the capability of maintaining a reasonably high turbine-inlet temperature under operating conditions. Again, the benefits of variable geometry could perhaps be incorporated without the need for a two-shaft device.

B. HAMMOND: I found the section entitled 'Thermodynamic Cycle Studies' pretty heavy going, and neither the sketch (Fig. 3) nor the accompanying text explain what the crucially important 'ceramic rotary regenerator' is. Although this was disappointing, it was taken to be an efficient heat exchanger. Outside this section, however, the LPR (low-pressure-ratio) idea begins to look feasible and attractive. Once I had grasped the principle that the CBEX engine uses the heat in its exhaust stream to increase the energy content of the gas before it enters the expander section, allowing a lower pressure-ratio compressor to be used for the same final power-output, I read on with mounting enthusiasm!

The proposed LPR engine, with a cheap, strong compressor equipped with reinforced-plastics blading, seems to embody the rugged principles and lower costs that will carry the gas turbine into everyday service, beginning with these marine engines and proceeding to the locomotive and (final accolade) the automotive engine.

Sulphidation attack is taken to refer to the 'nibbling' of turbine blades which currently precludes fired washing, making it necessary to stop the gas turbine and allow it to cool before washing the salt off the compressor blading. Users find this a serious nuisance and a major drawback of the gas turbine, so a safe return to fired washing, accomplished whilst underway, would be valuable and important.

When the authors consider the inevitable battle of diesel versus LPR, they should recall that ancient contest in which a (well developed) set of paddle-wheels were towed remorselessly astern by the (novel) screw propeller and take heart!

Prof. Dr Ing. GROSSMAN (Technical University of Berlin): I congratulate the authors of this interesting paper, which shows propulsion units with a thermal efficiency of 0.58 for temperatures which we can manage with today's technology.

If we consider a mechanical efficiency of 0.98 and a gear efficiency of 0.97, we would have an overall efficiency of 0.55, which is greater than the medium-speed diesel engine. These systems therefore look very attractive.

In Fig. 21 the comparison diesel seems to be rather old. Today a 1MW diesel has a fuel consumption of 0.210 kg/kWh with diesel oil, which means an overall efficiency of 0.40. (With LHV = 42 700 kJ/kg, this includes the mechanical losses of the engine.)

For normal low-powered merchant vessels the power range should go down to 10% meaning 46% of design speed and rev/min with a fixed-pitch propeller. Is this possible with the low-pressure-ratio gas turbine?

I would like to put the following questions:

1. When use as a generator turbine on bigger merchant vessels is taken into consideration (which I see as the best possibilities for the LPR gas turbine) what is the influence of the inlet temperature (up to 40 °C) on the thermal efficiency and on the exhaust gas temperature?

2. How high would the exhaust gas temperature be with T = 1473 K?

3. What kind of gear is assumed to reduce 16 700 rev/min down to 1200 or 1800 rev/min?

4. How long can this gas turbine run without overhaul or inspection (the service time of a diesel generator is about 8500 hours per year)?

Dr W. J. SEALE and **Prof. R. WHALLEY** (University of Bradford): This study highlights the fact that the combination of a low-pressure-ratio and a regenerative heat exchanger has the potential to yield high-efficiency gas turbine power plant. This fact is not widely known since it is overshadowed by the use of high-pressure-ratio turbines derived from aviation practice.

To what extent the conclusions rely upon the use of a regenerator having an extremely high effectiveness (0.95) which can be capable of being maintained throughout its working life is open to conjecture. The value of the paper would be considerably enhanced by some indication of the sensitivity of the results to regenerator effectiveness and duty cycle.

One of the prime factors in comparing the use of gas turbines with medium-speed diesel engines for the applications considered by the authors is the ability of the diesel to burn low-cost residual fuel oil, which is almost half the price of high-grade turbine fuel. Thus the gas turbine must have an efficiency of about 60% if its fuel costs are to be comparable to those for a diesel of efficiency 40%. However, in naval applications where common fuel policy is often adopted, this ratio would be reduced.

