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High-Efficiency Brayton-Cycle Engines for Marine Propulsion

MIT Sea Grant College Program
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ABSTRACT

Three alternative gas-turbine cycles were examined for their potential to provide greatly improved marine propulsion (with particular reference to fishing boats): the direct-plus-inverted (DIC) cycle; the regenerated (CBEX) cycle; and the intercooled regenerated cycle (CICBEX). The DIC cycle can be considered to be a simple unregenerated (no-heat-exchanger or CBE) cycle with an expander-heat-exchanger-compressor (EXC) group added as a gas-turbine "bottoming" cycle.

While the DIC cycle showed the potential for considerable improvements in efficiency and specific power (ie, power output per unit engine airflow) for converting simple CBE engines, it was unattractive for fishing-vessel propulsion. First, there are no fishing vessels with CBE-cycle engines at present available for conversion; and, second, one of the other cycles examined showed potentially far higher performance when considered as a new engine.

This highly attractive cycle is the highly regenerated low-pressure-ratio CBEX cycle, the advantages of which are discussed more fully below. A variation of this cycle is the intercooled regenerative, or CICBEX, cycle. The net output of a gas-turbine engine is the difference between the turbine power produced and the compressor power absorbed, so that a reduction in compressor power can have a large effect on overall engine efficiency and specific power. It seemed initially that an intercooled cycle would prove highly attractive for marine vessels because the sea could provide an always-available low-temperature heat sink, and because intercooling a high-pressure-ratio compressor can reduce its power consumption markedly. However, we eventually rejected the CICBEX cycle for two reasons. One was that the optimum pressure ratio of the highly regenerated CBEX cycle turned out to be so low that intercooling did not produce a significant reduction in compressor power. The second reason is that the use of two low-pressure-ratio compressors in an intercooled cycle produced an overall pressure ratio too large for ceramic rotary regenerators to be employed. At this pressure ratio leakage becomes penalizing in this type of heat exchanger, while cost, size and other limitations in the available alternative types of heat exchanger greatly reduce the attractiveness of the CICBEX cycle.

The cycle that maintained its attractiveness throughout our scrutiny is, then, the CBEX cycle in its highly regenerated, low-pressure-ratio (LPR) form. It shows a strong potential for the development of an engine that would be more efficient than the advanced diesel engine both at full and part power. It would also cost less initially, have a lower mass and volume, and require less maintenance. Although this engine could be produced with conventional technology and materials, to realize its fullest potential it depends, as does the advanced diesel engine, on the development of improved ceramics. There are at present several research and development programs, funded

privately and by the U.S. and other governments, that are showing very promising results for the application of ceramics to both types of engines.

Ceramics, or certain other nonmetals, would enable high turbine-inlet temperatures to be used in small gas-turbine engines. Large gas-turbine engines, such as aircraft jet engines, employ air-cooled ceramic-coated metal turbine blades to allow high temperatures to be used. Small engines (below 1 MW) cannot use air-cooled blades economically and reliably. Yet high inlet temperatures must be employed if the efficiency of small gas-turbine engines is to be improved. The use of ceramics would bring other advantages: the material cost would be far lower; designers could use more turbine stages with lighter loading to give increased turbine efficiency: we would not be dependent on strategically scarce materials like cobalt and chromium; and wasteful use of compressed air for turbine-blade cooling would no longer be needed.

The engines we are proposing appear to have some advantages over current experimental gas-turbine engines using ceramics. In a "clean-sheet design" of an "optimum" engine (defined in a following chapter) operating on a low-pressure-ratio CBEX cycle we show that the use of highly effective ceramic heat exchangers enables the pressure ratio to be reduced from the frequently used range of 5-15 to about 3. The result is an engine in which stresses and speeds can be so reduced as to allow the compressor to be made from a commercial reinforced plastic, while giving outstanding efficiency and range of operation. The turbine-blade stresses would also be greatly reduced, greatly lowering stady-state blade stresses and the effects of foreign-object impacts (from grit, carbon particles etc.). The resulting engine is predicted to give 10 to 30 percent improvement in fuel consumption over the advanced diesel engine at full and part power, while retaining its advantages of small size, high reliability, and potentially lower cost.

We believe that, if the present programs of development of ceramic materials for gas-turbine engines are successful, the design and development of low-pressure-ratio low-blade-speed engines should be far lower in risk and cost than those for the more normal higher-pressure-ratio high-blade-speed engines. Nevertheless, we investigated a possible short-cut: the conversion of "retired" helicopter and other engines to the LPR CBEX cycle by the elimination of high-pressure compressor and turbine stages and the addition of a rotary regenerator. This study is an extremely complex undertaking, and our findings cannot be regarded as final. It appears, however, that at least some lightweight highly rated engines of the type that are removed from service at a time well before the end of the useful life of the main components could be converted to a form giving high efficiency at reasonable first cost.

FOREWORD

The initial impetus for this program came from the congruence of two developments. The first was the series of sharp oil-price increases in the 1970s, resulting in the fuel-cost component of the price of landed fish rising from an insignificant level to a point where fishing vessels were being limited in speed and range in order to reduce fuel usage. The second development came from the principal investigator's involvement in two programs: one as a member of a consulting panel and symposium-session chair in a Department-of-Defense study of advanced military engines; the other as consultant to a company working on solar-thermal power systems. He (Wilson) became enthusiastic about three variations of the Brayton-cycle (gas-turbine) engine that seemed to promise very high thermal efficiencies. Two of these variations also offered a favorable environment for future ceramic turbine construction because of the very low blade speeds that were possible.

A proposal for the study of these alternatives was written in late 1980 and was funded by Sea Grant in 1981.

CHAPTER 1

INTRODUCTION

BACKGROUND

Although the events that set off the "energy crisis" of a decade ago produced an unprecedented increase in the price of petroleum fuels, the inflation induced in most countries has since reduced the relative effect considerably. Nevertheless, there is still sufficient differential to cause fuel costs to be a very significant component of the total cost of commercial marine propulsion. The long-term, and perhaps the short-term, trends are for further increases in the costs of fuel relative to other costs. The current study was undertaken to examine whether or not the principal engine used by fishermen today, the diesel engine, could be surpassed, at least in fuel efficiency, by turbine engines.

The diesel engine is already the most efficient of prime movers, exceeding in certain cases the efficiency of the largest steam-turbine plant. Moreover, the peak thermal efficiency of the most advanced diesel engines is being continually increased, reaching over 50 percent in some large marine units. In contrast, the efficiency of at least land-based steam-turbine plant is decreasing because of the energy-use effects of legal requirements to reduce sulfur emissions.

The diesel engine has other virtues. Its idling fuel consumption is the lowest of any of the principal engines, and its part-load efficiency is also very good. Its operation is little affected by water, a very favorable attribute in comparison with the spark-ignition engine, and although a salt-laden atmosphere is not beneficial, it is also not crippling. The engine is extremely reliable so long as somewhat demanding maintenance schedules are observed.

The diesel engine would appear to be a paragon, difficult to improve upon. That we should have the temerity to suggest that a better engine for such a specialized duty as that demanded by US fishermen could be produced requires a thorough and convincing explanation. A large measure of the justification for our approach is given in a just-published text (ref. 1).

In brief, the gas-turbine engine has proved itself extremely reliable in duties where it has received steady development, especially in airline service. Major-overhaul periods of 20,000 hours are not uncommon, with only minor maintenance requirements at shorter intervals.

The purchase price of industrial gas turbines has been similar to that of industrial diesel engines, and since the size and weight are lower, the housing and foundation costs are less. Gas turbines can tolerate a wider range of fuels. For instance, two similar gas turbines sold by Ruston and Hornsby when the principal investigator worked there operated on peat (turf) in Eire in one case, and on coal-tar fuel, in the Bank of England, London, in the other. Lubricating-oil needs are extremely small. Gas-turbine engines for industrial duty are generally quieter (or can be made so) and with far less vibration than diesel engines. The exhaust is almost nontoxic, with negligible carbon monoxides or, in the newer engines, nitrogen oxides: it is the only one of many considered as alternatives for future automobile propulsion that has met the most stringent of the EPA limits for nitrogen oxides.

Only in respect of fuel consumption is the present industrial gas turbine deficient (the aircraft gas turbine has long given an improved operating efficiency over the piston engines it replaced). And the specific fuel consumption of gas-turbine engines has been steadily improving as the so-called "firing temperature" has increased under the stimulus, principally, of military-engine requirements. Higher firing, or turbine-inlet, temperature increases the power output of a given size of jet engine; the improved fuel consumption appears mainly in fan and propeller and other shaft-power engines. The higher temperatures are spreading to industrial gas-turbine engines only slowly, lagging hundreds of degrees behind the peaks used in civilian aircraft engines. But these higher temperatures are reached through the use of very complex and somewhat sensitive turbine blades made from exotic alloys.

We are now on the verge of a major revolution that will change this situation dramatically: the substitution of ceramics for the costly alloys and complex manufacture. "Crash" programs are underway in several countries to endeavor to attain leadership in the new engineering materials (ref. 2). Ceramics are incrasingly used in the hot sections of experimental and some production gas-turbine engines.

When development is successfully completed, the way will be open for gas turbines to be designed to be better in fuel efficiency than any other engine, including the diesel engine. Whether the improvement will be sufficiently dramatic for a complete changeover to this new technology to be made in all appropriate industries will depend on other aspects of the design. The current project was proposed to ensure that the particular requirements of fishing-vessel builders, owners and operators be considered at the design stage of new propulsion equipment.

The recent history of the gas-turbine engine is reviewed in the following paragraphs, leading to a summary of the approach recommended here.

THE EMERGENCE OF THE HIGH-EFFICIENCY GAS-TURBINE ENGINE

Although the first gas-turbine engines were designed for industrial purposes, research and development were soon dominated by the particular requirements of military and commercial aircraft engines. All heat engines are endowed with improvements in both thermal efficiency and specific power (a measure of the power-to-weight ratio) if the maximum temperature of the "working fluid" - air in this case - is increased. (The diesel engine achieves its high efficiency principally because the short duration of its combustion processes allows very high gas temperatures to be used. Its specific power is low mainly because the combustion process occupies so short a proportion of each cycle.)

Under the intense stimulus of the rewards bestowed on aircraft turbine engines by higher gas (air) temperatures at turbine inlet, extraordinary developments first in metallurgy and then in methods of cooling turbine blades have led to temperatures only a little below 3000 F (1650 C) to be in current use in advanced gas-turbine engines (figure 1). The thermodynamics of the cycle require that, for the full advantages of the higher temperatures to be obtained, the compressor pressure ratio must also be substantially increased (ref. 1). Modern jet engines have compressor pressure ratios between 20:1 and 40:1.

These high pressure ratios have in turn led to extraordinary engineering-design developments. Compressors of high pressure ratio are extremely temperamental, and it has taken Herculean efforts to provide them with narrow but adequate working ranges of acceptable efficiency by either splitting them into several low-pressure-ratio compressors driven by separate turbines through complex concentric-shaft arrangements, or by equally complex systems whereby about half the stationary blades of an engine compressor are pivoted and moved through precise angles at different points in the operating schedule. A very large proportion of the huge expense necessary to develop new aircraft engines goes to the cost of producing an acceptable high-pressure-ratio compressor.

To a large extent, aircraft-engine developments have dominated much of the commercial gas-turbine field. Many industrial gas-turbine engines are, in fact, simply jet engines in which the exhausts pass through large shaft-power turbines in place of the normal propelling nozzles. The US Navy's principal gas-turbine propulsion engine, the LM 2500, is derived from the GE CF6 jet engine in just this manner. However, low-power gas-turbine engines have had to take a different approach. Their small physical size has made it impossible, for rather abstruse but definite fluid-mechanical reasons, to design compressors of high pressure ratio, and it is also impracticable to produce small turbine blades having tortuous cooling passages.

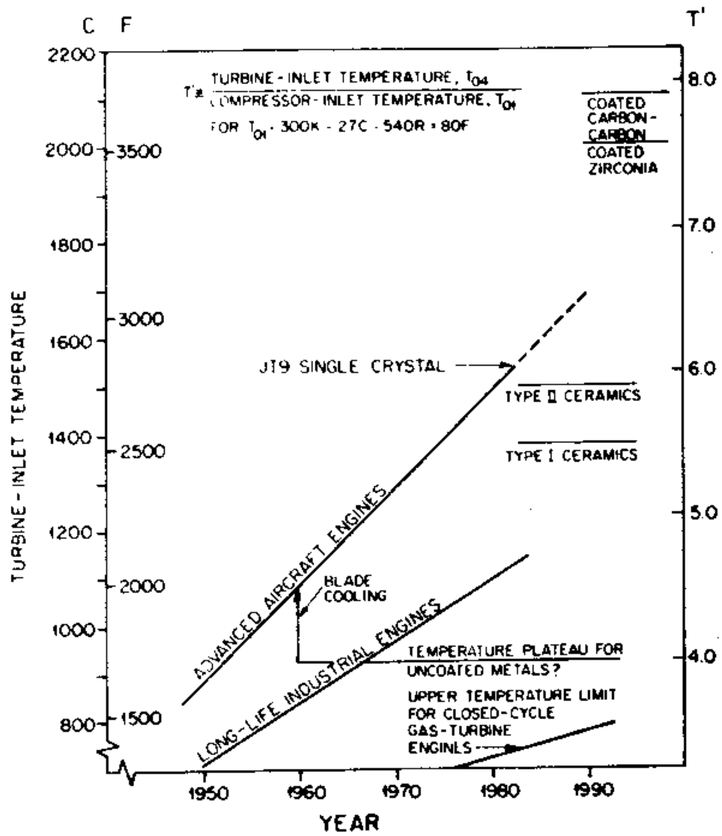


FIGURE 1 INCREASE OF TURBINE-INLET TEMPERATURE WITH TIME

from ref. 1

Designers of small turbine engines for, principally, automotive uses were forced to take a different approach: the heat-exchanger or "regenerative" (CBEX) cycle. In this a low-pressure-ratio compressor passes its flow through one "side" of a heat exchanger, through the other side of which flow the hot gases from the turbine exhaust (figure 2). The fuel flow required in the combustion chamber to produce the design turbine-inlet temperature can then be reduced compared with that required in a high-pressure-ratio compressor, and an acceptable thermal efficiency can be obtained.

We are proposing a rather minor extrapolation of the heat-exchanger cycle that simply makes a virtue of the necessity of using both the heat exchanger and the low pressure ratio. Instead of aiming for a pressure ratio near the maximum easily obtainable in small engines, an approach often used in the past, we are choosing to design for the maximum feasible heat-exchanger effectiveness coupled with a pressure ratio that will give the optimum set of characteristics for the engine duty specified. The benefits of this type of design have

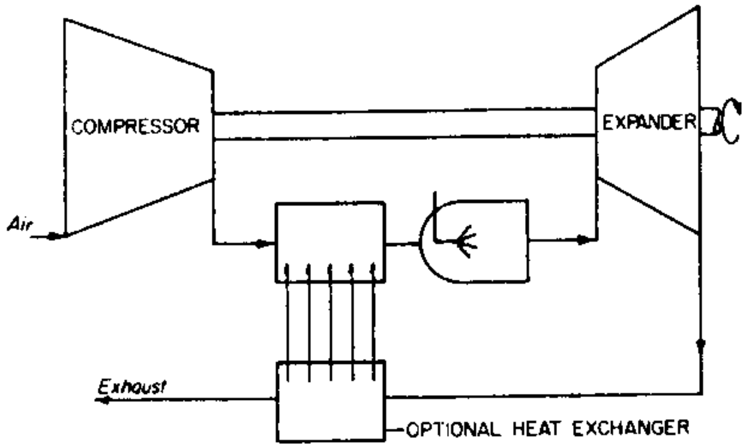


FIGURE 2 BLOCK DIAGRAM
OF GAS-TURBINE ENGINE
WITH HEAT EXCHANGER

from ref. 1

been appreciated before (ref. 3). It has turned out that this approach has serendipitously brought several unsuspected advantages.

THE LOW-PRESSURE-RATIO CYCLE

The specification of the maximum feasible heat-exchanger effectiveness requires engineering and economic judgement. In the first place, the only feasible heat exchanger for a high-temperature low-pressure-ratio cycle is a ceramic rotary regenerator (sometimes called a "heat wheel" in air-conditioning applications) (figure 3). The reason is that the maximum possible inlet temperature for a metallic heat exchanger is currently below 760 C (1400 F). Turbine-inlet temperatures are currently over 930 C (1700 F) for uncooled turbines and up to 1540 C (2800 F) for cooled turbines (figure 1). The temperature drop through a low-pressure-ratio turbine expander may be as low as 250 degrees C (450 degrees F), so that metallic heat exchangers could be used only for low-temperature (uncooled) gas turbines or for high-temperature turbines having a high pressure ratio. The size of engine that we have chosen as typical for fishing vessels in our study is 1.12 MW (1500 hp), large enough to have cooled blades, and thus large enough to be categorized as potentially a high-temperature engine. If we opt, therefore, for a low-pressure-ratio cycle, the use of metallic heat exchangers must be ruled out.

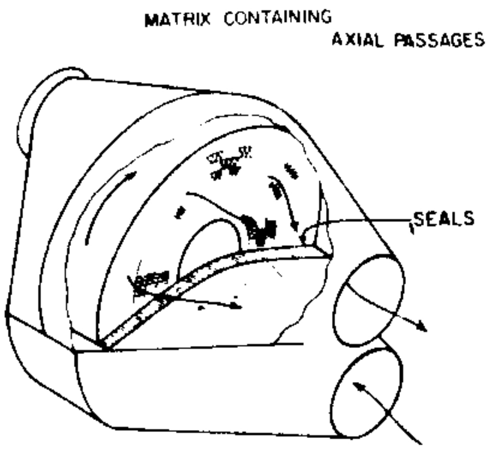


FIGURE 3 DISK-TYPE ROTARY REGENERATOR

from ref. 1

In any case, ceramic heat exchangers have certain distinct advantages for this application, and this is where economic judgement must be used. The size of heat exchangers for a given duty is roughly proportional to the square of the passage (hydraulic) diameter. The passages in a rotary regenerator can be made far smaller than those in a metallic recuperator for two significant reasons. In a rotary regenerator (figure 3) the flow reverses every few seconds through every passage, tending to clear away accumulated particles. It has also become practicable to extrude regenerator cores having passages of extremely small hydraulic diameter without a large increase in cost per unit surface area (possibly, in fact, a decrease in unit cost). These are advantages over metal recuperators in which the flow is unidirectional, and small passages tend to become clogged, and in which the unit cost increases as passages are made small. The overall size of rotary regenerators of given effectiveness can be a small fraction of the size of a metallic recuperator in consequence of this ability to form and to use small passages.

A disadvantage of rotary regenerators over recuperators is the leakage that inevitably flows past the wiping seals around the four ducts that lead the two flows to and from the faces of each disk, and that also occurs through the carrying of the gas trapped in the matrix passages into the opposing stream. The effect of both of these leakages is reduced by reduced pressure ratio. A limiting pressure ratio of 6:1 is generally applied to rotary regenerators (ref. 4); the cycles we are recommending are well below this limit.

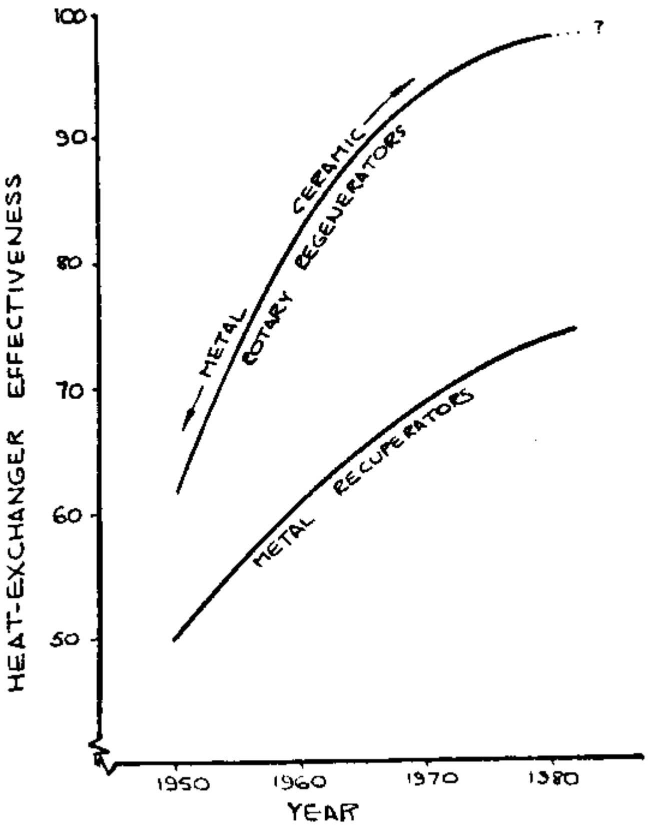


FIGURE 4 GROWTH OF PEAK EFFECTIVENESS USED IN GAS-TURBINE HEAT EXCHANGERS (ESTIMATED)

Therefore a strong case can be made for the specification of ceramic rotary regenerators. Judgement is required in the specification of the effectiveness - a characteristic that can be roughly translated to "heat-transfer efficiency". Maximum effectivenesses for gas-turbine heat exchangers has risen rapidly since the early problems with rotary regenerators have been solved, as shown in figure 4. The highest-effectiveness heat exchanger we know of in a gas turbine is in the Allison GT 404, at just over 0.95. This engine, designed in the early 1970s (ref. 5), uses the common arrangement of twin ceramic disks taking the compressor and turbine flows split evenly between them, disposed on opposite sides of the shaft with their common axis of rotation intersecting the main turbine axis at right angles (figure 5). They are of moderate size. To increase the effectiveness to 0.975 - halving the thermal losses - could be done by doubling the thickness of the ceramic disks, which would keep the size within reasonable bounds and would improve the flow distribution to give further gains. (Another advantage of the rotary regenerator, low pressure drop, can become a disadvantage if the pressure drop is so low that the flow does not distribute itself over the disk surface). We have therefore specified this value - 0.975 - of heat-exchanger effectiveness in the engines we are proposing. It could well be economically justified to propose even higher levels of effectiveness - which would entail even lower optimum pressure ratios for the gas-turbine cycle - but it is our judgement that we should proceed with some caution.

ADVANTAGES OF LOW PRESSURE RATIOS IN GAS-TURBINE ENGINES

The combination of low compressor pressure ratio with high heat-exchanger effectiveness has some particular advantages for the type of high-efficiency engines needed for marine propulsion. The design-point thermal efficiency will be higher. The maximum possible thermal efficiency of a heat engine is set by the thermodynamic Carnot limit, which is $(1 - 1/T')$, where T' is the ratio of the (absolute) turbine-inlet temperature to the compressor-inlet temperature. For marine engines in the late 1980s, T' will be between its present value of about 5 to a future value which should be reached with ceramic turbine blades of at least 6. The Carnot efficiency limit is therefore between 0.80 and 0.83. The closeness with which actual gas turbines approach the thermodynamic limit will depend on the sum of the component losses. Compressor losses (roughly taken as (1 - polytropic efficiency) decrease with decreasing pressure ratio (figure 6), but seem to reach a limiting low value of 0.06. Heat exchangers, on the other hand, seem to have no theoretical lower limit of losses. In practice, the sum of the thermal and pressure-drop losses can be considerably less than 0.06, and far less than compressor losses would be for a high-pressure-ratio no-heat-exchanger cycle.

Heat exchangers also have efficiency advantages over compressors, especially those of high pressure ratio, at part load. Heat exchangers actually improve their performance at part load, while compressors deteriorate markedly (figure 7). We are working to substantiate these estimated variations with analytically derived characteristics, but

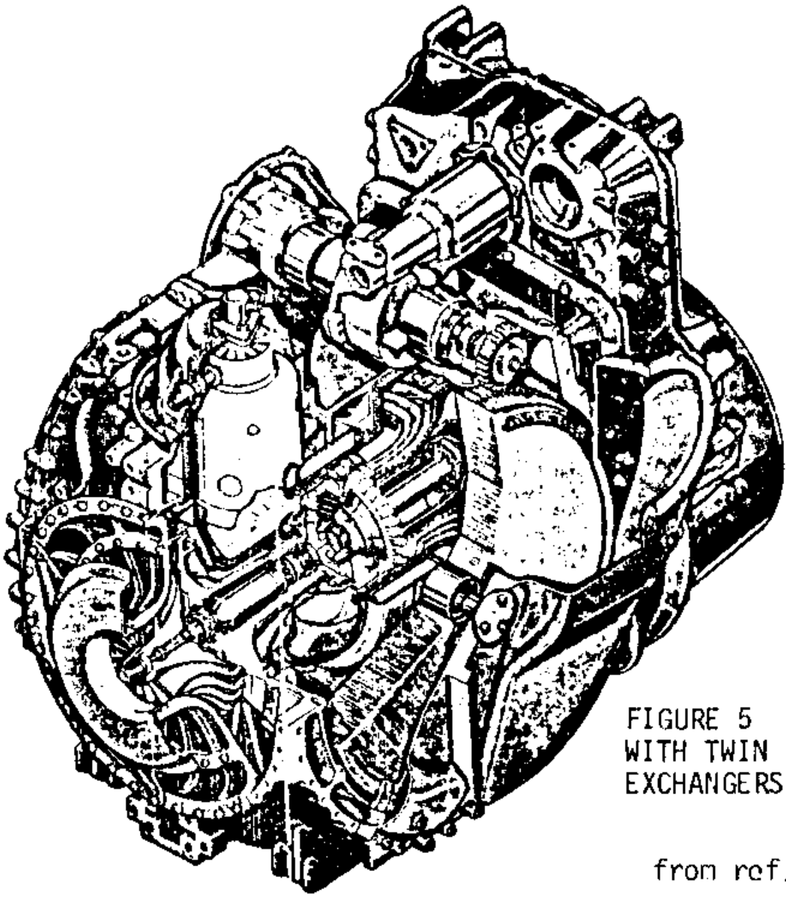


FIGURE 5 GAS-TURBINE ENGINE WITH TWIN REGENERATIVE HEAT EXCHANGERS (DDA GT 404)

from ref. 5

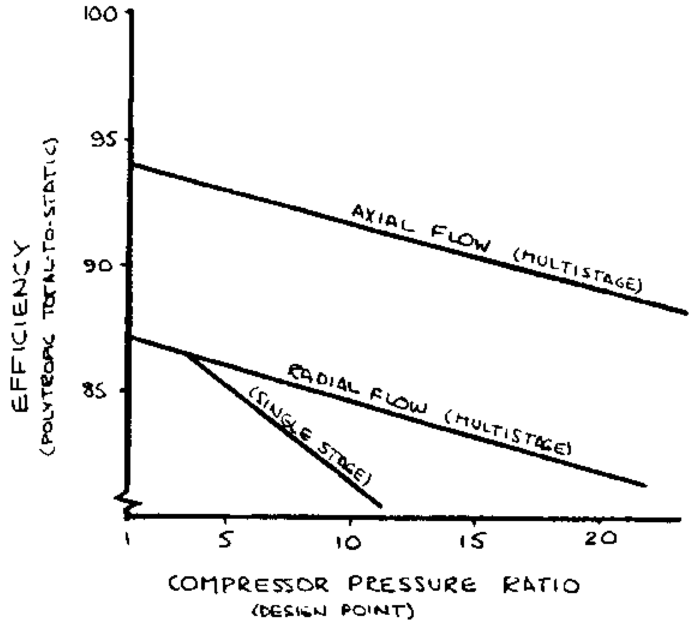


FIGURE 6 PEAK COMPRESSOR EFFICIENCIES (ESTIMATED)

there is no doubt as to the trend. We have shown one calculated curve of part-load efficiency (ref. 6) for a converted, and therefore compromised, helicopter engine, and have estimated a performance curve for a "clean-sheet" engine design in figure 8. We give more recent results in chapter 5.

A major uncertainty with regard to future high-efficiency gas turbines at present is the possibility of using nonmetallic materials. The chief candidates are various forms of ceramics, all being extremely brittle, and tending to fail after a time of successful use in ways as yet imperfectly understood; and graphite fibers in a graphite matrix (called "carbon-carbon") which is a tough composite material actually increasing in strength up to at least 3500 F (2200 K), but requiring the absolute exclusion of oxygen by means of coatings. The ceramic materials seem closer to realization, but have experienced failures when incorporated into engines having very highly loaded turbines with high peripheral speeds and consequent high centrifugal stresses. The LPR cycle we are proposing here uses lightly loaded turbines of low peripheral speeds and low centrifugal stresses, and accordingly might be an ideal candidate for the introduction of ceramic blades, possibly strengthened with graphite fibers.

Being able to use ceramic or other nonmetallic blades in a gas-turbine engine could increase the efficiency at full- and part-load for three reasons.

1. There would be need for only a small amount of cooling air bled from the compressor discharge (figure 9). The compressor work required to provide this cooling air has a damaging effect on engine thermal efficiency, and therefore the smaller the quantity required the better the performance.

2. Turbine blade shapes would not need to be compromised to provide for internal cooling passages, and the discharge of cooling air would be eliminated, yielding lower fluid-mechanic losses in both areas.

3. Because ceramic (or other nonmetallic) blades are expected eventually to be relatively inexpensive to produce, there would no

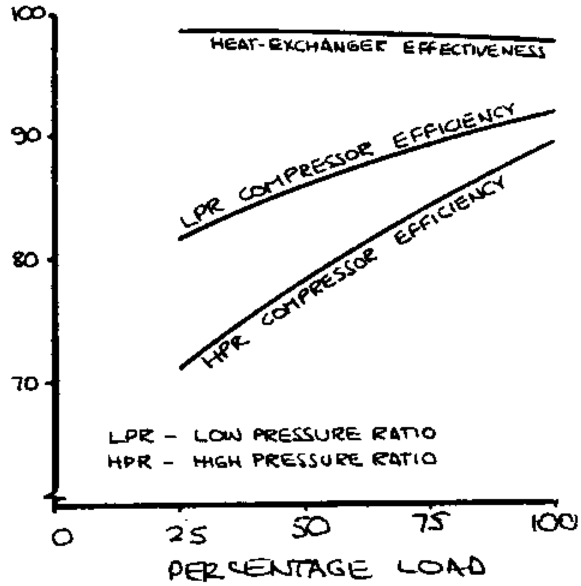


FIGURE 7. VARIATION OF HEAT-EXCHANGER EFFECTIVENESS AND COMPRESSOR EFFICIENCY WITH LOAD (ESTIMATED)

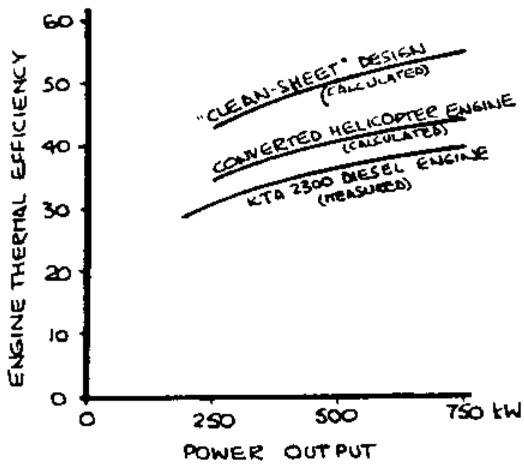
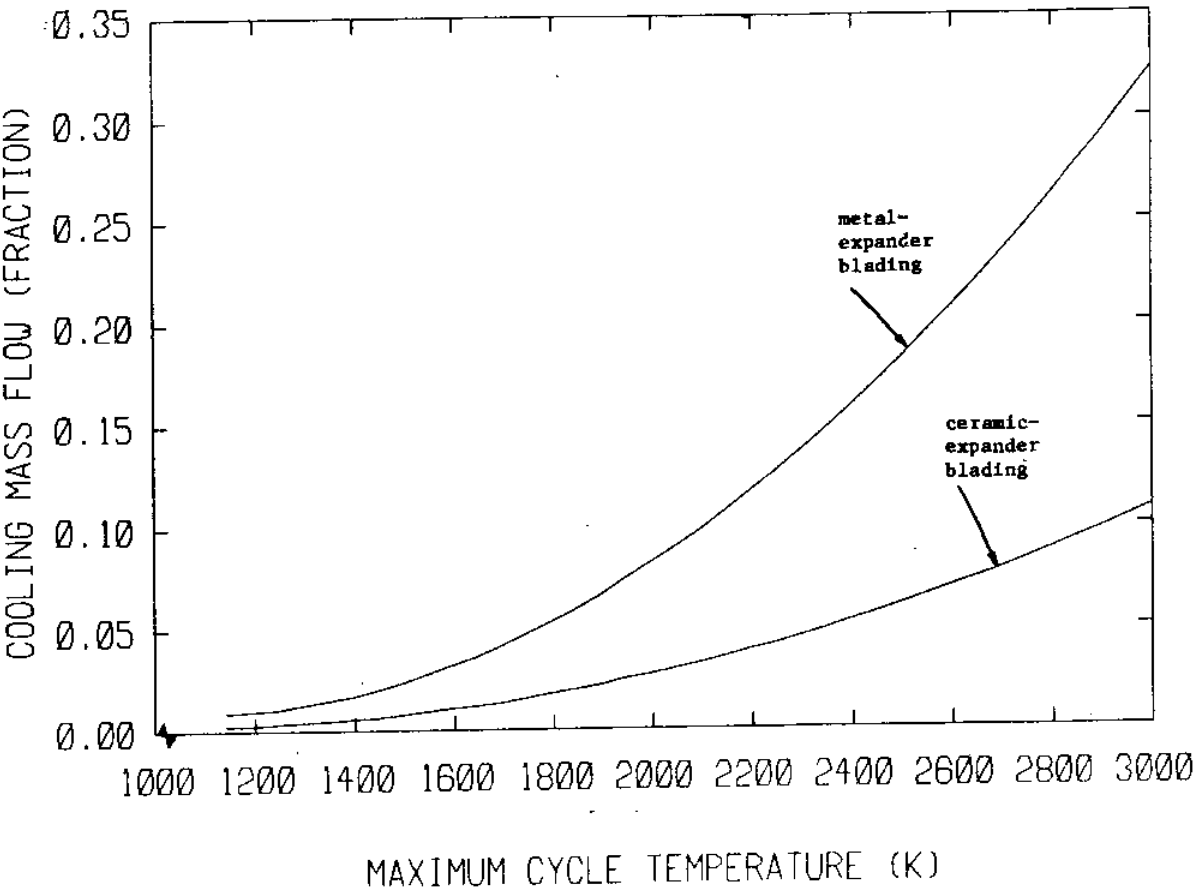


FIGURE 8 PART-LOAD PERFORMANCE OF A "CLEAN-SHEET" LPR ENGINE (estimated) COMPARED WITH PRESENT DIESEL ENGINE



From ref. 42

FIGURE 9 ESTIMATED COOLING-AIR REQUIREMENTS FOR METALLIC AND NOMETALLIC TURBINES

longer be a cost reason to keep the number of turbine stages to a minimum. The need to minimize the number of blades to be cooled by expensive compressed air would also disappear. Accordingly, in most nonaircraft situations it would be found to be economically optimum to choose to have the number of turbine stages above the minimum, and thus to gain the benefits of higher turbine efficiency that come with lower loading (figure 10).

The higher turbine efficiency that would result would have a snowball effect, because it would result in an optimum pressure ratio that would be lower than before, other things being equal. This lower pressure ratio would in turn require a compressor and turbine of lower loading and therefore higher efficiency, further lowering the optimum pressure ratio.

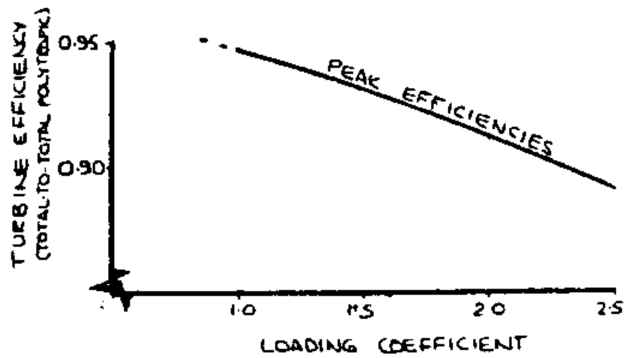


FIGURE 10 TYPICAL VARIATION OF TURBINE EFFICIENCY WITH LOADING

Using nonmetallic materials in place of alloys high in chromium and cobalt would have significant strategic advantages. There may, in addition, be reduced material attack from fuel constituents and from aerosols in the airflow in marine atmospheres.

Some of the advantages of ceramics are summarized by Helms and Heitman (ref. 7) in a paper on a GM study funded through NASA by DOE:

"Engineering use of ceramic components in heat engines is approaching a state-of-readiness for production in the near future. . . . . . The advantages offered by ceramic components employed in the (advanced gas turbine) are . . . (220-300 deg K increase in turbine inlet temperature. and 220-275 deg K increase in regenerator inlet temperature compared with metal; 70-90 percent reduction in production costs, and 60-70 percent reduction in density compared with superalloys.)

These improvements may become significantly larger as the structural (fine) ceramics technology expands into a more mature family. . . ."

The "LPR" approach increases the size 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 alternative engines, as will be seen below. The shaft speed is considerably reduced compared with high-pressure-ratio gas turbines, which is an advantage.

PURPOSE OF PRESENT PROJECT

To collect information on present and projected requirements for propulsion and auxiliary power and heating and cooling loads of fishing vessels; to carry out extensive studies of one selected class of fishing boats, arriving if possible at an optimum arrangement; to predict the design-point and off-design fuel consumption and performance and to compare this with that provided by a diesel engine. Also to investigate the possibility of producing the chosen engine type from available, perhaps used, engines and components.

RATIONALE

Because the fishing-vessel market is too small to encourage manufacturers to develop special-purpose engines, and because propulsion-system acquisition and operating costs have become a significant proportion of the costs of fishing, it seemed desirable that an independent review of alternative engines using gas-turbine technology be made. The present time also offers the prospect of outstanding opportunities to use new materials, especially high-temperature ceramics and reinforced plastics, for cost and weight reduction and the improvement of thermal efficiencies.

GUIDE TO THIS REPORT

This report is a summary of the work done by many students. They reported their work in their theses (ref. 6 and 8 through to 13), in other reports (ref. 14, 15 and 16) and in various memoranda (ref 17 through to 24). In some cases we have used parts of their work, and in others we have summarized and referred to their work for more-complete information. In the next chapter, a survey of fishing vessels is described and the choice of a "baseline" vessel for further study is made (a 36-m, 119-ft, stern trawler with a 1.1-MW, 1500-hp, engine).