Nevertheless, there is obviously much potential for development and further improvement in the proposed cycle. It needs to be taken up and pursued vigorously by industry.

Prof. E. MARKLAND (University College, Cardiff): The authors make a most convincing case for the use of regenerators to improve the efficiency of the gas turbine as a prime mover for ship propulsion. The combination offers many advantages; in particular, good fuel economy over a wide range of duties and the application of low-pressure-ratio gas turbines, with the promise of extreme reliability.

It would be interesting to know whether this prospect is matched by the regenerators. Are regenerators presently available of sufficient dependability, with the necessary thermal effectiveness, and sufficiently compact for marine application?

Or is this a field which calls for a substantial development programme before the attractive concepts advanced in the paper may be realized?

Dr I. K. SMITH (The City University): Having left the gas turbine industry over 20 years ago, it is interesting to learn from this paper how much the design art has progressed in this time. The Joule or Brayton cycle with regeneration is of course capable of high ideal cycle efficiencies and these are best realized at low pressure ratios because only with low compressor delivery temperatures is it possible to recover a significant portion of the heat rejected in the turbine exhaust. Like all air cycles it has a rather low work ratio and hence is very sensitive to component efficiencies. Thus in the 1930s improvements in compressor and turbine design together with improved materials made the aircraft jet engine a practical proposition when efforts of the previous one hundred and forty years to produce an effective gas turbine had been in vain.

Thirty years ago we carried out similar cycle analyses assuming turbomachinery efficiencies of around 85%, regenerator efficiencies of about 75% and top temperatures of 1000 °C, and the results were not very favourable in comparison with other prime movers. It is very timely to make reviews such as this in the light of all round improvements in component design and attainable operating temperatures and by including all significant factors in a computer program trends can readily be detected.

The authors are to be congratulated on their clear presentation of what should be the way ahead. However, it must be borne in mind that the gas turbine only prospered because there existed no alternative power plant for high-speed flight and this need was stimulated by a World War. I fear that it may take some time before engine manufacturers will undertake the costly development of a radically new design when the market is uncertain.

On first inspection I find the authors' proposed cycle for power generation to be far more attractive for possible land-based applications in industry, commercial buildings and hospitals than many of the various gas turbine steam combinations and other hybrid power systems with novel bottoming cycles which have been suggested or actually built in recent years. These are especially relevant to the needs of the USA where air conditioning rather than heating is the major building requirement. Perhaps such land-based applications could be the proving ground for marine engines which have to operate in a more hostile environment.

S. S. WILSON: The authors state that 'the maximum possible thermal efficiency of a heat engine is the thermodynamic Carnot limit (the Carnot coefficient), which is equal to $1 - 1/T'$, where T' is the ratio of the maximum to the minimum cycle temperatures. In gas-turbine engines T' is the ratio of the (absolute) turbine-inlet temperature to the (absolute) compressor-inlet temperature'.

These statements imply a number of misconceptions. As I have pointed out,[1] Carnot never defined thermal efficiency, since he did not recognize the energy equivalence of heat and work. He also had no knowledge of absolute temperature, defined much later by Kelvin. He did, however, define an ideal cycle for which later scientists defined the expression for thermal efficiency which has become known as the Carnot efficiency.

However, Carnot's cycle was quite clearly defined as based on the idealized concept of a constant-temperature (isothermal) source of heat and an isothermal heat sink, conditions almost never occurring in the real world since they imply infinite sources and sinks. Consequently, the vast majority of real heat engines have to use a finite source of heat, though most heat sinks are virtually infinite, especially at sea.

For an internal-combustion engine or gas turbine the heat is added over a range of temperature in the combustion chamber, while heat is rejected at a temperature well above ambient. Hence the Carnot cycle and its efficiency are not good ideals for evaluating the performance of gas turbine or an internal-combustion engine. I have proposed a more realistic cycle, the trilateral cycle,[2,3] and its variation, the quadlateral cycle, both of which are based on a variable-temperature heat source, but isothermal heat rejection.