Alternative forms of gas-turbine engines are discussed in chapter 3, with three engine cycles being selected for further analysis.

These three cycles are studied in chapter 4. The low-pressure-ratio highly regenerated (LPR) cycle is chosen as being the most appropriate for future fishing vessels.

In chapter 5 a turbine-inlet temperature and a compressor pressure ratio are chosen for the LPR cycle, together with a heat-exchanger effectiveness. Alternative design philosophies are pursued for, in particular, the compressor design, and an axial compressor with low blade speed and high reaction is chosen to produce a high efficiency and to allow composite materials to be considered for construction. The overall configuration of an engine that might be manufactured as a new venture is proposed.

Alternative methods of manufacture of LPR engines are studied in chapter 6. We found that some helicopter engines, for instance, that are likely to become available with a large proportion of their useful lives still to run, could be fairly easily modified to operate on an LPR cycle, while others would present an apparently uneconomic prospect of difficulty.

In chapter 7 we look at potential degrees of freedom given to the ship designer who chooses to use a gas-turbine instead of a diesel engine. The principal alternative to the use of the conventional engine location was to have the turbine engine mounted vertically on the pilot-house roof (0-1 deck) with the air inlet facing aft at the top of the engine, and the shaft passing vertically down to a right-angle bevel-gear drive in the conventional engine position. We concluded that this arrangement had some attractions, but that its advantages did not seem sufficiently large to warrant such a dramatic change from the conventional position. We also investigate the potential of using absorption chillers to satisfy the freezing requirements of a fishing boat, an approach that can be used both with a gas-turbine engine and with a diesel engine.

In chapter 8 we give the performance comparison of the LPR engine and a diesel engine (the diesel engine that currently powers the baseline fishing boat). We estimate the respective annual fuel consumptions (for propulsion only) in both U.S. gallons and 1984 dollar value. We also compare and discuss some other aspects of the two engines.

In chapter 9 we include the results of a survey of the response of the New England fishing industry to the LPR engine proposal and in chapter 10 we give the overall conclusions and recommendations of this report.

CHAPTER 2

SURVEY OF FISHING VESSELS AND CHOICE OF BASELINE VESSEL

The size and the power rating of U.S. fishing boats vary widely. In order to increase the applicability of the conclusions, we needed to base the study on a vessel size and power-rating "representative" of U.S. fishing boats. With this aim, we initially identified "representative" with "average" and undertook a survey of the industry.

According to reference 25 the average stern trawler is about 75 m (246 ft.) long and has a propulsive power of about 1500 kW (2012 hp), which allows for a cruising speed of 14.5 knots (7.46 m/s). The main winch is powered by an electric motor of 260 kW (349 hp). Additionally, trawlers are classified as: distant water, usually 41 m (134.5 ft.) and above; middle water, between 34 m and 41 m (111.5 ft. to 134.5 ft.); and near water, between 24 m and 34 m (78.7 ft. and 111.5 ft.).

Still according to reference 25, the 1973 Statistical Tables of Lloyd's Register of Shipping give the world totals at 16,374 trawlers and fishing vessels of gross tonnage 6.99 million LT. The last two numbers when divided suggest that the average fishing vessel has a displacement of 427 LT. (However this number should be used only as an indication, as the distribution is skewed).

References 26, 27 and 28, although they may now be obsolete for today's fishing industry, include a wealth of papers on fishing boats of various dimensions. Skimming through most of these papers supplied some information that suggests an average boat size of 35 m (115 ft.), displacement about 600 LT, powered by engines in the range of 597 kW to 895 kW (800 hp to 1200 hp) and a winch electric-motor power of about 186 kW (250 hp).

Reference 29 applies to fishing systems on Georges Bank. Much has changed since the study was performed, but, as an indication, the optimum vessel selected was 38 m (125 ft.) long, of 593 LT displacement, powered by 768 kW (1030 hp), providing power for cruising at speeds up to 12.5 knots (6.44 m/s).

According to reference 30 the optimum vessel for the New England shrimp fishery is 18.3 m (60 ft.) long, powered by 196 kW (263 hp) and trawling a net 25.9 m (85 ft.) long.

A collection of data on U.S. fishing boats is available in references 29, 30, 31, 32, 33, 34, and various recent and old editions of 35. It is difficult, if not impossible, to arrive at any conclusion about the size and power of the average U.S. fishing boat, because the values vary depending both on application and on location. For example, a variety of small vessels of gross tonnage 5 to 10 LT and length 9 m to 12 m (30 ft. to 39 ft.) are very popular in the Chesapeake Bay area. A variety of boats of gross tonnage from 15 LT to 65 LT and lengths from 11 m to 21 m (36 ft. to 69 ft.) are used in all other U.S. areas, the majority of vessels being in the smaller sizes. A considerably smaller number of fishing boats from 65 LT to 200 LT and 21 m (69 ft.) and above are spread throughout the fishing areas. A very small number of boats larger than that are found in the Gulf and Pacific areas. Finally, as reported in various articles in recent editions of reference 35, a large number of boats of length 27 m to 40 m (89 ft. to 131 ft.) have recently been built for U.S. owners (mainly fleet owners).

The conclusions of the survey are summarized in the following. The majority of U.S. fishing boats are of smaller size than their foreign counterparts. This is partly attributed to the fact that they are owned by proud individual fishermen who cannot (or do not want to) obtain loans from financial institutions for larger boats. Another major influence is the reluctance of U.S. fishermen to embark on long trips. The shorter trips in which they are willing to embark result in smaller catches that require smaller vessels and less refrigeration. The optimum size of U.S. fishing boats is larger than the actual size of the fishing boats currently in use, which are less efficient (in terms of catches), than they could be, and also less efficient than their foreign counterparts. However, as reported in various editions of reference 35 and in reference 36, recently some of the owners have realized that larger boats are more efficient in terms of catches and have built modern and larger boats of lengths 27 m to 43 m (89 ft. to 141 ft.). Some of these larger boats have been fitted with fixed-pitch propellers (FPP), some have been fitted with controllable-pitch propellers (CPP) and some have been fitted with controllable-reversible-pitch propellers (CRPP). Additionally, some of these vessels have been designed with shrouded propellers (Kort nozzles). These modern and larger fishing boats are an indication of the trend for future U.S. fishing boats.

At this point we decided that our research should not perpetuate inefficiency by designing the LPR engine for one of the smaller, less-efficient vessels. We redefined "representative" vessel as one of the new-generation, larger and more-efficient boats and chose as a baseline fishing boat (the boat on which the study was to be based) the vessel presented in the September 1983 issue of reference 35. This particular boat is a stern trawler of length 36.3 m (119 ft.), conceived with the New England fishing grounds in mind. It is a design by John W. Gilbert and Associates, Inc. that, with slight modifications, has been around for over ten years. Several of these vessels have been built and tested; more of the same type of vessels are on order. Sea-going experience has already been accumulated. Some of these vessels are powered by the Caterpillar D399 medium-speed diesel engine which is a 16-cylinder, vee arrangement, prechambered, turbocharged, jacket-water aftercooled, four-stroke-cycle engine, rated 839 kW (1125 hp) at 1225 rpm. Other sister vessels are powered by the replacement of the D399 model, the Caterpillar 3516 medium-speed diesel engine, which is a 16-cylinder, vee arrangement, direct injection, turbocharged, jacket-water aftercooled, four-stroke-cycle engine, available at the following continuous-power-rating options:

839 kW (1125 hp) at 1200 rpm;
1051 kW (1410hp) at 1600 rpm; and
1051 kW (1410 hp) at 1800 rpm.

Apparently some of these vessels initially fitted with the lower-power-rating engine (839 kW) did not have sufficient power in adverse-weather conditions. It was found that (for one of the versions of the baseline boat) the effective horsepower required in adverse weather conditions may increase by 30% (ref. 37). Accordingly we specified the power rating of the LPR engine at 1119 kW (1500 hp) and decided to compare its performance to that of the Caterpillar 3516 medium-speed diesel engine rated 1051 kW (1410 hp) at 1600 rpm.

According to the designer of the boat, Mr. J. W. Gilbert, (ref. 22), although the operating profile of the boat varies widely, typically it will operate about eleven months per year (one month is taken up with repairs etc.), which will allow for 24 to 32 trips. A typical ten-day, dock-to-dock trip will generally involve seven days of fishing operations and three days cruising (to and from the fishing grounds), followed by a two-to-four-day port call. For a fixed-pitch propeller (FPP), a typical operating profile is the following:

cruising from port to the fishing grounds and back at full speed (10 to 12 knots);
150 minutes cruising (at the fishing grounds) at 75% speed (about 20% power);
180 minutes trawling at 4 knots (requires about 80% power);
20 minutes hauling at about 80% power; and
120 minutes idling.

CHAPTER 3

ALTERNATIVE FORMS OF GAS-TURBINE ENGINE

A gas-turbine engine (often simply called a "gas turbine", although this term also applies just to the gas-expansion device) consists of the following components (figure 11).

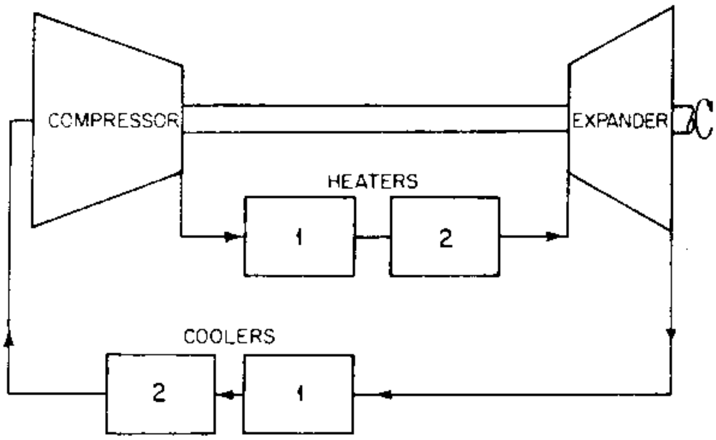


FIGURE 11 BLOCK DIAGRAM OF GENERAL GAS-TURBINE ENGINE

- A compressor, or series of compressors separated by intercoolers.
- A heat-addition unit, or series of units. The first of these may transfer heat from the turbine exhaust.
- An expander or turbine, or series of turbines. Conceptually, the turbines could be separated by "interheaters", but in practice this step has not been taken.
- A cooler or series of coolers to take the gas back to the temperature at which it enters the compressor. The first of these coolers may be the heat-transfer device - the heat exchanger - mentioned above. The last "cooler" in an "open-cycle" gas turbine using air is the atmosphere.

In a "closed-cycle" gas-turbine engine the same gas, which could be air but is usually something having more convenient properties such as helium, is enclosed within the system and recycled. In an "open-cycle" gas turbine the compressor draws in air from the atmosphere and the exhaust is rejected hot to the atmosphere (figure 2). The environment therefore accomplishes the cooling and regeneration of the air. The thermodynamic-cycle model is always closed.

Since the thermodynamic cycle is always closed, it can be conceptually entered at any point. The compressor is not necessarily the "first" component. We examined an adaptation of an arrangement in which the turbine expander came first.

We rejected, however, true closed-cycle machines. They have several advantages, including outstanding part-power efficiencies. But because the enclosed high-pressure gas has to be heated through the walls of a heat exchanger, which at the present state of technology must be metallic, the maximum gas temperature has to be limited to 1200 K, 1700 F, too low to produce attractive design-point performance.

We also rejected the simplest open-cycle gas turbines. These consist of a compressor, combustion system and expansion turbine - termed a "CBE" cycle. The highly successful gas-turbine engine used in many US Navy ships, the LM 2500, works on the simple cycle. To attain high efficiency in design-point operation a CBE cycle must employ not only high turbine-inlet temperatures but very high pressure ratios, over 25 to 1. Such compressors have very poor part-load performance unless they are made extremely complex by incorporating pivoting stator blades, or by breaking up the compressor into several separately driven units. Such an engine would not be appropriate for future fishing vessels.

We did study three variations of the general gas-turbine-engine cycle. The simplest of these variations, and the one we eventually recommended, is the regenerative cycle, or CBEX (for "Compressor-Burner-Expander-heat-eXchanger"). We also examined a variation of this cycle having an intercooled compressor, the CICBEX cycle. The third cycle we studied was more complex, and needs some explanation.

A simple-cycle CBE gas turbine has an exhaust of fairly high temperature, a fact that is connected to its generally modest cycle efficiency. From the earliest industrial gas turbines it has been common practice to use the exhaust heat in various ways. When maximum power output and maximum overall efficiency is required, the exhaust heat can be used as input to a steam-turbine cycle. The combination of a gas turbine with a steam turbine is called a "combined cycle". Efficiencies in the mid-fifties have been attained in central-station units, a very substantial increase over efficiencies in the low forties that have been the peak for steam cycles alone.

The combination of gas and steam technology for a small vessel is, however, less attractive partly because of the stringent maintenance skills required, and partly because much of the efficiency advantage is lost when steam turbines of small size are used. We elected to study an alternative to the steam "bottoming" cycle: a gas-turbine "bottoming" cycle, producing power from a stream of hot gases.

Such a cycle is obtained when the general gas-turbine engine is entered at the expander inlet. After the expansion the gas is further cooled. then compressed, and finally discharged to the atmosphere. The cycle produces power because more work is given by expanding hot gas than is taken by compressing the same gas when cold. This cycle alone is termed the "inverted cycle". We called the combination of the simple cycle with the inverted cycle the "direct-plus-inverted" cycle, or DIC cycle. It had attractions for fishing-vessel propulsion because it would involve a relatively simple "add-on", rather like a turbocharger and a waste-heat boiler, to an off-the-shelf simple-cycle gas turbine. Unfortunately its performance did not measure up to its promise. The results of the cycle studies are given in the next chapter.

CHAPTER 4

THERMODYNAMIC CYCLE STUDIES

BACKGROUND

Selection of an appropriate thermodynamic cycle is fundamental to the gas-turbine design process. In this section we have summarized the results of a preliminary comparison made among the simple Brayton cycle (CBE) and the three proposed cycle modifications (CBEX,CICBEX and DIC). 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 rank the cycles in terms of potential gains over simple-cycle (CBE) performance. Our conclusions are that the low-pressure-ratio heat-exchanger cycle (CBEX) is the most promising; that the intercooled cycle (CICBEX) may be equal or slightly higher in efficiency and smaller in size, but is more complex and probably less reliable; and that the direct-plus-inverted cycle (DIC), while providing substantial performance gains for existing installations, and having a very high specific power, is not competitive with the heat-exchanger cycle for a new application.

Two factors motivating development of gas-turbine engines for small-scale marine propulsion are the need for increased fuel efficiency, and the need to provide onboard waste-heat in a readily usable form. This latter requirement is particularly appropriate to marine propulsion, where refrigeration chillers and various forms of processing are necessary for many applications.

All of the three proposed cycles are capable of thermal efficiencies substantially higher than the upper limits attainable using diesel technology, and all provide waste-heat in the form of non-noxious exhaust gas. All of the proposed cycles, but especially the CBEX cycle, provide opportunities for the use of high-temperature ceramic components, which are expected to find widespread application in small engines.

The higher efficiency of the proposed cycles is realized at the expense of increased size and weight over simple-cycle gas turbine engines, although size and weight reductions may be possible compared to diesel engines. These size and weight penalties do render the proposed cycles inappropriate for aircraft application, which is the reason for the reluctance of the major gas-turbine engine manufacturers to commit private funds to the investigation of these cycles.

The preliminary cycle analysis reported here has been used to examine the potential of each candidate cycle. This analysis served as groundwork for the preliminary design studies that follow.

An invaluable tool in connection with this work has been a computer code developed by Andre By at Northern Research Inc., Woburn, Massachusetts and modified at the joint computer facility of MIT by Thomas Wolf and Theodosios Korakianitis. This code calculates performance characteristics for a wide variety of cycle configurations, given specified operating conditions and component efficiencies. Although the algebraic relationships used in preliminary cycle analysis are straightforward, hand calculation is laborious and subject to error, so that much efficiency has been gained through automation of the procedure.

Since the purpose of the thermodynamic cycle analysis is to identify the efficiency potential of each cycle, and because of the large number of parameters to be specified, a complete exploration of performance characteristics for each of the candidate cycles would be an ambitious undertaking, and not necessarily worth-while for the purpose of the analysis. Our strategy has been to specify component-performance parameters representative of hardware currently available in the size and cost range of interest, and to include up-graded performance estimates in areas where recent experimental evidence indicates possibilities for improvement during the next several years. To be specific, increases in maximum permissible turbine inlet temperatures are expected with continued development 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.

CANDIDATE CYCLES: GENERAL CONSIDERATIONS

The reader is referred to one of the standard references (1, 38, 39) for complete discussions of gas-turbine cycles and their analysis. The present section is intended to facilitate interpretation of the results which follow, and to introduce some important definitions.

SIMPLE BRAYTON CYCLE (CBE)

All of the three cycles to be investigated can be thought of as derivatives of the simple (CBE) cycle, pictured in figure 12, which in its ideal form is comprised of isentropic compression, followed by heat addition at constant pressure, followed by isentropic expansion to ambient static pressure. The net power produced by the cycle is given as the difference between the power produced during expansion and the power absorbed during compression. Losses in the non-ideal cycle appear as entropy increases during the compression and expansion processes, and as the pressure drop during combustion, all of which can be seen to decrease the net power of the cycle. Two quantities of key interest are the thermal efficiency, defined as the power output of the cycle divided by the rate of heat 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, heat capacity, and stagnation temperature at inlet. The thermal efficiency is a partial measure of the fuel efficiency of the engine, although other losses such as bearing and disc friction are not included in its definition, and these will reduce overall fuel efficiency by perhaps 1-5 percent. 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.

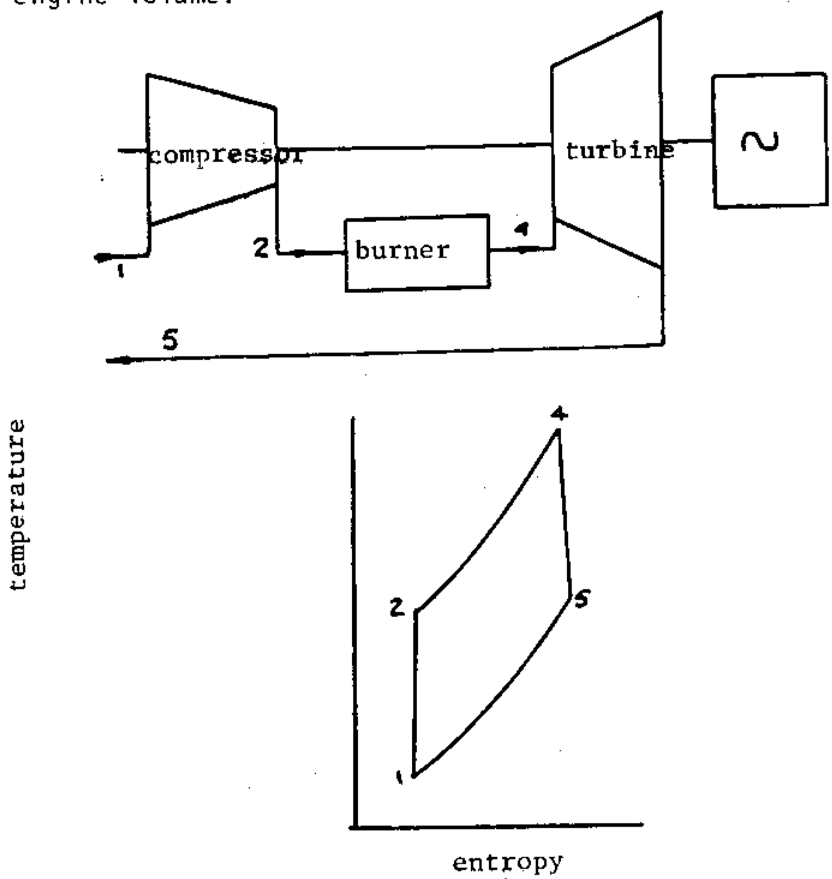


FIGURE 12. BLOCK DIAGRAM AND TEMPERATURE-ENTROPY DIAGRAM OF A SIMPLE GAS-TURBINE CYCLE.