The latter is closely approximated by condensation of a vapour at constant pressure, as in a steam turbine reciprocating engine installation. This accounts for their relatively good efficiency in spite of their low maximum temperature. Modern high-performance steam reciprocating engines (eg Skinner Unaflow) may indeed be competitive for marine work in view of the relative costs of coal and oil.

However, to obtain the highest thermal efficiency (by no means the only important consideration) the best way to approximate to a trilateral or quadlateral cycle by combining a high average temperature of heat reception with a low and constant heat rejection process is a combined cycle, consisting of a primary gas turbine (or internal-combustion cycle) rejecting its heat to a vapour cycle using condensation at low ambient temperature. Such gas-turbine/vapour-turbine cycles have been proposed for marine use, and some prototypes have been built. They should be considered as alternatives to the present proposals.

Briefly, the advantages are the use of orthodox aircraft-derived gas turbines without the use of heat exchangers (a major redesign problem), replaced by a vapour boiler and good part-load performance, due to the high overall work ratio (ratio of nett work done in the cycle to the expansion work) stemming from the inherently high work ratio of a vapour cycle. There also exists the possibility of a 'get-you-home' operation by running either the gas turbine or the vapour turbine separately in an emergency.

The main disadvantage compared with the present proposals is the complexity of a combined cycle against a simple gas-turbine system. However, it may be feasible to use a sealed vapour-cycle system, using a fluid other than water, in which case reliability and ease of maintenance should be high.

It is perhaps unlikely that a combined cycle would compete for small vessels, in view of the need for simplicity.

With regard to low-speed performance of the LPR engine described in the paper (Fig. 19), if the gas turbine is coupled to the propeller via a fixed-ratio gearbox then the power/speed cube law characteristic is such that the operating line would meet the compressor surge line at about 75% of maximum speed. Presumably, this would mean that some form of blow-off would be needed at lower speed, with a corresponding reduction in efficiency. Scaling off, approximately, from Fig. 19 the performance shown in Table DI may be expected.

Although many ships maintain cruising speed for a large part of their life, they all have to spend time manoeuvring at low speeds. Hence, there may be problems of low-speed operation, including stability of combustion. No doubt the authors have considered this aspect, but it would be of interest to know their views.

1. S. S. Wilson, 'Sadi Carnot'. *Scientific American*, pp. 134–145 (Aug. 1981).
2. S. S. Wilson, and M. S. Radwan, 'Appropriate thermodynamics for heat engine analysis and design'. *Int J. Mech. Engg. Education*, Vol. 5, No. 1, pp. 68–80 (1977).
3. S. S. Wilson, 'Discussion on selecting a working fluid for a Rankine-cycle engine'. *Applied Energy*, Vol. 24, pp. 65–68 (1986).

Captain R. F. JAMES (Ministry of Defence): The potential advantages of the Joule/Brayton cycle have been recognized for some time. However, as the authors admit, the low-pressure-ratio (LPR) engine has not yet been built and would require development to demonstrate its practicality and reliability.

In a commercial world the viability of an LPR engine would be determined by the overall cost effectiveness of the development programme and cost of ownership of the resulting

Table DI: Expected performance from Fig. 19

Speed (%)	*Power* (%)	*Efficiency* (%)
100	100	55
90	72.9	53.5
80	51.2	51.8
70	34.3	49.5
60	21.6	42
50	12.5	35
40	6.4	30
30	2.7	?
20	0.8	?

developed engine. Do the authors see the required cost-effective balance being met in the foreseeable future?

The authors note that marine gas turbines derived from highly rated aircraft units have achieved low installed maintenance requirements but at the same time note that a trial complex-cycle engine, the RM 60, was not pursued. Do they believe that the LPR, which uses a more complex cycle than that used in present generation aero-derivative engines, can achieve the same maintenance requirement levels?

Authors' reply

We should first like to thank the contributors to the discussion for their many useful comments and suggestions and for their friendly encouragement.

In reply to Dr Burnside, predictions of thermal efficiency should not be sensitive to the type of liquid fuel. We used a lower heating value of 42.6 MJ/kg in calculating the specific fuel consumption from the thermal efficiency. Combustion efficiency in gas-turbine engines has to be well over 99.5% if carbon build-up and unacceptable emissions are to be avoided, and pumping power is small for any liquid fuel. Gas turbines cannot presently burn as low a grade of residual oil as low-speed diesels; we do not know how this situation might change were ceramic hot parts to be used.

With regard to the possibility of using an organic Rankine or other bottoming cycle on a diesel or a gas-turbine engine, the latter should be even better suited to the match than a diesel, because virtually all of the waste heat is in the exhaust rather than being split between exhaust and cooling water. The economic viability of this and many high-efficiency power systems has been endangered by the recent reduction in fuel prices. Presumably this reduction is an excursion on a long-term upward trend.

Mr Freeman makes the point that for the higher-power levels of naval vessels, the intercooled recuperated gas-turbine cycle, pioneered by Rolls-Royce in the RM 60 engine, is the prime candidate for future engines. He wonders if radial-flow turbomachinery might not be more appropriate for small engines such as the 1 MW unit we studied. Our work leads us to think that radial-flow compressors and turbines would not be optimum. Radial compressor losses, perhaps a more appropriate measure here than efficiency, are about double those for axial compressors in their overlapping size range, which includes the 1 MW engine compressor. The lower losses of the axial compressor increase not only the engine thermal efficiency but the specific power.

The low pressure ratio that can be used at the design point considerably improves the part-load performance of the engine because of the reduced off-design compressor losses. We believe that the improved part-load efficiency more than compensates for a relatively small reduction in specific power, particularly in small engines. The radial turbine does not have losses that are greater than its axial counterpart by as large an increment as is the case for the compressor, but we are predjudiced against the use of radial turbines in gas-turbine engines because of the frequently encountered erosion damage. Small particles cannot pass inward through a radial turbine. In addition, it is difficult to configure multi-stage radial-inflow turbines and, moreover, the outlet flow does not lend itself to efficient recovery of dynamic pressure. These additional 'induced' losses and the erosion danger make the radial-inflow turbine unacceptable, in our view, to high-efficiency long-life applications.

We recognize that a low design-point cycle pressure ratio will give a high temperature at the turbine outlet, ruling out the possibility of metal heat exchangers, but the recent progress in the properties of ceramics indicates that the required duty of a ceramic regenerator should be relatively mild.

An effectiveness of 0.975 for the regenerator does seem high. However, so did 0.75 only a few years ago. Several engines have been built with heat exchangers in the 0.90–0.95 range of effectiveness. Although a heat exchanger with an effectiveness of 0.975 will be approximately double the volume of one of 0.95 effectiveness using the same matrix core, the volume can be reduced by using a smaller hydraulic diameter. (The volume of a heat-exchanger core is proportional to the square of the hydraulic diameter.) Periodic-flow regenerators are less susceptible to fouling than are steady-flow recuperators because of the flow reversal every revolution, allowing the use of very small hydraulic diameters. In the particular application we studied, the volume of the LPR gas-turbine engine remained considerably smaller than the volume of the equivalent diesel. The weight was so much less that considerable freedom would be given in engine placement in the hull. Air and exhaust ducts would be larger, but would not affect any aspects of the layout of the ship we studied.

We agree with Mr Watson that the proposed engine should be considered at present only for power levels of about 1 MW and below. The principal reason is that ceramic heat exchangers are not yet available in large sizes, and using many small units in a large plant does not seem appropriate. Also, the intercooled heat-exchanger cycle is inappropriate for low-power engines because the high-pressure compressor blading becomes too small for either high efficiency or long life.

We also emphatically agree that the rotary ceramic heat exchanger should receive major development funding. On one point we disagree: gyroscopic action is of no concern for this type of heat exchanger because the rotational speed is extremely low, for instance 3 rev/min.