In the simple cycle the energy of the hot turbine exhaust is wasted; increasing the pressure ratio of the cycle (for fixed turbine-inlet temperature) reduces exhaust temperature, thereby increasing 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. Using typical aircraft gas-turbine temperature ratios and component efficiencies, this optimum pressure ratio turns out to be quite high, i.e. greater than twenty.

LOW-PRESSURE-RATIO-REGENERATED CYCLE (CBEX)

In the case of the CBEX cycle, shown in figure 2, the requirement of low turbine exhaust temperature is obviated because heat is transferred from the turbine exhaust to the combustor inlet via a heat exchanger. As a result, optimum pressure ratios are much lower than for the simple cycle. A CBEX cycle incorporating "perfect" components can be shown to have a pressure ratio of unity; if a high-effectiveness (roughly 95% or greater) heat exchanger is used, and assuming typical temperature ratios and component efficiencies, optimum pressure ratios in the two-to-six range are found. (The heat-exchanger effectiveness is defined as the actual increase in cool-side temperature divided by the theoretical maximum increase associated with heating the compressed air to the turbine-exhaust temperature.)

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 small 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 favors the reliable operation of ceramic turbines. Compromises take the form of reduced specific power and the necessity of fitting the heat exchanger, both of which contribute to increased engine weight and volume.

INTERCOOLED-REGENERATED CYCLE (CICBEX)

The thermal efficiency of the CBEX cycle can potentially be improved further still by segmenting the compression process and cooling to near ambient temperature between stages (figure 13). This practice of "intercooling" capitalizes on the fact that the compression work required for a specified pressure ratio is directly proportional to inlet temperature. However, intercooling will increase the optimum pressure ratio of the cycle, which is likely to degrade heat-exchanger effectiveness if a rotating ceramic regenerator is to be used.

From the standpoint of preliminary cycle analysis, important issues to be addressed include the possibility that the expected pressure loss across the intercooler may be sufficient to nullify gains, and that the increase in optimum pressure rario may jeopardize heat-exchanger performance.

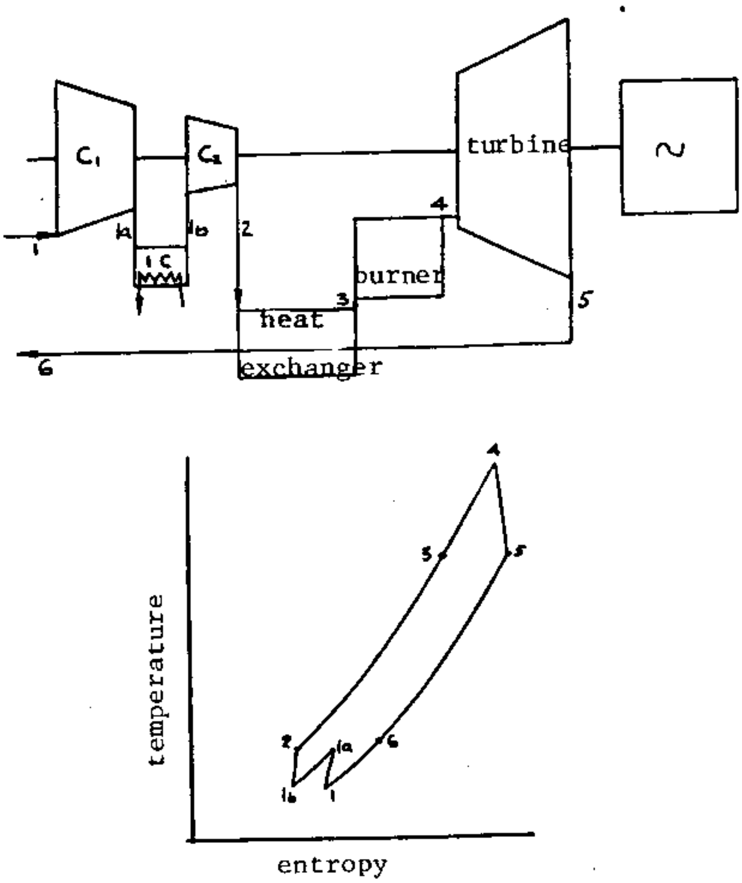


FIGURE 13. BLOCK DIAGRAM AND TEMPERATURE-ENTROPY DIAGRAM OF AN INTERCOOLED-REGENERATED GAS-TURBINE CYCLE.

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 recompressing to atmospheric pressure (figure 14). The thermal efficiency of a simple cycle can be improved with 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. An optimum inverted-cycle pressure ratio can be found for a specified simple cycle that gives maximum enhancement of overall thermal efficiency; at a fixed temperature ratio this optimum pressure ratio is found to decrease monotonically with increasing simple-cycle pressure ratio.

Preliminary cycle analysis can be used to estimate potential gains in efficiency, and to establish whether or not inverted-cycle pressure ratios can be kept sufficiently low that the scheme can continue to be regarded as a low-cost "add-on."

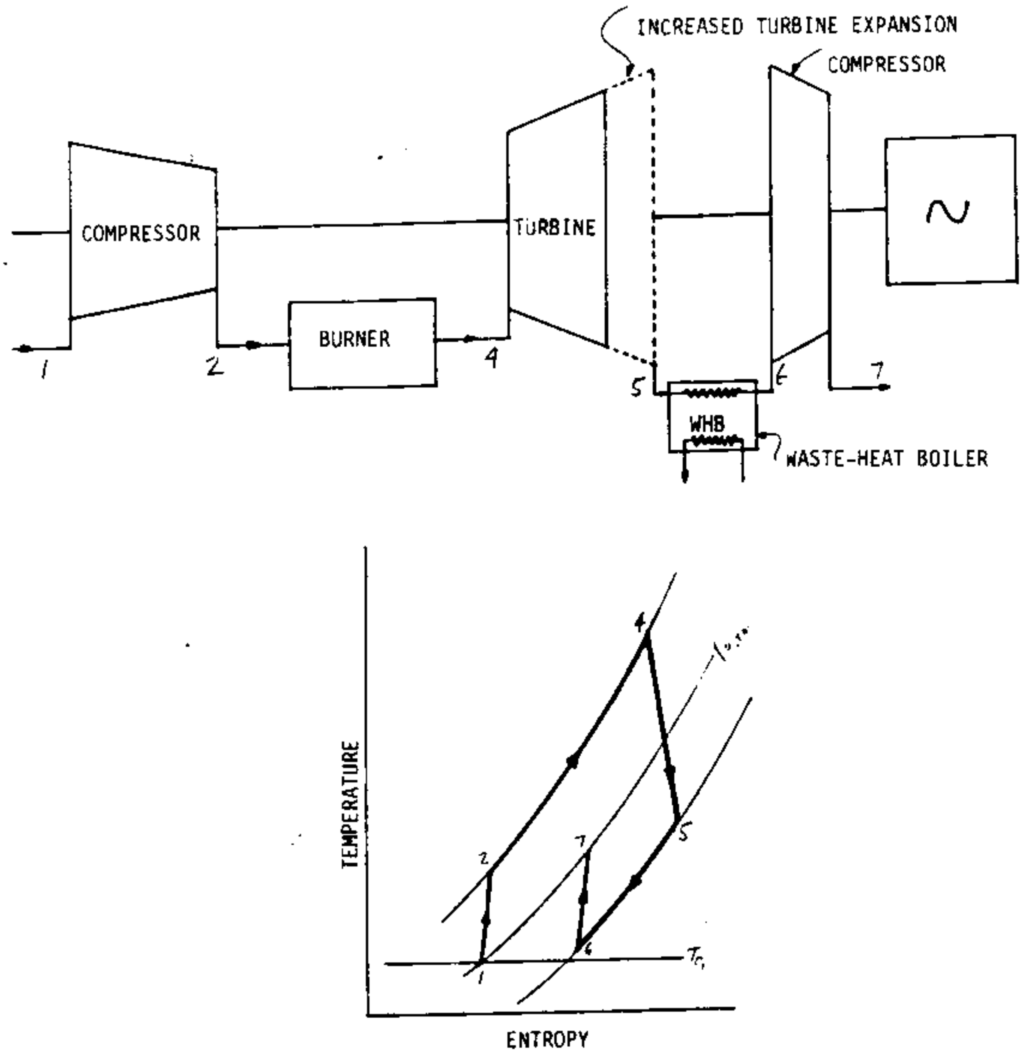


FIGURE 14. BLOCK DIAGRAM AND TEMPERATURE-ENTROPY DIAGRAM OF A DIRECT-PLUS-INVERTED GAS-TURBINE CYCLE.

SPECIFICATION OF CYCLE PARAMETERS

The CBE cycle is a useful reference for evaluating the relative gains of the three proposed cycles. CBE cycle parameters have been specified as 86% and 89% for compressor and turbine polytropic efficiencies, respectively, with a 6% pressure drop across the combustor. These values are typical of a well-designed engine in the 200-2000 hp size range. Temperature ratios from 4 to 7 will be considered, encompassing a broad spectrum from routinely manufactured commercial engines to experimental prototypes using ceramic components. Pressure ratios as high as 20 will be investigated, althogh the upper end of this range is likely to be inaccessible if low cost is to remain a priority.

To investigate performance using the CBEX cycle, the component efficiencies of the CBE cycle are again adopted, with the additional specification of either a 95% or 98% regenerator effectiveness. The lower value has been consistently achieved in experimental automotive applications, and the higher value is the anticipated target of the present program. Pressure drops across the hot and cool sides of the regenerator have been specified as 5%, which is perhaps somewhat conservative.

For the CICBEX cycle, additional specification of a 95% effectiveness and 3% pressure drop for the intercooler seems reasonable· this performance should be readily attainable given the availability of seawater as a cooling medium.

Evaluating direct-plus-inverted cycle performance is rather less straightforward· the designer has wide freedom in specifying inverted-cycle components. depending on the intended application and economic constraints. Because low cost is an attractive feature of this approach, it is unlikely that a highly efficient compressor would be chosen for the inverted cycle. We have specified an 80% polytropic efficiency for this compressor, which probably represents a reasonable compromise. A 3% pressure drop has been specified across the heat exchanger. which is again a typical value.

As already mentioned, an optimum inverted-cycle pressure ratio 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.

RESULTS· COMPARISON OF THE PROPOSED CYCLES

Plots of thermal efficiency vs. specific power for the CBE cycle (reference case) and each of the three proposed cycles are shown in figures 15 through 20. The optimum inverted-cycle pressure ratio as a function of the direct-cycle pressure ratio was calculated for the cycle performance parameters of figure 20 and the results are shown in figure 21. The performance data are summarized in figure 22.
In each of figures 15 through 20 the curves connect operating points of constant temperature ratio, and pressure ratio varies along the curves as indicated. For the direct-plus-inverted cycle (figure 20), each of the operating points on the plot is calculated at the corresponding optimum pressure ratio. In figure 22. performance at optimum pressure ratio is shown for every instance in which the optimum pressure ratio is 20 or less· where the optimum pressure ratio exceeds 20, as in the case of the CBE and direct-plus-inverted cycles, performance is calculated at a pressure ratio of 20.

With reference to figure 22a, the relative ranking of maximum thermal efficiencies is led by the high-effectiveness CICBEX cycle, followed by the lower-effectiveness CICBEX cycle and high-effectiveness CBEX cycle. The direct-plus-inverted cycle has the lowest efficiency, not substantially ahead of the CBE cycle except at rather high temperature ratio. Thus from the standpoint of thermal efficiency. the regenerated cycles look very attractive, whereas the direct-plus-inverted cycle is not very interesting.

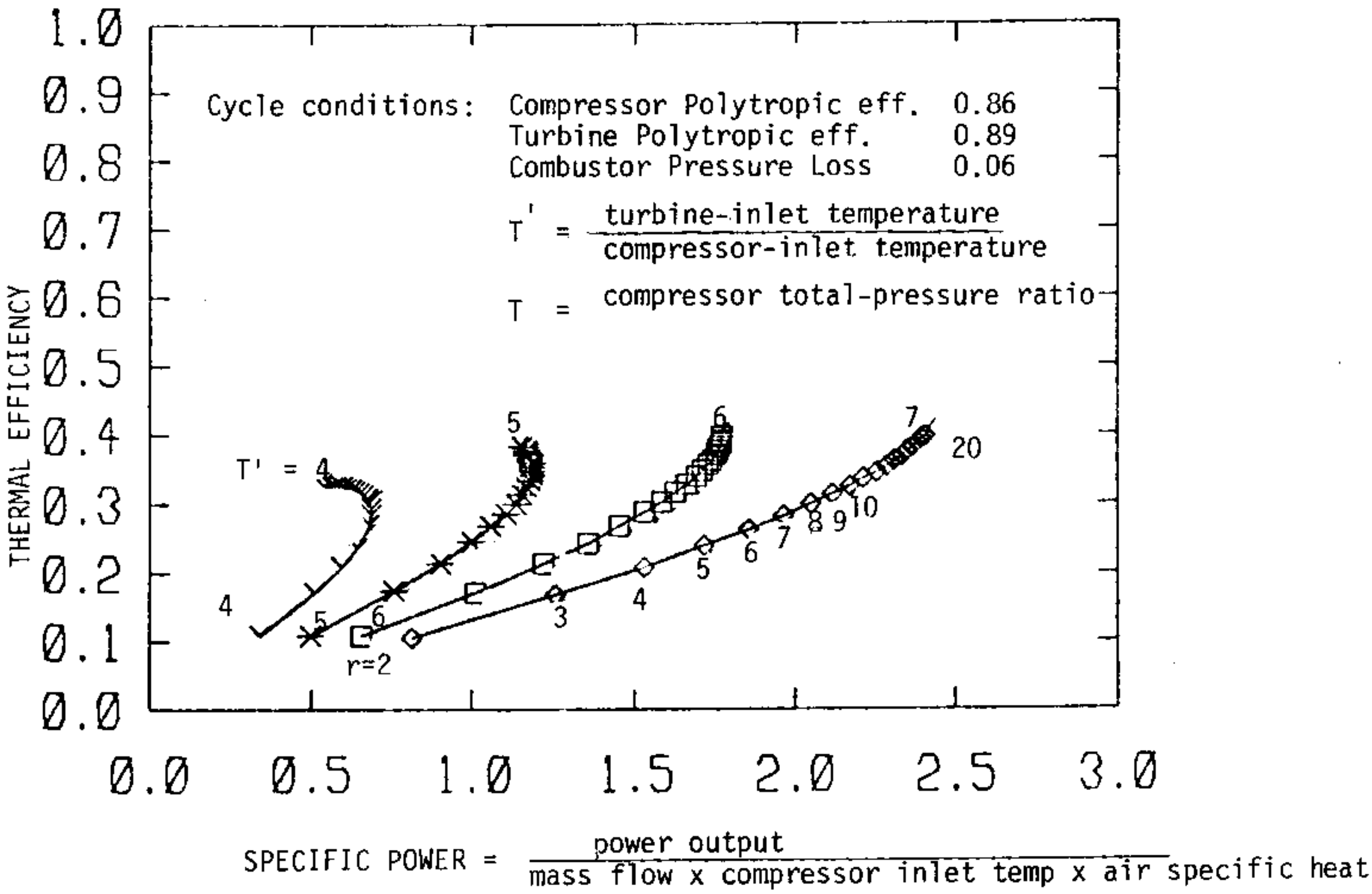
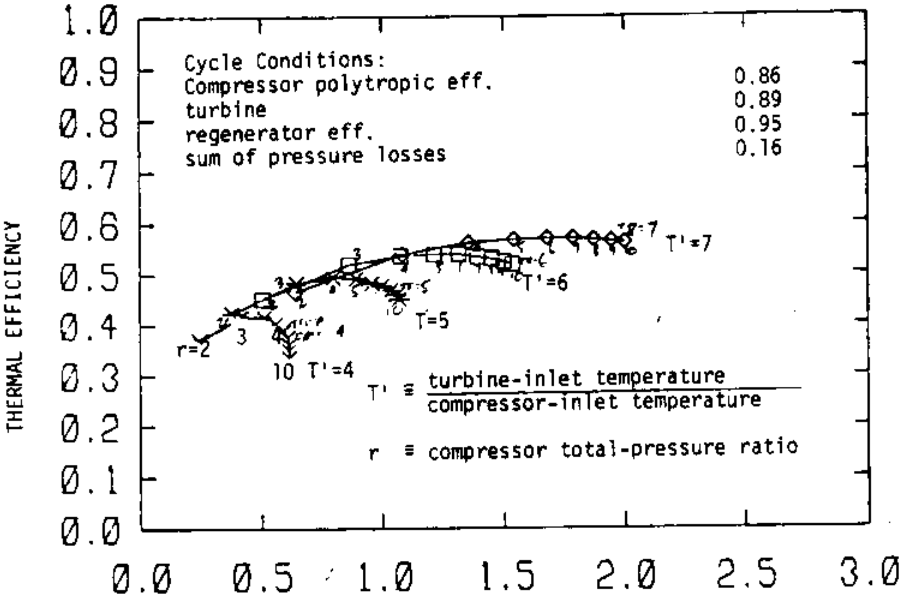
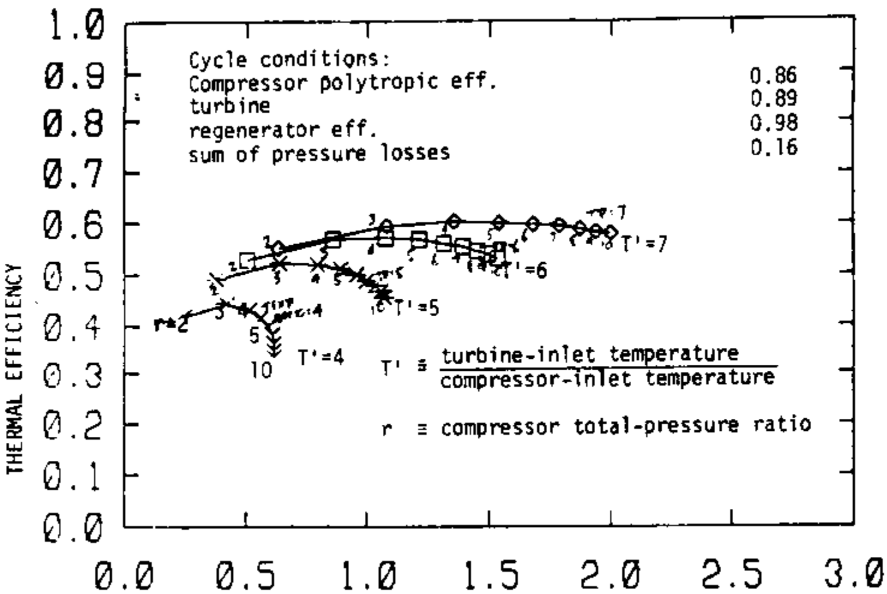