Prof. Winterbone gives some useful history of the heat-exchanger gas-turbine engine. We should like to add the following. The early rotary regenerators used stainless-steel cores, wrapped from a pair of metal strips, one plain and one wavy. Much of the development into sealing methods and other details of design was carried out by the National Gas Turbine Establishment. Stainless steel has a high expansion coefficient, and the disks, having one face at a high temperature and the other near compressor-outlet temperature, would tend to warp. The development by Corning Glass of ceramic honeycomb material having not only a high temperature capability but a very low thermal expansion coefficient was greeted with enthusiasm.

Noel Penny, then head of Rover Gas Turbines, fitted one of the first into the Rover-BRM car entered in the Le Mans 24 hour race. At some point during the race, a bolt came loose and embedded itself into the ceramic matrix. This damaged the core and the seal, increasing the gas leakage and reducing output power. Nevertheless it was decided to continue with the race.

The robustness of the ceramic heat exchanger in this severe test encouraged Ford USA to specify ceramic heat exchangers for its truck (lorry) gas turbine, scheduled for major production. Unfortunately, all testing was carried out on low-sulphur fuels, and the ceramic turned out to be susceptible to sulphur attack serious enough to cause Ford to shut down the entire production. This one episode has shaken faith in the ceramic heat exchanger, despite the rapid development by Corning of a material that is sulphur-resistant.

A two-shaft engine would have performance lines different from those of the single-shaft engine. It is impossible to use the single-shaft-engine performance to evaluate or deduce the two-shaft-engine performance, since two expander maps would be used in the component-matching procedure of the two-shaft engine. However, we do agree that a full investigation is warranted. As for the cycle-performance plots, the thermodynamics of the cycle shown in Fig. 9 are not affected by single- or twin-shaft-engine considerations.

Prof. Grossman comments on the thermal efficiency of the diesel. At about the time this study was initiated (1983) some European manufacturers of medium-speed diesels introduced engines with higher thermal efficiencies and with the ability to burn heavier fuels. At the same time the US industry was dominated by two medium-speed diesel engines in the power range of 1 MW. The performance shown in Fig. 21 was derived using the performance map of the US-built engine that had the higher thermal efficiency. In fact, this engine was the one that powered the 'hypothetical vessel' of Ref. 2.

The performance maps of all engines include regions in which the engine cannot operate. For diesel engines such limits may include: a manufacturer-suggested minimum power level for each speed (near full speed the minimum power level is high enough to present some problems in light-ship conditions); the maximum-torque limit; a turbocharger-matching limit; a bearing-load limit; and others (see *Marine Engineering*, ed Harrington, SNAME, (1971)). For a fixed-pitch propeller and a cubic speed–power law the relationship between speed and power is shown in Table DII.

Table DII: Relationship between speed and power

% Power	0	10	20	30	40	50	60	70	80	90	100
% Speed	0	46	58	67	74	79	84	89	93	97	100

Using these figures, the LPR engine approaches the compressor surge line between 40 and 60% power, as shown in Fig. 21a. Note that the lower-pressure-ratio engine LPR1 can operate at lower speed and power levels than the others (LPR2, LPR3 and LPR4), which is an additional advantage of lower pressure ratios. The corresponding limit for the diesel engine with a fixed-pitch propeller is 20% power.

With controllable-pitch propellers it was specified that the CPP would receive the required power at the respective engine's optimum speed for maximum thermal efficiency at this power. With this assumption engines LPR1 and LPR2 can operate down to 10% power, while LPR3, LPR4 and the diesel can operate down to 20% power, as shown in Fig. 21b.

With constant-speed drive (100% speed) all LPR engines operate down to 10 or 20% power. Diesel-engine manufacturers do not publish the low-power, high-speed performance of their engines because at these levels carbon and sludge buildup are unacceptable for reasonable life. Thus with constant-speed drives the diesel does not operate below 60% power.

The design-point thermal efficiency for T_{01} = 288 K is 55%. By increasing T_{01} to 313 K (40 °C) and keeping T_{04} at 1555 K the design-point thermal efficiency is reduced by about 2.7%. For T_{04} = 1473 K the temperature at engine (heat-exchanger) exhaust is 440 K.