FIGURE 15. SIMPLE (CBE) CYCLE DESIGN-POINT EFFICIENCY VERSUS PRESSURE RATIO AND TEMPERATURE RATIO



SPECIFIC POWER ≡ $\frac{\text{power output}}{\text{mass flow x compressor inlet temp. x air specific heat}}$

FIGURE 16. REGENERATIVE (CBEX) CYCLE THERMAL EFFICIENCY VERSUS TEMPERATURE AND PRESSURE RATIOS; HEAT-EXCHANGER EFFECTIVENESS 0.95



SPECIFIC POWER ≡ $\frac{\text{power output}}{\text{mass flow x compressor inlet temp. x air specific heat}}$

FIGURE 17. REGENERATIVE (CBEX) CYCLE THERMAL EFFICIENCY VERSUS TEMPERATURE AND PRESSURE RATIOS; HEAT-EXCHANGER EFFECTIVENESS 0.98

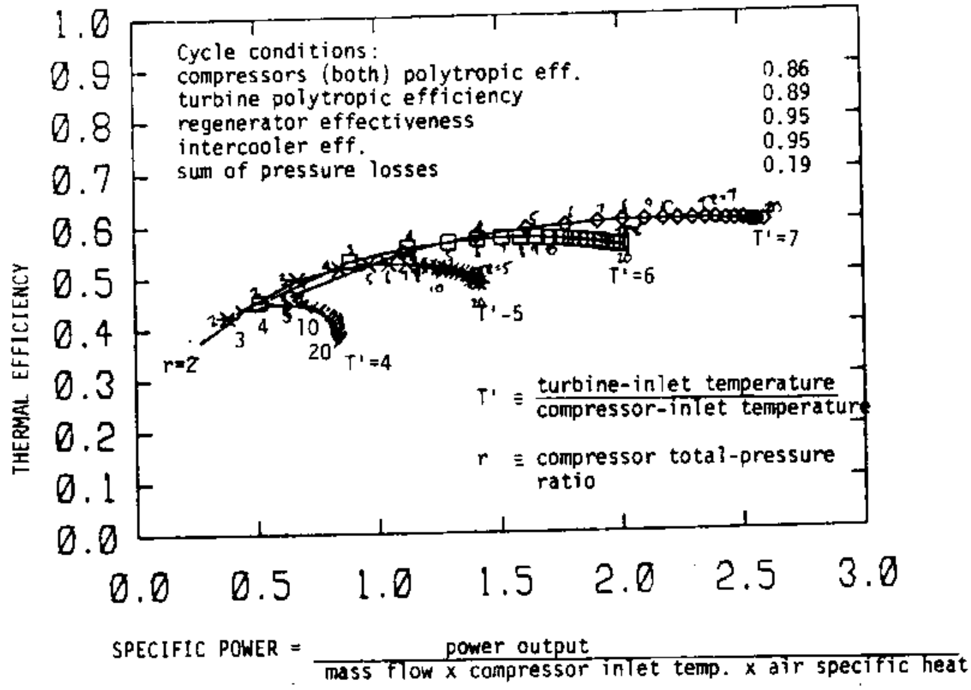


FIGURE 18. INTERCOOLED (CICBEX) CYCLE THERMAL EFFICIENCY VERSUS TEMPERATURE AND PRESSURE RATIOS; HEAT-EXCHANGER EFFECTIVENESS 0.95.

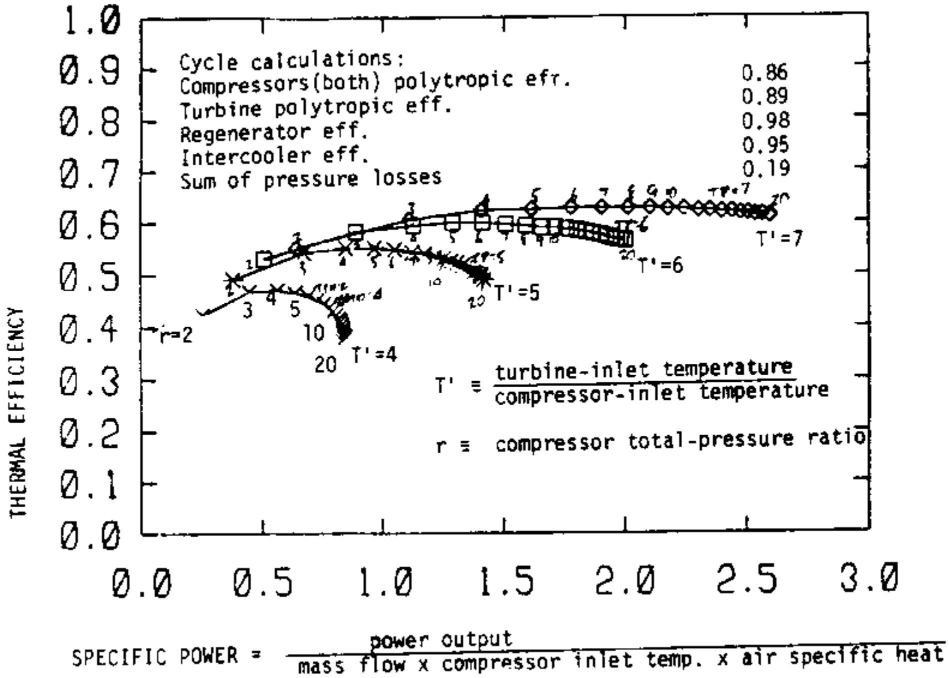


FIGURE 19. INTERCOOLED (CICBEX) CYCLE THERMAL EFFICIENCY VERSUS TEMPERATURE AND PRESSURE RATIOS; HEAT-EXCHANGER EFFECTIVENESS 0.95.

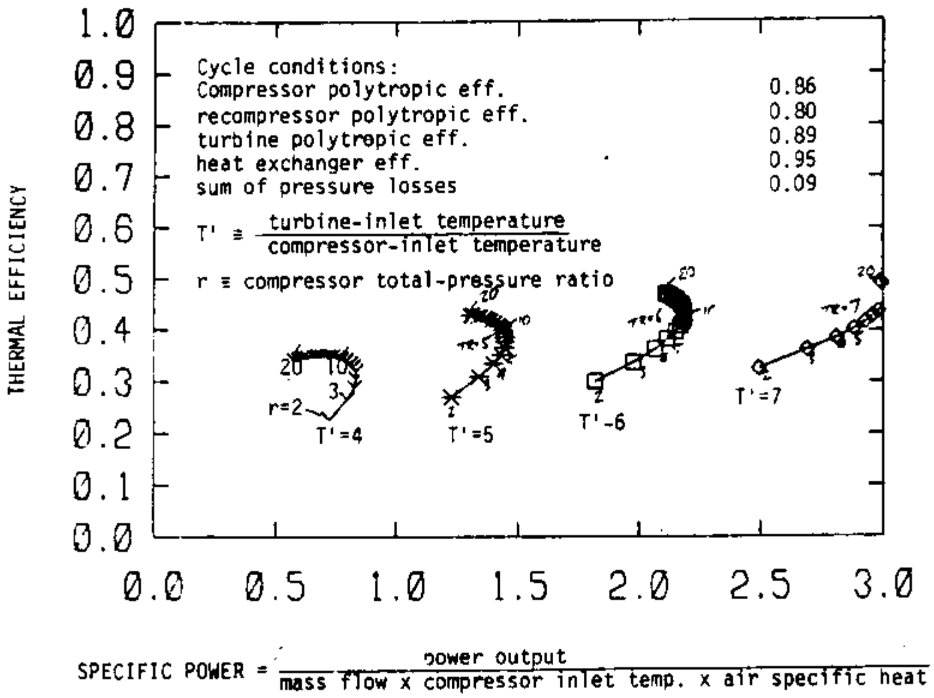


FIGURE 20. DIRECT-PLUS-INVERTED (DIC) CYCLE THERMAL EFFICIENCY VERSUS TEMPERATURE AND PRESSURE RATIOS.

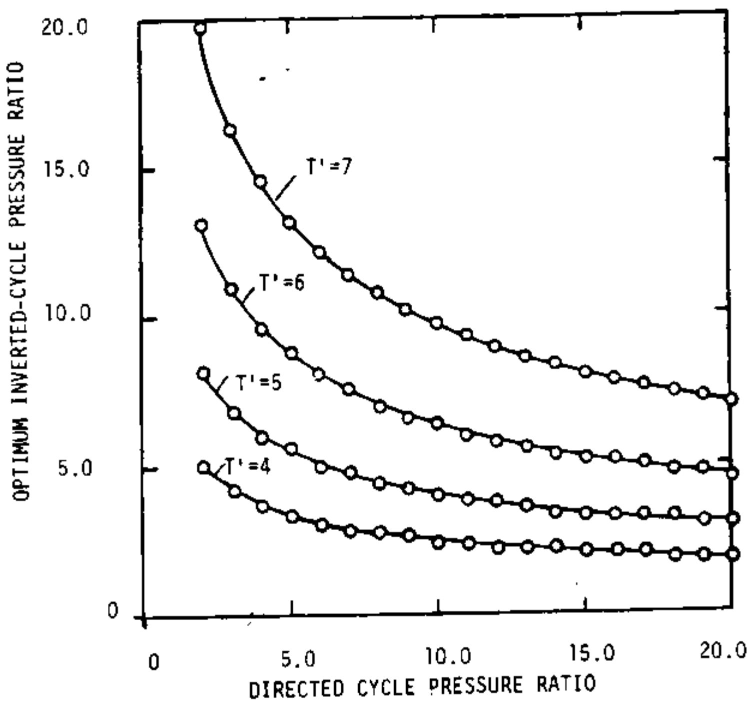


FIGURE 21. OPTIMUM PRESSURE RATIOS FOR INVERTED CYCLES AS FUNCTION OF DIRECT-CYCLE PRESSURE RATIO.

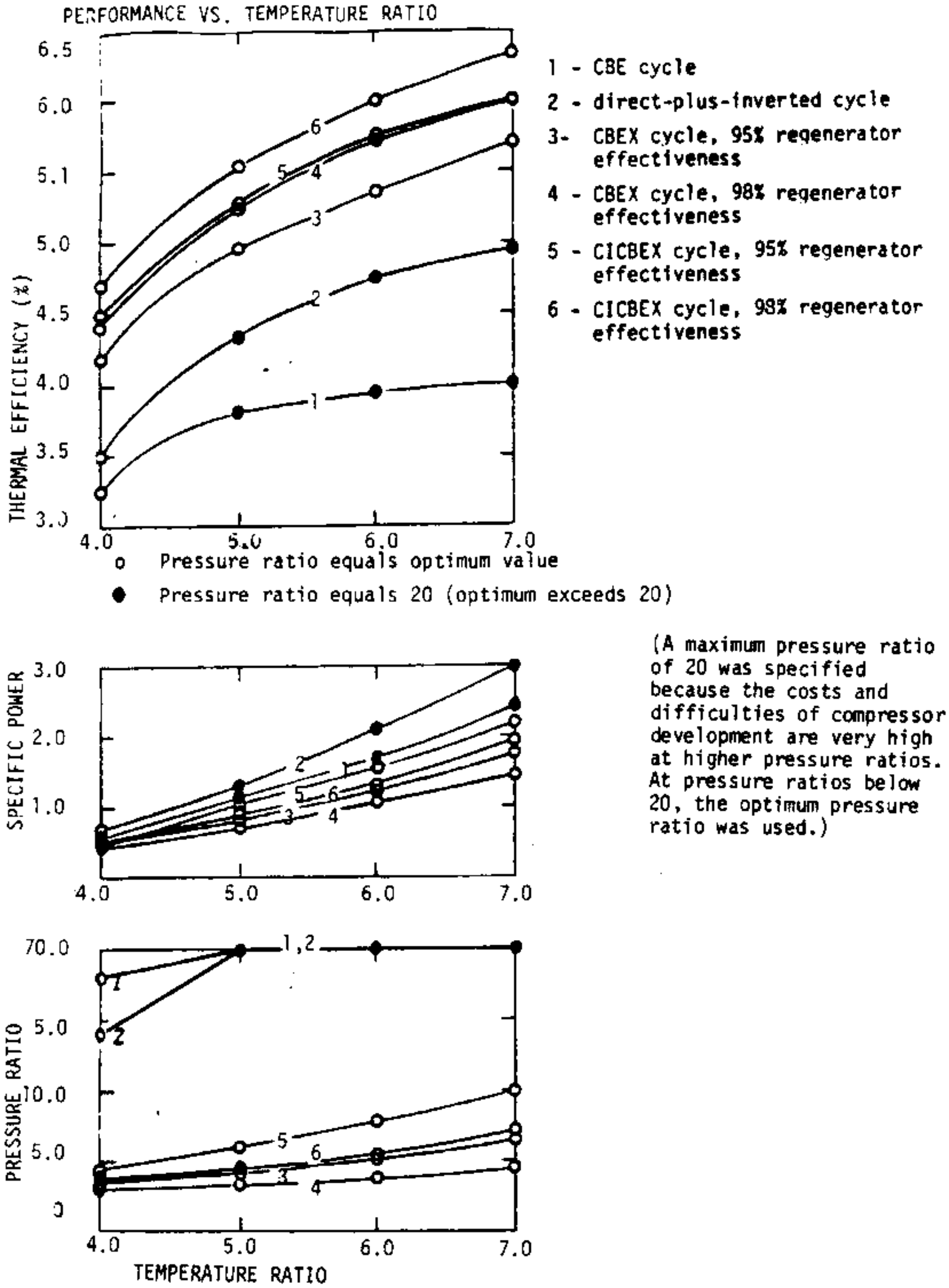


FIGURE 22. SUMMARY OF PERFORMANCE OF ALTERNATIVE CYCLES

The direct-plus-inverted cycle suffers from the additional disadvantage that the optimum pressure ratio is much higher than for the regenerated cycles, which raises serious questions as to the economic attractiveness of the scheme for new engine development. Even if some compromise in thermal efficiency can be tolerated and lower-than-optimum pressure ratios are used, this approach leads to increased optimum inverted-cycle pressure ratio (figure 21); compromises can of course be made here as well, but at further expense of efficiency. Finally, the attractiveness of reduced pressure ratios with respect to the survivability of ceramic components is a further incentive to dismiss the direct-plus-inverted cycle from serious consideration. The direct-plus-inverted cycle nonetheless retains its appeal as an add-on to existing simple-cycle engine.

This leaves the regenerated cycles as the primary focus, and here the advisability of intercooling becomes an issue. The use of intercooling is seen to enhance thermal efficiency somewhat, provided regenerator performance is unimpaired, but the increased optimum pressure ratios indicated in figure 22c are bound to degrade regenerator performance. Here the necessity of maintaining a very high regenerator effectiveness is evident; the reduced optimum pressure ratio associated with a high degree of regeneration favors good regenerator performance, which in turn contributes to the reduced pressure ratio, etc. The key question is whether or not a regenerator of reasonable proportions can be made to give a 98% effectiveness using moderate pressure ratios, say around 5:1. If this can be accomplished, then intercooling obviously makes sense.

One strategy that would facilitate high regenerator effectiveness in an intercooled cycle would be to specify a design pressure ratio below optimum; figures 18 and 19 indicate that considerable reduction in pressure ratio is possible without substantial sacrifice in efficiency, particularly at high temperature ratio. Obviously this approach cannot be carried too far, otherwise the efficiency falls to the level of the non-intercooled cycle, defeating the purpose. Also, specific power is directly diminished, possibly leading to unreasonably large increases in engine volume.

CONCLUSIONS

For each of the three high-efficiency cycles proposed for small-scale marine propulsion, preliminary cycle calculations have been performed over the expected range of design-point operating conditions, using present and projected values of component efficiencies.

Dramatic gains over thermal efficiency of the simple cycle were found using the low-pressure-ratio-regenerated and the intercooled-regenerated cycles, but results were less promising using the direct-plus-inverted cycle. The direct-plus-inverted cycle also suffers from the requirement of high pressure ratio, which from the standpoint of new engine development generally implies high cost and reduces the likelihood that ceramic components can be utilized successfully. Therefore the direct-plus-inverted cycle was found to be an attractive alternative only as an add-on to an existing simple-cycle engine.

The intercooled-regenerated cycle was found to offer potentially significant gains in efficiency over the low-pressure-ratio-regenerated cycle, provided regenerator performance is not appreciably degraded at the higher optimum pressure ratios associated with intercooling. A worthwhile strategy may be to design intercooled-regenerated engines with lower-than-optimum design-point pressure ratio, which was found at high temperature ratio to have only relatively small effect on thermal efficiency for rather significant reductions in pressure ratio.

In chapter 5 we look more closely at the regenerated and intercooled-regenerated cycles, introducing losses not considered in the simplified approach so far used.

CHAPTER 5

PRELIMINARY DESIGN OF A BASELINE ENGINE

BACKGROUND

We undertook a preliminary design for two reasons. One was to give people who might consider manufacturing or using the LPR engine a set of example specifications covering overall size, speed, number of stages and so forth. While a final design might vary in some or all respects, the differences are not likely to be major. The second reason was to give us a specific engine to analyze, particularly with respect to its part-load performance.

CHOICE OF TEMPERATURE RATIO

With improvements in materials and cooling techniques the allowable turbine-inlet temperatures are gradually increasing. Currently turbine-inlet temperatures are over 1200 K for uncooled turbines and up to 1800 K for cooled turbines (from figure 1). For the LPR engine we limited the combustor-exit temperature to 1555 K (2330 F), which for an ambient temperature of 288 K (58 F) makes T' about 5.4. This value for combustor-exit temperature is typical of current naval gas-turbine engines with metal blading (ref. 40), and therefore allows the LPR engine to be designed with metal or ceramic blading.

CHOICE OF PRESSURE RATIO AND HEAT-EXCHANGER EFFECTIVENESS

The primary choice is of heat-exchanger effectiveness, as this choice determines the optimum pressure ratio. Thermodynamically, this can be interpreted as an alternative way of transferring energy from the post-heat-addition to the pre-heat-addition phases of the cycle. In a simple CBE cycle, the energy is transferred as shaft energy from a high-pressure-ratio turbine to a high-pressure-ratio compressor. In a heat-exchanger cycle, some of the energy can be transferred as heat, so that the shaft-energy transfer can be less and the pressure ratio accordingly reduced.

In the past, a combination of lack of experience and technological limitations put restrictions on the maximum effectiveness for which gas-turbine heat exchangers could be designed (figure 4). Cryogenic heat exchangers, handling extremely clean fluids at low temperatures, have by necessity had effectivenesses of over 99 percent in many duties. Gas-turbine heat exchangers handle large volumes of fairly dirty gases over a very large range of temperatures up to a present maximum of about 1100 C (2000 F). Only the development of the rotary ceramic regenerator has made it possible to design for high effectivenesses at these temperatures.