The propeller of the 'hypothetical vessel' in Ref. 2 was designed for 290 rev/min at design-point operation. The required reduction ratio of 58:1 is too high to obtain in one gearbox. This would require either two epicyclic gearboxes; or one epicyclic gearbox (near the turbine) and one single-input single-output gearbox (near the propeller). The choice of gearing is directly linked with the choice of the location of the engine in the engine room. (Various solutions to this problem were investigated by Hewon Hwang in her BSME thesis, MIT, September 1984.)

With regard to maintenance requirements, many gas-turbine engines used for pipeline service are inspected every two years and designed for a 100 000 h life. Current aircraft and aero-derived gas-turbine engines are overhauled every 10–15 000 h.

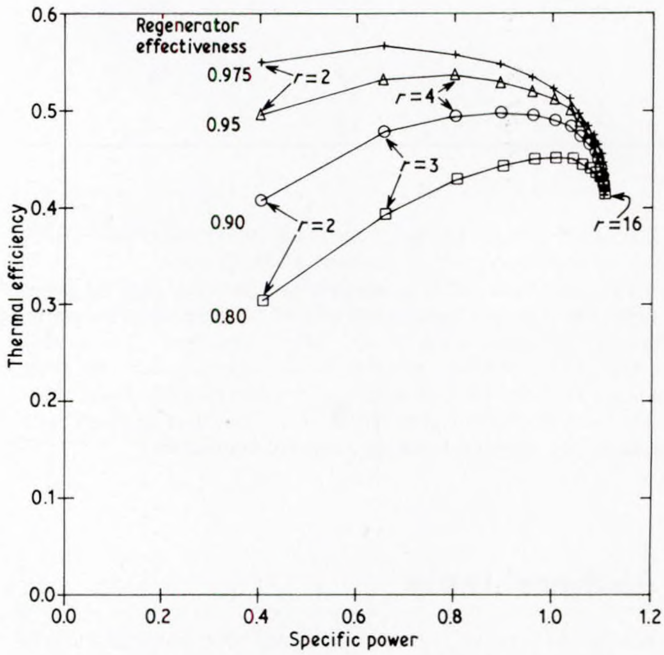

FIG. D1: Plot of thermal efficiency against specific power as a function of regenerator effectiveness for T_{04} = 1555 K

Dr Seale, Prof. Whalley and Prof. Markland all question the dependence on the high-effectiveness heat exchanger. The influence of heat-exchanger effectiveness (for T_{04} = 1555 K) can be seen from Fig. D1, which was produced with high cooling-mass-flow fraction and other parameters as shown on page 4. The optimum pressure ratio for maximum thermal efficiency is increasing with decreasing regenerator effectiveness. As a philosophical comment, it seems to us illogical to strive for minimum losses in the compressor, combustor, turbine and ducting and yet to choose losses of, for instance, 20% in the heat exchanger. It is more in accord with the general approach to design to choose to reduce losses as far as is economically feasible. This is possibly where the discrepancy in treatment of turbomachinery and heat exchangers originates as it is theoretically possible to achieve simultaneously an effectiveness close to 100% and pressure losses near zero in a heat exchanger. It would, of course, become uneconomically large, but it marks this component as different from compressors and turbines, which seem to show a fundamental limit of around 5–6% on losses.

It seems strange to us that the rotary regenerators have received comparatively little government funding in the last two decades. In the small sizes used in the AGT-100 and AGT-101 ceramic automobile engines they suffer leakage of about 10% of the compressor flow, an unacceptably large amount. In an earlier US government programme at Allison Gas Turbines (Ceramic Applications in Turbine Engines), the ceramic regenerators on a 300 kW engine had leakage losses of about 4% and effectivenesses of over 90% in some conditions of operation. The twin rotary regenerators used were still small, about 600 mm diameter. The largest used on gas-turbine engines so far to our knowledge, those for a Ford truck engine, were about 1 m in diameter.