The Allison GT-404 engine, for instance, has twin rotary ceramic regenerators giving an effectiveness of between 0.95 and 0.96. We chose to halve the thermal losses of the 0.95-effectiveness regenerator by taking a value of 0.975. As a simplistic guide to the design of rotary regenerators one can regard the diameter as set by the mass flows of the compressed air and exhaust, and the thickness as set by the effectiveness. Increasing the effectiveness from 0.95 to 0.975 would entail, approximately, doubling the thickness of the regenerator disk. Because much of the overall heat-exchanger volume is taken up with headers, seals, drive mechanisms and so forth the overall heat-exchanger casing volume is far from doubled when the disk thickness is doubled. There are even advantages. One is that designers often fall into the trap of having the cold-side pressure drop too small, resulting in poor velocity distribution across the regnerator face. Doubling the core thickness helps to distribute the flow evenly.

TURBINE MATERIALS

As discussed earlier, the turbine stator vanes and 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, and ceramic coatings of metal blades, and, in the past few years, into the use of ceramic and other nonmetallic materials from which vanes, blades and disks 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. 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.

We chose our design parameters - in particular the blade speed - to give especially favorable conditions for ceramics, in reducing the steady-state centrifugal stresses and the foreign-object-impact stresses. However, in order to be conservative it seems unwise to require that the design we advocate use ceramics for the turbine vanes and blades. We have therefore carried out our cycle studies with two alternative assumptions: that metal blades would be used, and would require cooling air bled from the compressor discharge; or that the turbine vanes and blades would be manufactured of nonmetallic material, and would require compressor cooling only for bearings, seals, shaft and possibly disks and casing. We used cooling flows that varied with turbine-inlet temperature as described below.

CHOICE OF CYCLE

It was found (ref. 20) that the performance of the LPR engine is greatly affected by all secondary flows (such as cooling flows, leakages etc.). We therefore modified the computer code which was used to produce the results of chapter 4 to incorporate the above flows by deducting the sum of the cooling-air-flow fraction and the regenerator-leakage fraction from compressor delivery and assuming that they were re-admitted to the cycle at engine exhaust. The cooling and leakage flows were modelled with the following reasoning.

For uncooled turbine blades the blading-material temperature (and in particular the first-row blading) is dominated by the temperature of the working fluid at turbine inlet. Turbine-material properties deteriorate at high temperatures. For today's materials the alloy temperatures are limited to 1200 K for industrial applications (ref. 41). For marine applications the maximum-metal temperatures are limited to 1145 K (to avoid sulfide corrosion in the marine enviroment). This temperature limit is higher for ceramic materials.

To increase the cycle temperature ratio T', while limiting turbine-blading temperatures to the above levels, cooling air is extracted from compressor delivery and admitted to the turbine blades to lower the temperature of the first and sometimes the second row of blading in most modern gas-turbine engines. To limit blading-material temperature to 1145 K the required cooling-flow fraction increases with maximum cycle temperature.

The temperature-difference ratio, w, is defined (ref. 42) as:

$w \equiv Ta/Tb$, where

$Ta \equiv$ (turbine inlet temperature - blading material temperature), and

$Tb \equiv$ (turbine inlet temperature - compressor outlet temperature).

The highest cooling-air-flow fractions are required for film-convection-cooled vanes (figure 23 of reference 42). Based on this figure we calculated the cooling-air-flow fractions for blading material temperature of 1145 K and compressor exit temperature of 408 K. We assumed that the cooling-air-flow fraction required for ceramic blading was 1/3 of the cooling- air-flow fraction required for metal blading. The results are shown in figure 9.

We assumed that a rotary regenerator would be used (which is justified by the results below). The mass leakage for rotary regenerators was modelled as shown in figure 23. It was assumed that there would be a minimum amount of seal leakage even for the smaller regenerator, and that it increases with effectiveness (and hence size).

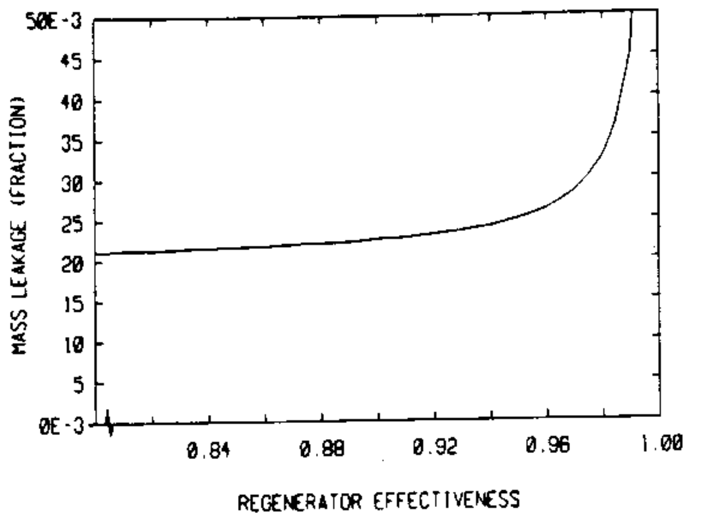


FIGURE 23. MODEL OF REGENERATOR-MASS-LEAKAGE FRACTION AS A FUNCTION OF REGENERATOR EFFECTIVENESS

The modified computer code was run for the CBEX and CICBEX cycles (the most promising cycles in chapter 4) for the conditions shown in table 1. The performance plots for the two cycles and for high (metal blading) and low (ceramic blading) cooling-flow fractions are shown in figures 24, 25, 26 and 27 respectively. The effects of secondary flows are illustrated by the differences between figures 16 through 19 and 24 through 27.

From inspection of the above cycle 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 expander blades, or ceramic expander blades with less cooling than the metal ones) and second, the optimum-pressure ratio pr for the selected value of T'.

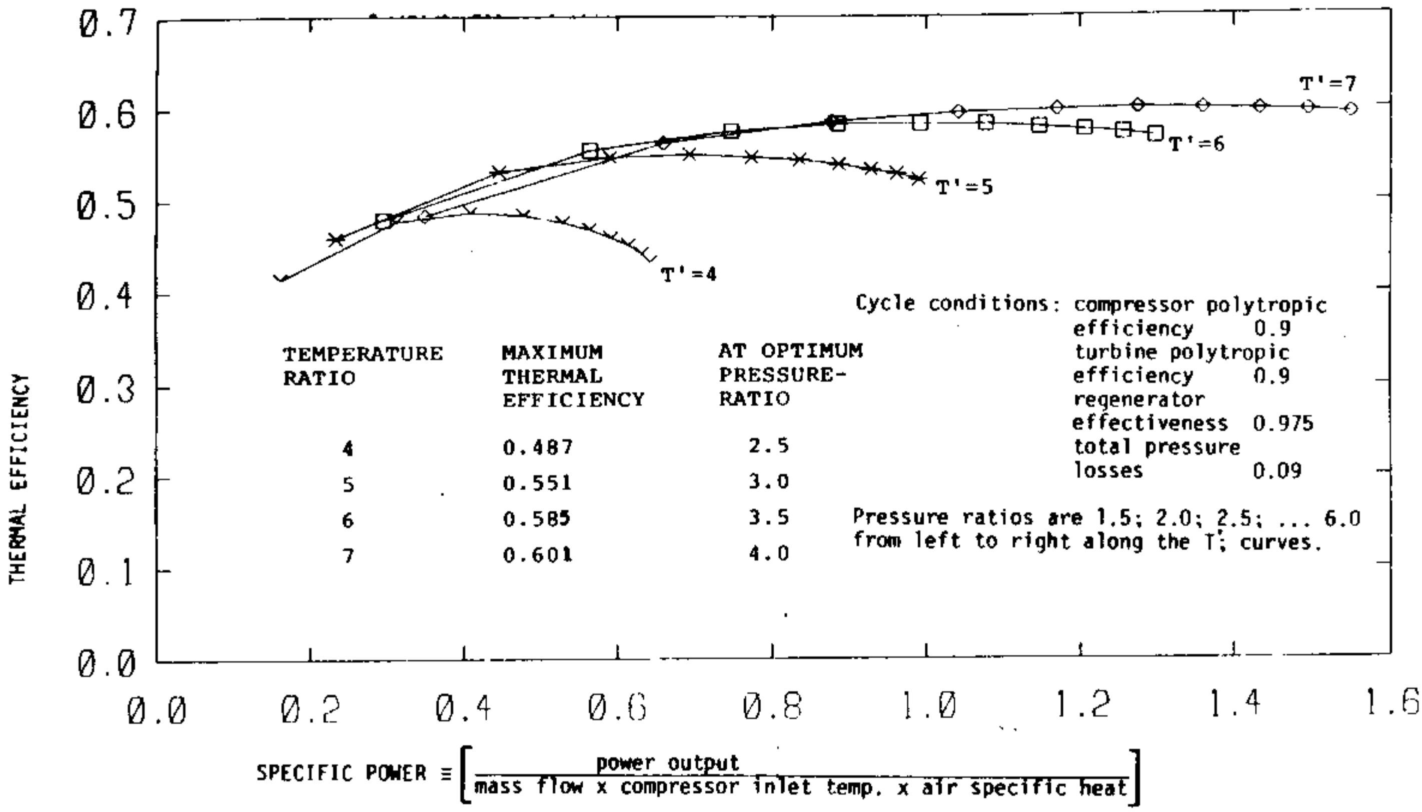


FIGURE 24. CALCULATED PERFORMANCE OF THE CBEX LPR CYCLE WITH HIGH COOLING-FLOW FRACTION.

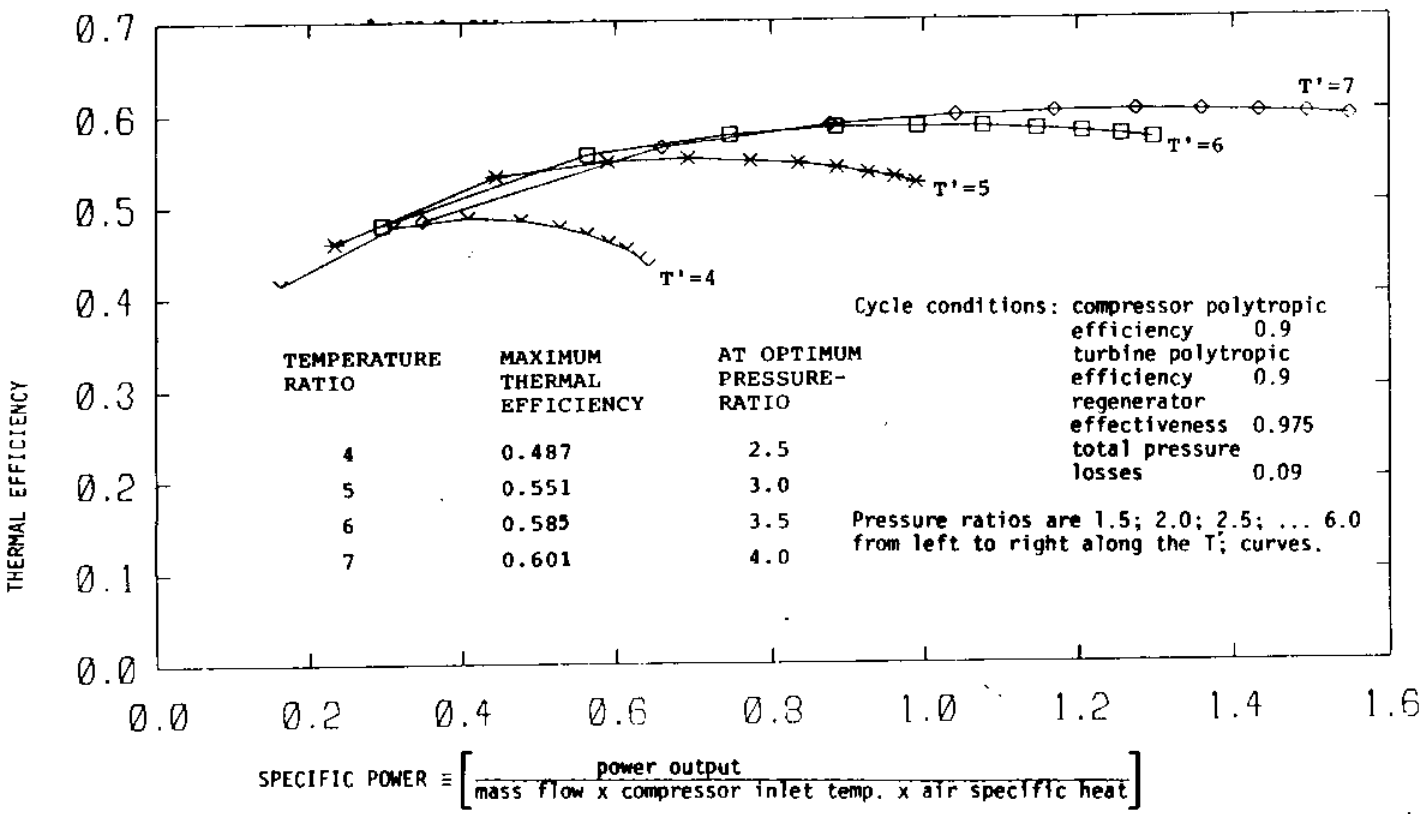
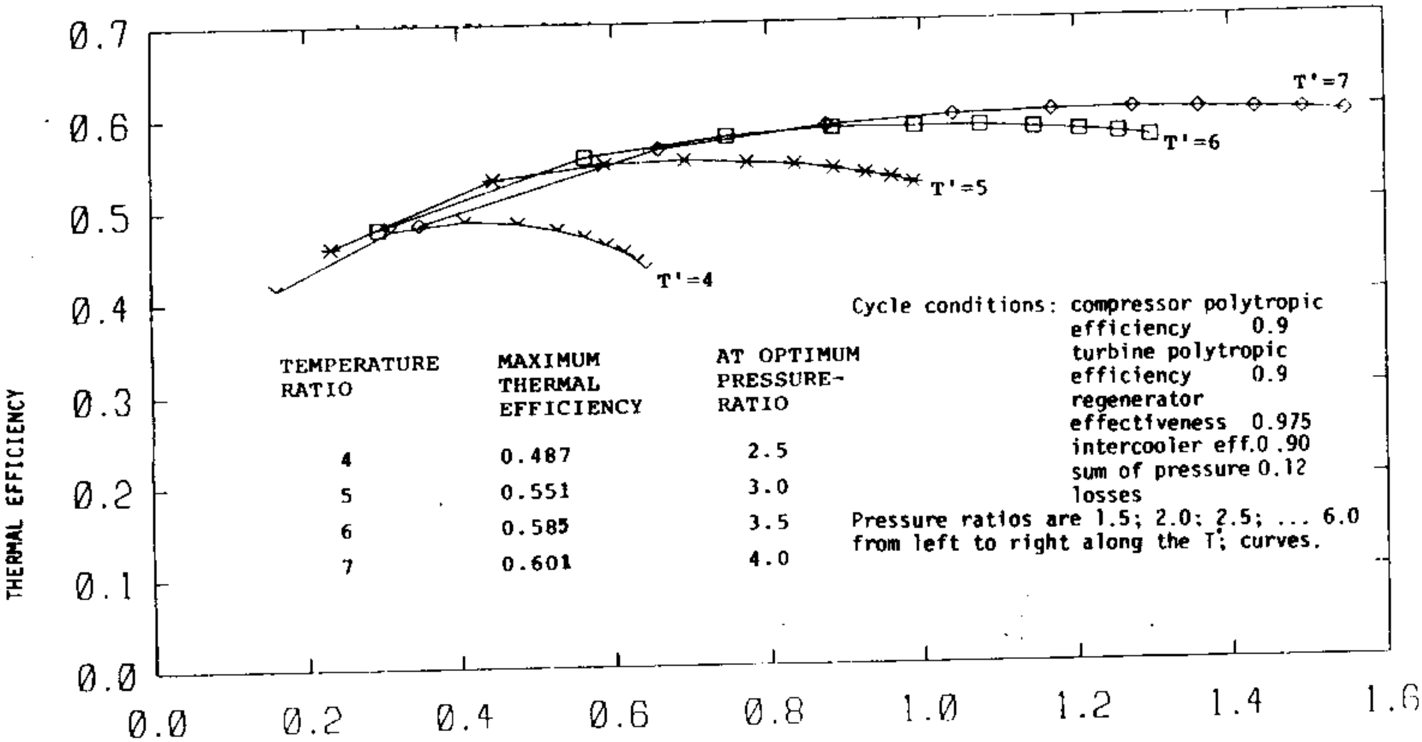
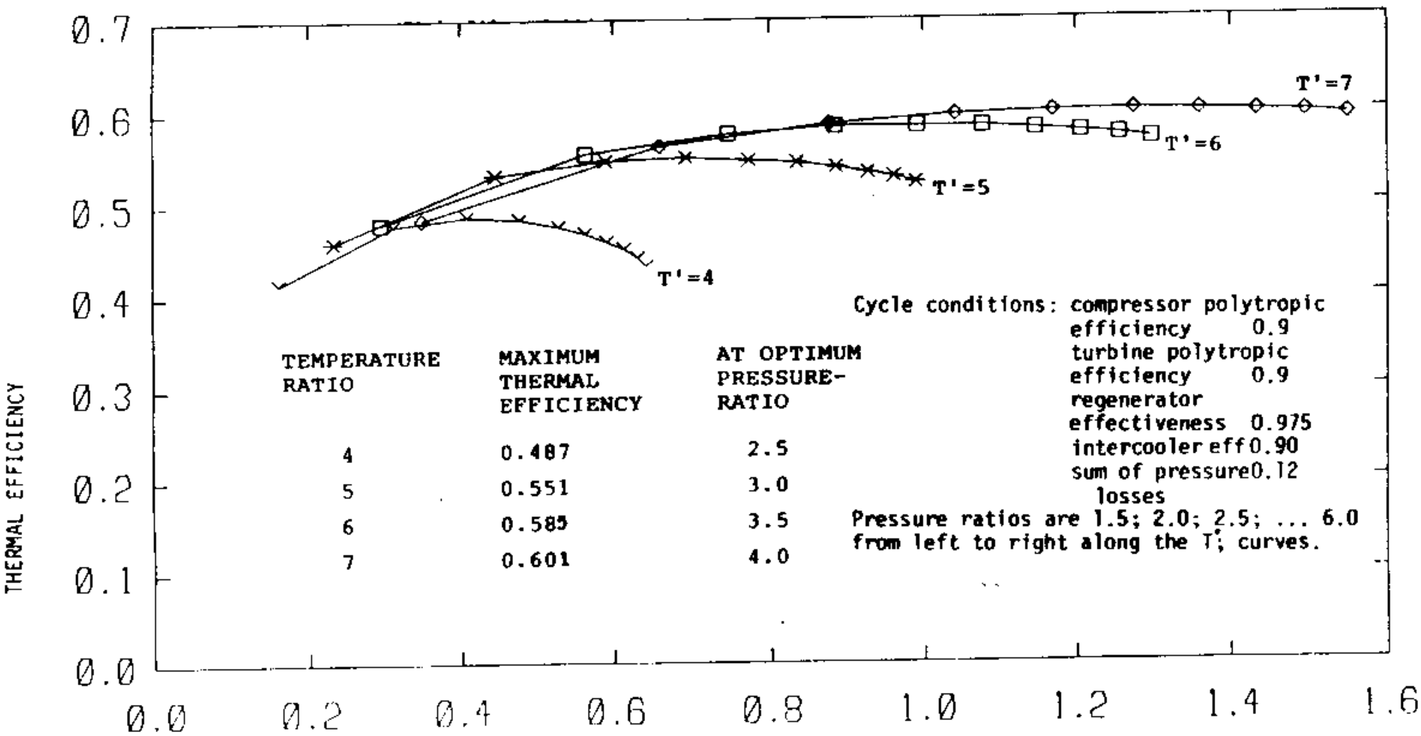


FIGURE 25. CALCULATED PERFORMANCE OF THE CBEX LPR CYCLE WITH LOW COOLING-FLOW FRACTION.



SPECIFIC POWER $\equiv \left[\dfrac{\text{power output}}{\text{mass flow x compressor inlet temp. x air specific heat}} \right]$

FIGURE 26. CALCULATED PERFORMANCE OF THE CICBEX CYCLE WITH HIGH COOLING-FLOW FRACTION.



SPECIFIC POWER $\equiv \left[\dfrac{\text{power output}}{\text{mass flow x compressor inlet temp. x air specific heat}} \right]$

FIGURE 27. CALCULATED PERFORMANCE OF THE CICBEX CYCLE WITH LOW COOLING-FLOW FRACTION.

As shown in the performance plots of figures 24, 25, 26 and 27, for T' = 5.4 and for the conditions of table 1 the optimum pressure ratio for maximum efficiency is:

about 3:1 for the CBEX cycle; and
about 4:1 for (each compressor of) the CICBEX cycle.

The CICBEX cycle shows higher thermal efficiencies than the CBEX cycle. However, this is at higher pressure ratios that may decrease the viability of the rotary regenerator. Additionally, an engine operating on the CICBEX cycle requires a more complex machinery arrangement (two or three shafts) and would be more expensive than an engine operating on the CBEX cycle because of the intercooler. At this point we decided to aim for a heat-exchanger effectiveness of 0.975 and the resulting lower optimum pressure ratio (about 3:1). An additional unquantified advantage of the CBEX over the CICBEX cycle is that the lower pressure ratios allow for higher compressor efficiencies (as shown in figure 6).

A word of caution is due here. The modified computer code, used for the production of the cycle performance plots, inherently assumes that the cooling-air-flow fraction (as derived from figure 9) and the regenerator-mass-leakage fraction (as derived from figure 23) are re-admitted to the cycle at engine exhaust. Although this assumption was the one that allowed modification of the computer code (rather than re-writing the code), it is not the most conservative one. As illustrated in Appendix A of reference 12 the most conservative results for thermal-efficiency values are obtained by reading the results of figures 24, 25, 26 and 27 for a lower value of T'. This lower T' value should be obtained from the turbine inlet temperature that results from mixing the main-stream flow and the turbine-cooling air-flow fraction (by energy balance). However, we believe that our approach is sufficiently conservative for credibility.

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

CBEX cycle;
compressor pressure ratio pr of about 3:1;
single-shaft configuration;
regenerator effectiveness of 0.975;
combustor-exit temperature of 1555 K; and
rated power 1119 kW (1500 hp).

As shown in appendix A of reference 12, for the above conditions of the LPR engine and for metallic expander blading, the modified value of T' is about 5.27. The expected thermal efficiency of the LPR engine at design point is about 55%.

COMPRESSOR PRELIMINARY DESIGN

The preliminary design of turbines and compressors is customarily accomplished by choosing what are known as the "vector diagrams" at the mean diameters of the stages. This choice also includes choosing the blade speeds and enthalpy changes per stage, and therefore determining the number of stages.

We have already chosen the design-point pressure ratio of the compressor at 3:1. This low value gives the designer a great deal of freedom. A single centrifugal compressor stage could be used, but its peak polytropic efficiency would be about 0.87, which would about double the losses to be expected from the best axial-flow compressor, and would probably be even less favorable in off-design conditions.

We could achieve a pressure ratio of 3:1 in about four high-speed axial stages, but Mach numbers and aerodynamic loading would be high, and the design and off-design efficiency would be less than could be achieved with lighter loading.

Accordingly we chose in addition to investigate some more-conservative designs, with wholly subsonic Mach numbers and very conservative aerodynamic loading (to produce a wide range of high efficiency and a tolerance to factors such as blade erosion, dirt or salt deposition or foreign-object distortion). As a control, we first calculated a more-conventional design using a mean-diameter vector diagram having 50% reaction and a mean blade speed of 350 m/s. This would almost certainly result in a transonic first stage (the flow would be slightly supersonic relative to the rotor blades at the tip); only four stages would be needed. This compressor was named "K1" by Korakianitis (ref. 12).

Compressor K2 (ref. 12) was a subsonic design, also of 50% reaction at mean diameter, but with a mean blade speed of 275 m/s, and therefore wholly subsonic. Six stages were found to be necessary to achieve a pressure ratio of 3:1.

The third compressor (K3 in ref. 12) used 100% reaction and the very low blade speed of 200 m/s, and six stages were again necessary.

Korakianitis used several common assumptions to compute the characteristics of these three compressors (in reference 12). With these assumptions, the two compressors with 50% reaction showed peak polytropic efficiencies of over 92%, whereas the 100%-reaction compressor was one point lower. On the other hand, the part-load performance of the 100%-reaction compressor was slightly better than that of the others. We chose to specify this high-reaction compressor not only because of the possibly improved part-load performance, but because we felt that the advantages of low blade speed were large, and the differences between the design-point efficiencies were probably not significant in view of the many assumptions that the calculation of these differences relied upon. The computed characteristics of the K3 compressor are shown in figure 28.

The low blade speed reduces foreign-object damage, 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 stresses open up another attractive area of design freedom: the possibility of using reinforced polymer resins, possibly in low-cost molding. Three materials identified in ref. 43 as having outstanding high-temperature fatigue and creep resistance are polyphenylene sulfide (PPS), polyetheretherketone (PEEK), and polyethersulfone (PES), reinforced with glass, Kevlar or carbon fibers. They should be excellent in a marine environment.

The preliminary design of an engine to give 1500 hp (1.12 MW) shows that an axial compressor with design-point pressure ratio of about 3:1 with a mass flow of 5.5 kg/s would have six stages with 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.

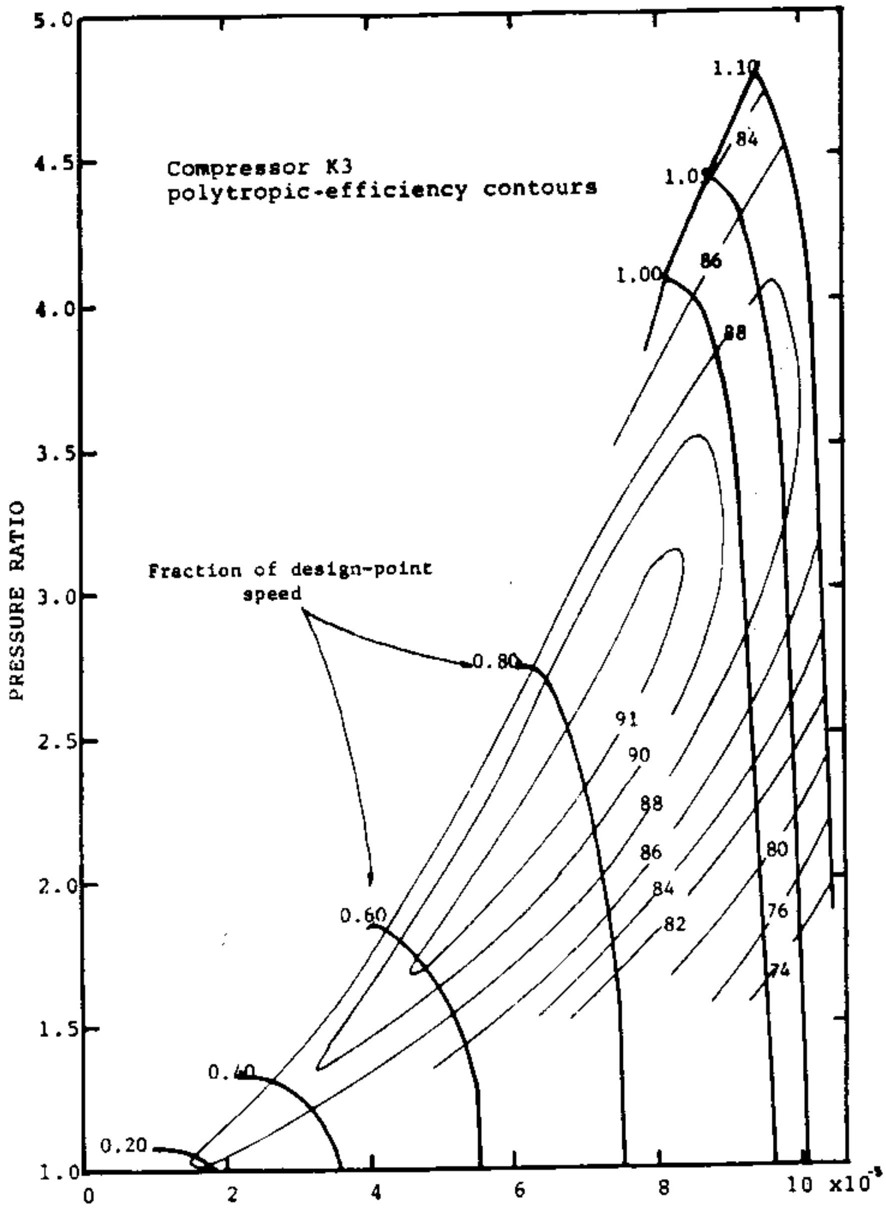
TURBINE PRELIMINARY DESIGN

The axial-flow turbine required for this engine would have three stages (six rows of blades) with an outside diameter of about 450 mm (17.7 in). The shaft speed would be about 16,700 rev/min, giving very low turbine-blade stresses compared with conventional design, and therefore providing favorable conditions for the application of nonmetallic blades. A two-stage epicyclic reduction would probably be used if the engine is directly coupled to a controllable-reversible-pitch propeller.

For the calculation of design-point and off-design-point performance of the LPR engine we assumed that the expander had the performance map shown in figure 29 (which was taken from page 128 of reference 44), whose design conditions are very similar to the design-point conditions of the LPR engine.

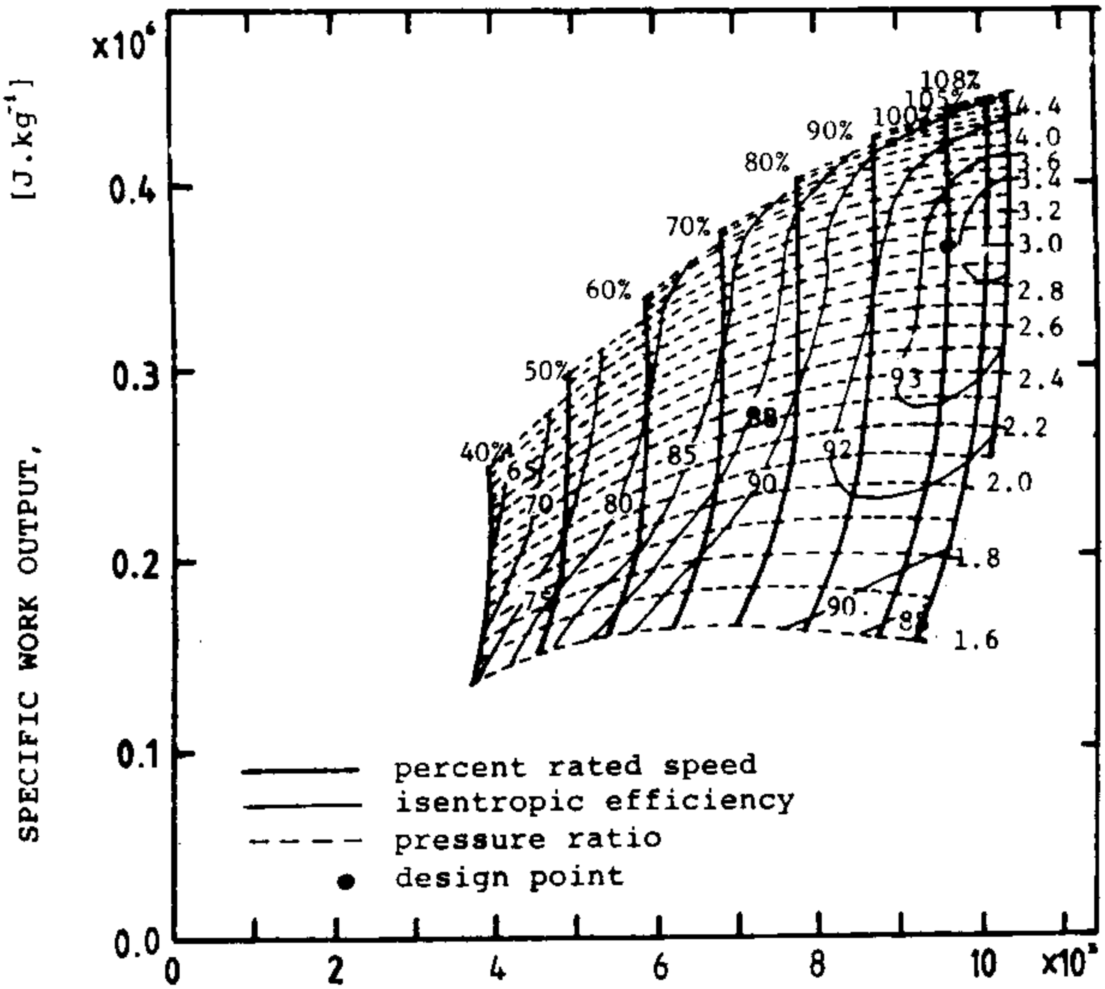
CONCEPTUAL DESIGN OF ROTARY REGENERATORS

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 disks, one on each side of the turbine (figure 5). If this scheme were used for the 1500-hp engine, the disks would be 1.75 m (69 in) diameter and 136 mm (5.4 in) thick. At the present stage of production technology, a disk of this size would be manufactured by building up from smaller sections (ref. 45). Each disk pair would be independently driven by a fractional-horsepower



CORRECTED MASS FLOW ≡
(mass flow) x (inlet temperature) $^{\frac{1}{2}}$ x (inlet pressure)$^{-1}$

FIGURE 28. PREDICTED CHARACTERISTICS OF THE K3 COMPRESSOR WITH CONTOURS OF POLYTROPIC EFFICIENCY



PRODUCT OF MASS FLOW AND ROTATIVE SPEED (kg. rad. s^{-2})

FIGURE 29. TURBINE CHARACTERISTICS WITH CONTOURS OF ISENTROPIC EFFICIENCY.

electric motor through a standard gear reduction and rim drive. The cool exhaust gases would leave the opposite faces of the disks and be ducted up the stack, perhaps giving up further heat to a waste-heat boiler and/or an absorption chiller/freezer on the way. The use of exhaust heat for freezing is discussed in chapter 7.

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, which is described in reference 46. NEPCOMP can be used with or without component characteristic maps and consists of modules that represent engine components (compressor, regenerator, turbine) 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.

NEPCOMP must first be run in the design-point mode, in which it internally satisfies the above conditions. Part of the design-point-mode output is the component-map scale factors which adjust the actual map inputs so that the design-point speeds, pressure ratios, mass flows etc. can be matched. The scale factors become part of the input for the off-design-point mode (and that is why the design-point mode must be run first). In the off-design-point mode the user must specify additional control components that satisfy the conditions of equilibrium mass flow and horsepower balance. Usually this is done by specifying that the interface flow error at entrance to each inlet, compressor, flow splitter, expander and heat exchanger is zero, and satisfying horsepower balance.

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 that we used in NEPCOMP is shown in figure 30. The compressor and turbine performance maps of figure 28 and figure 29 respectively were used (NEPCOMP uses isentropic efficiencies).

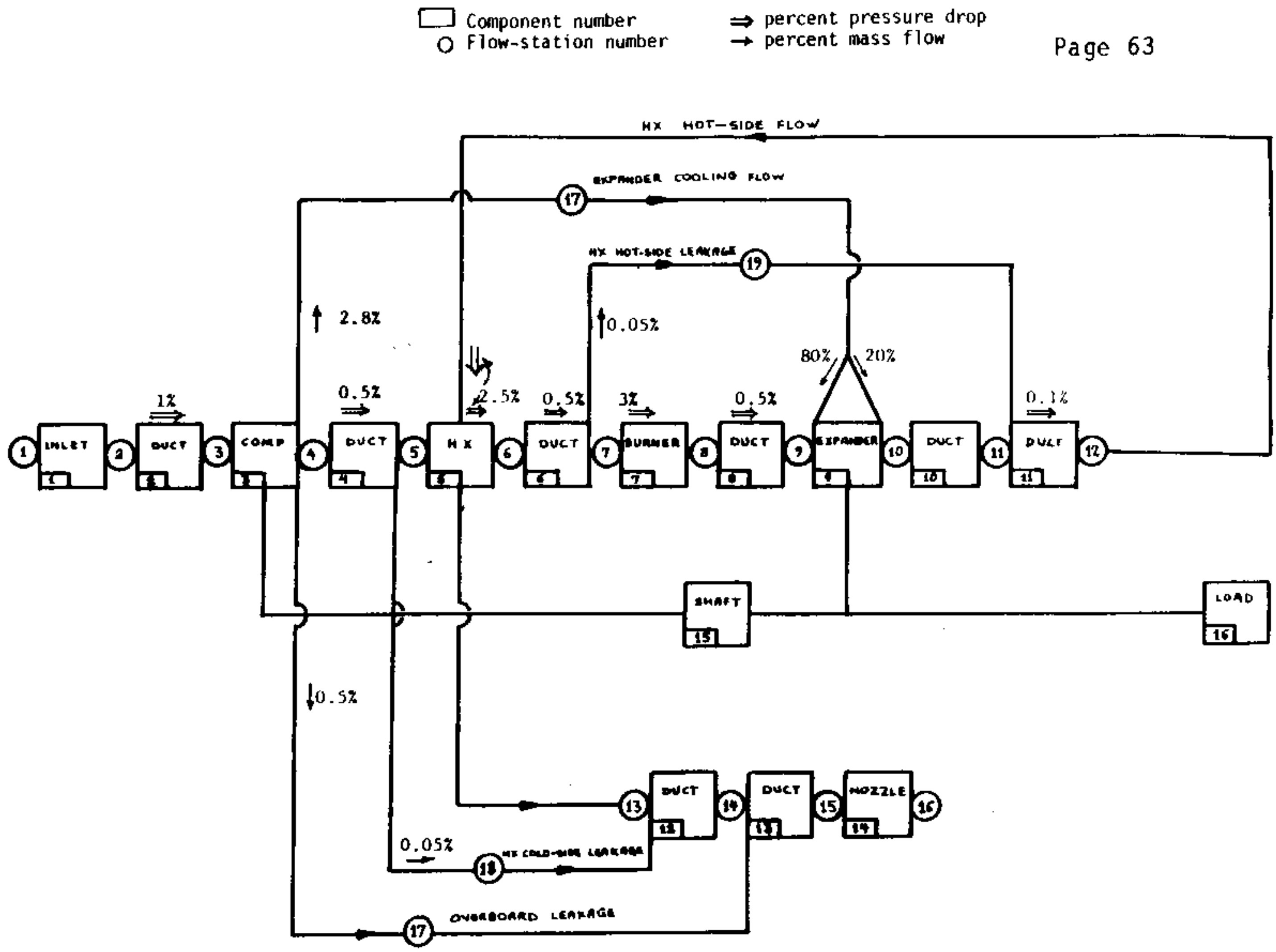


FIGURE 30. MODEL OF LPR ENGINE USED IN NEPII CALCULATIONS.