Accordingly, some development in manufacturing capability is required for these units to be used in higher-power engines, but we expect the leakage rate to be reduced partly from the square-cube law and partly from further developments in seal technology. The improvement in ceramic properties reported, for instance, in the annual 'Contractors' Coordination Meeting' organized by the US Department of Energy has been remarkable, and few of these gains have been applied to regenerators. Hence we believe that a concerted programme could yield impressive advances.

The impact of the use of ceramics in gas-turbine engines on the potential for burning residual fuel oil is, as far as we know, unexplored. It would be delightfully serendipitous if ceramics did bring about the capability for the use of low-cost fuels.

Dr I. K. Smith's comments are all accurate. Although the low-pressure-ratio cycle is being proposed for several widely different applications in addition to marine duty (for instance, the high-temperature gas-cooled nuclear-reactor power system) it would be ideal if it were first applied to a land-based power-generation or total-energy plant.

Mr Wilson has advocated more rigour in the use of Carnot's name when attached to cycle efficiency than is customary among today's perhaps careless thermodynamicists. There will always be differences as to the most appropriate ideal process or cycle with which to compare an actual process or cycle, and he prefers a more-realistic comparison rather than an ideal. However, we believe that the so-called Carnot cycle and efficiency are useful measures of performance for gas-turbine cycles, and particularly low-pressure-ratio heat-exchanger cycles, because as the pressure ratio is reduced and the heat-exchanger effectiveness is increased, the heat-addition and -rejection processes occur over a closer and closer approach to isothermal conditions. In the limit there could be a succession of intercooled compressor stages each with a pressure ratio close to unity, and a succession of turbine stages with working-fluid reheat between stages, and with a perfect heat exchanger the most appropriate ideal (reversible) cycle would then be Ericsson's rather than Carnot's.

We agree with Mr Wilson's comments about the position of the combined cycle for small vessels. With regard to the low-speed performance of the LPR gas turbine, we had contemplated the use of variable-pitch propellers to avoid the working line running into compressor surge at part load. We have commented on this aspect earlier. Diesels are so thrifty at idle and at very low power levels that we would not claim turbine superiority under that condition. Based on the wide range of operation of aircraft combustors we see no stability problems in the combustion system of a sea-level engine.

In reply to Captain James, we believe that the LPR engine should be cost-effective as regards development and operation for the following reasons:

1. Much of the cost of development of a new aircraft engine is that of the high-pressure-ratio compressor. The LPR has a compressor of extremely low pressure ratio, which is easy to design and should need virtually no development.

2. The rotary regenerator has been the topic of no major development programme since the pioneering work of the National Gas Turbine Establishment in the 40s and 50s. The ceramic version performed outstandingly in the Rover-BRM car in the Le Mans race over twenty years ago, and has had some more development in recent Department of Energy programmes in the US. Once developed, the only difference between regenerators for different engines is core size. Again, no design problems are foreseen and no development effort should be required once improved seals (such as those developed by the NGTE) are married to ceramic cores made from one of the new ceramics. One or more manufacturer has to tool up to extrude cores of a size suitable for larger engines, eg 2 MW, in contrast to the present limit of about 350 kW.

3. The LPR engine should require less maintenance and be more reliable because it uses lower blade speeds, and hence will be subject to lower erosion and foreign-object damage. In addition, it has none of the very short, fragile, high-pressure blades nor the long, low-hub-shroud-ratio low-pressure blades of high-pressure-ratio compressors. Neither will it have multiple spools with complex bearing and shafting arrangements, nor actuating gear for changing the setting angle of several rows of stator blades. Turbine blades are not as susceptible to blade-length problems as are compressor blades, but there are advantages in not having to cover as wide a range in length, or hub-shroud ratio, as is necessary in a high-pressure-ratio engine.

Published for THE INSTITUTE OF MARINE ENGINEERS by Marine Management (Holdings) Ltd
(England Reg. No. 1100685) both of 76 Mark Lane London EC3R 7JN.
Photoset by Paston Press, Loddon, Norfolk. Printed in England