We specified a regenerator effectiveness of 0.975. Design-point corrected mass flow CMF is defined as:

$$CMF \equiv \frac{\dot{m}\sqrt{\Theta}}{\delta}, \text{ where}$$

$$\Theta \equiv \frac{T_{ic}}{T_{01}}; \text{ and}$$

$$\delta \equiv \frac{p_{ic}}{p_{01}}.$$

In the above m denotes mass-flow rate, T denotes temperature, p denotes pressure, ic denotes total condition at inlet to the component and 01 denotes total condition at inlet to the compressor.

We assumed that the pressure-drop fraction in the regenerator hot and cold passages varies linearly with the fraction of design-point CMF. For the ducts and the combustor we assumed that the relationship followed a square law, as shown in figure 31.

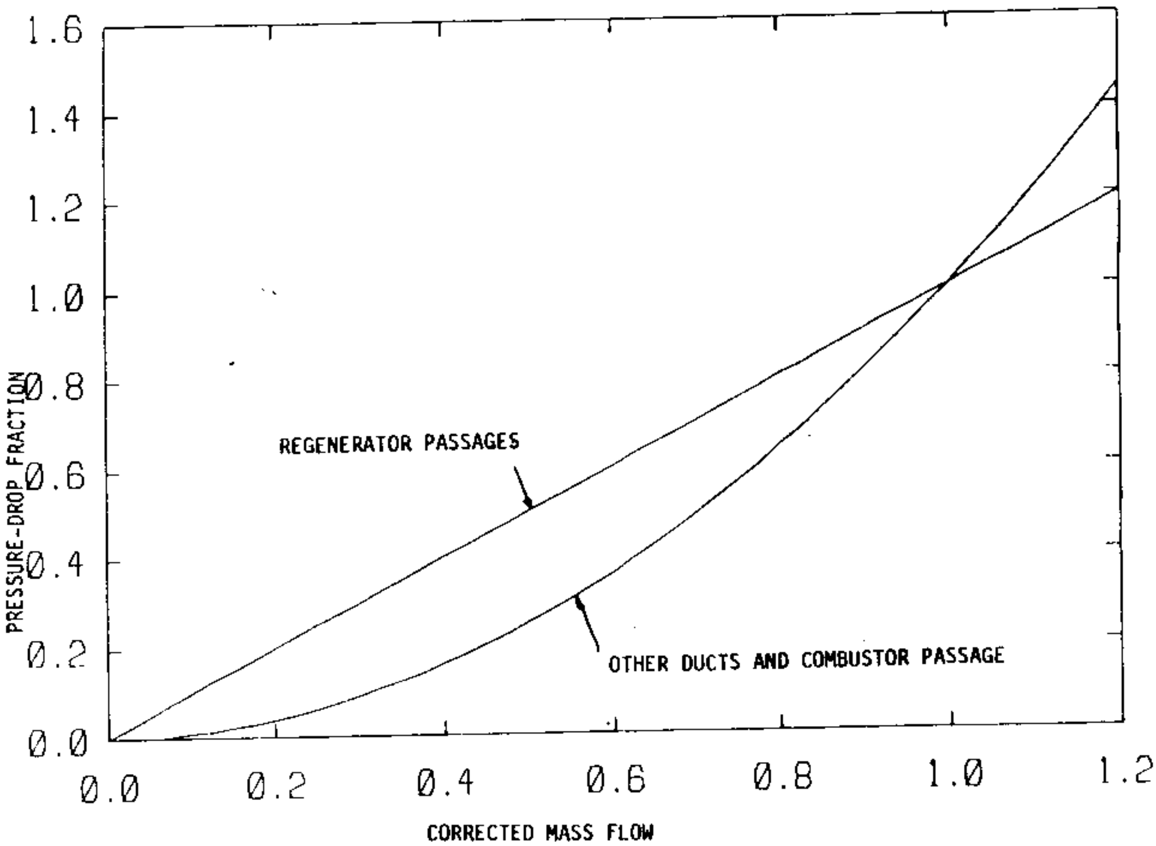


FIGURE 31. ASSUMED VARIATION OF PRESSURE DROPS WITH VARIATIONS IN CORRECTED MASS FLOW.

In reference 12 the output of NEPCOMP was translated into a series of normalized plots that illustrate the performance of the LPR engine. Here we show T' contours in figure 32 and thermal-efficiency contours in figure 33.

As shown in figure 33, the predicted thermal efficiency of the LPR engine is higher than the thermal efficiency of all prime movers today.

As mentioned before, the performance of Brayton-cycle engines is greatly affected by leakage flows and cooling flows. To quantify the effect of these flows on the LPR engine performance we ran NEPCOMP for various values of cooling flows and leakage flows. The results, (from ref. 12), normalized about the LPR-engine design-point values, are shown in figure 34. (The normalizing values are thermal efficiency of 0.55, which corresponds to a cooling-flow fraction of 0.03325 from the compressor; and leakage-flow fractions of 0.0005 each from the hot and

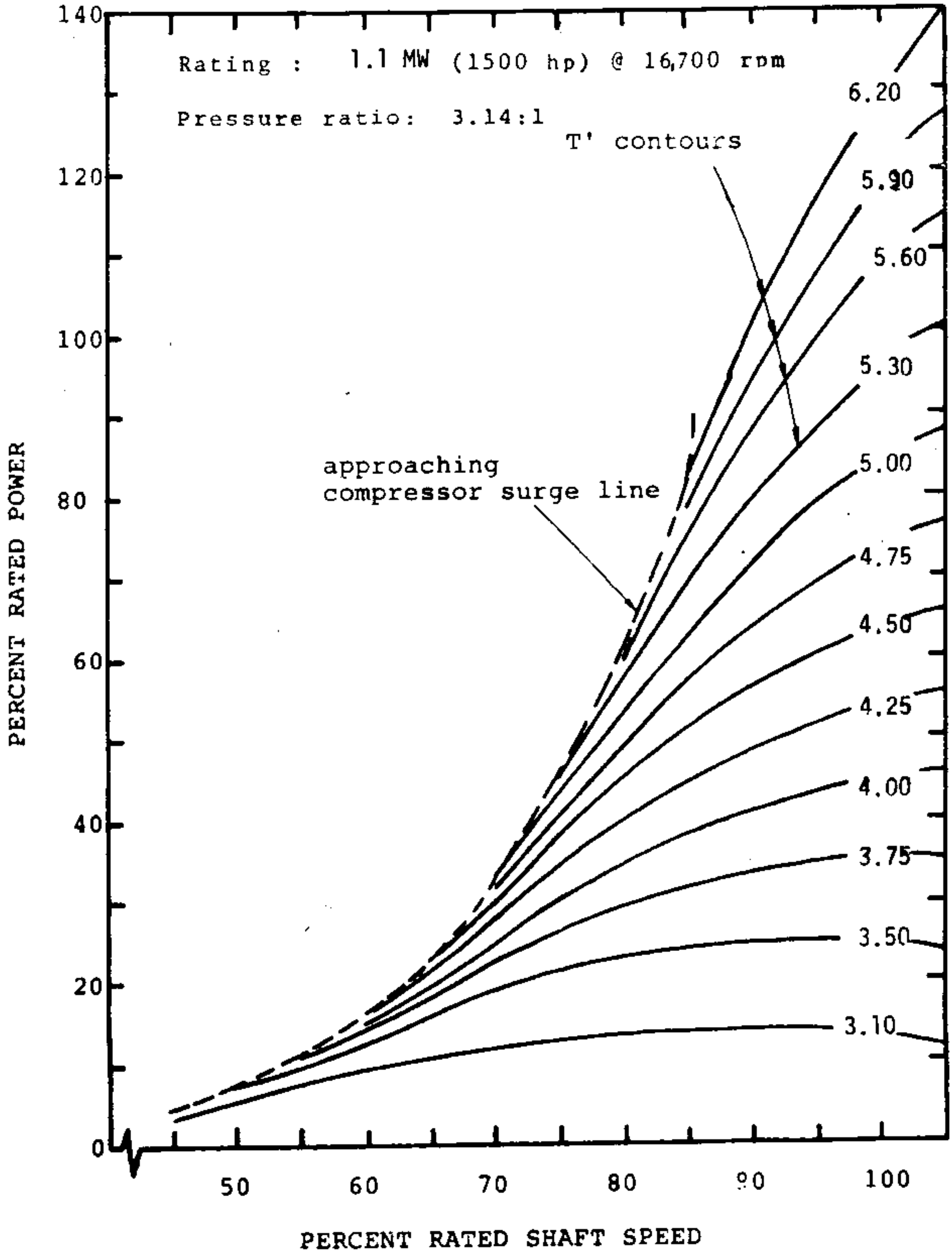


FIGURE 32. TURBINE-INLET TEMPERATURE (EXPRESSED AS T') AS FUNCTION OF PERCENT POWER AND SPEED.

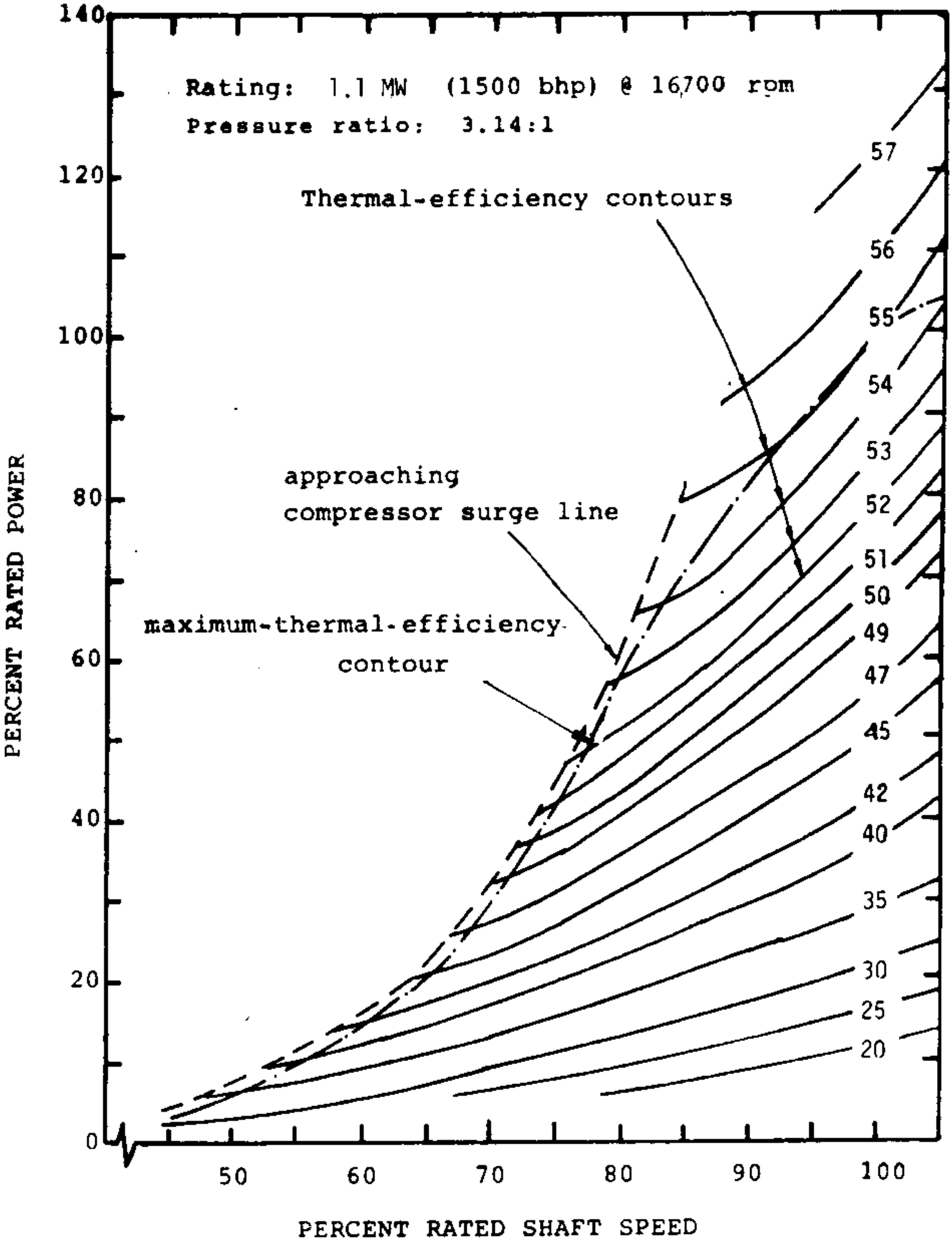


FIGURE 33. THERMAL EFFICIENCY OF THE LPR ENGINE AS FUNCTION OF PERCENT RATED POWER AND SPEED.

the cold side of the heat exchanger, as shown in figure 30).

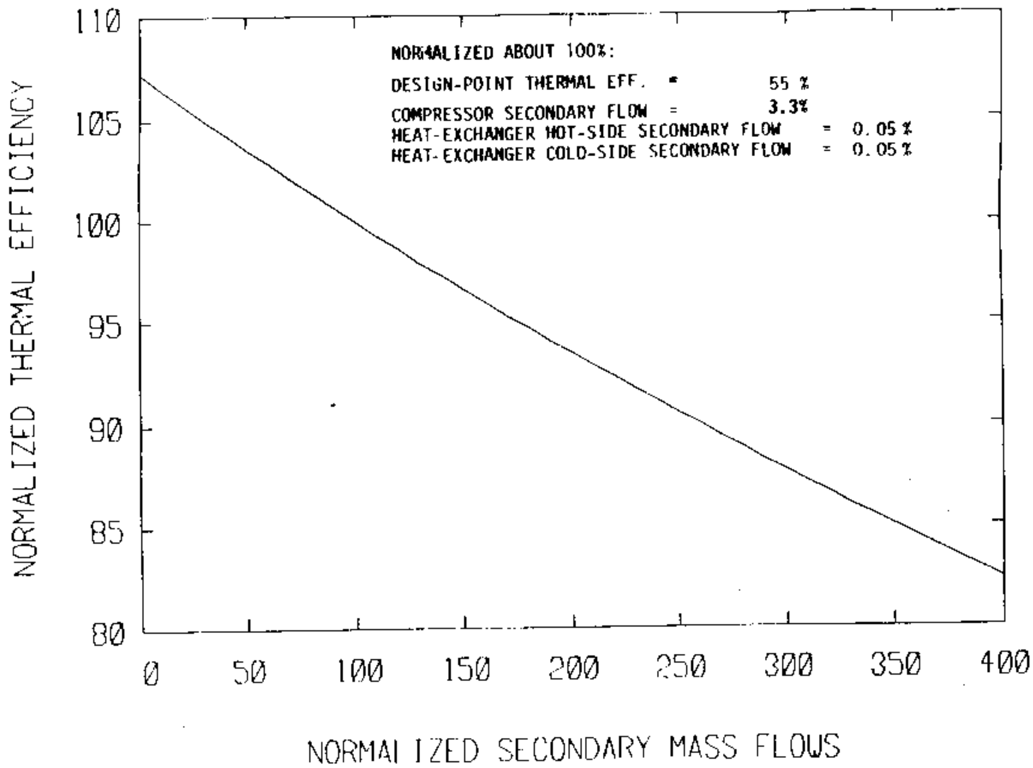


FIGURE 34. CALCULATED EFFECT OF COOLING AND LEAKAGE FLOWS ON THE THERMAL EFFICIENCY OF THE LPR ENGINE.

OVERALL CONFIGURATION

The compressed air leaving the compressor would be ducted to sectors on the disks, 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. The overall arrangement is sketched approximately to scale in figure 35.

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 with the US and Royal Navies (ref. 47) with highly rated aircaft-derivative units, has indeed required exceptionally low maintenance. 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 turbine units make them fairly easy to remove and replace, for instance through the stack.

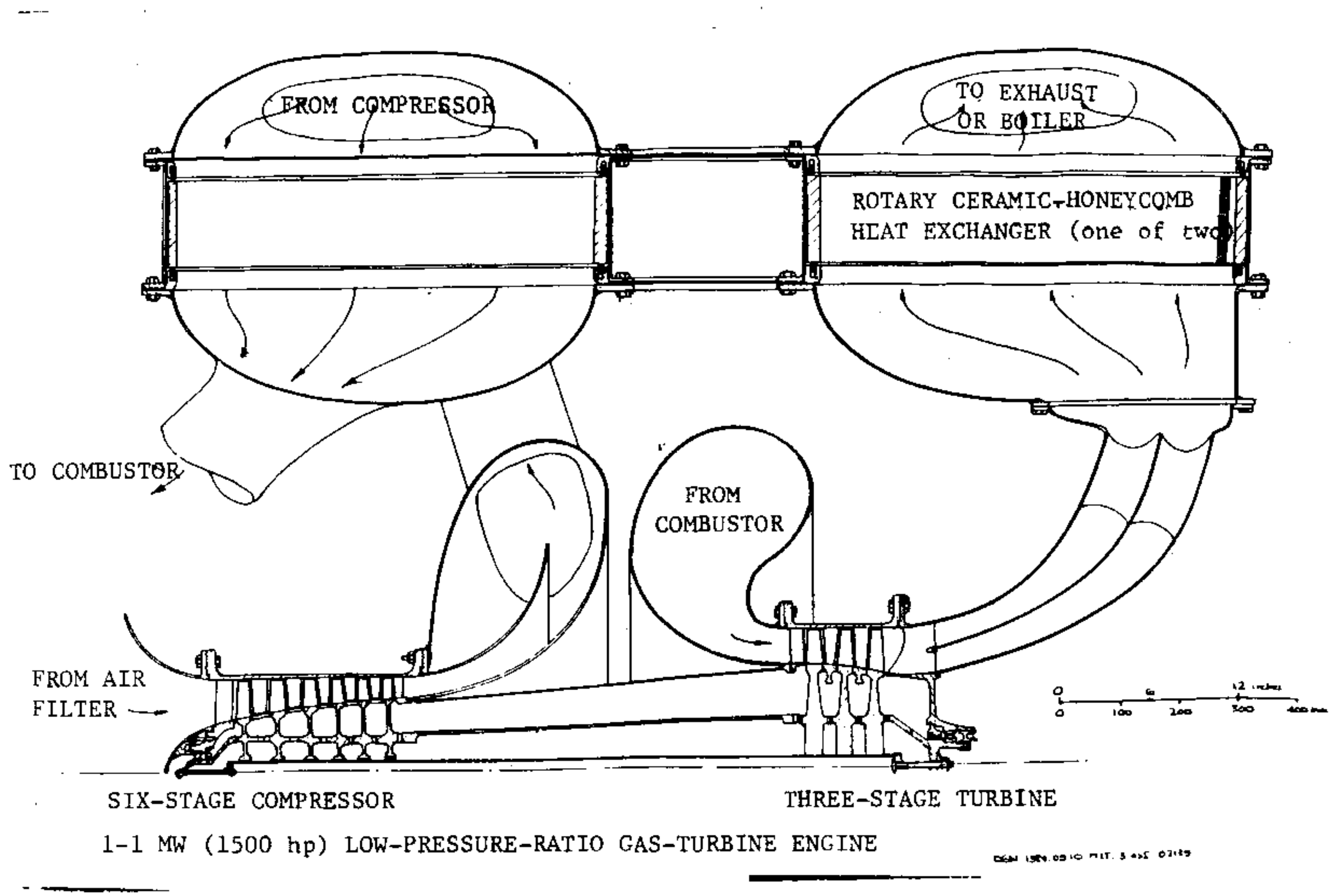


FIGURE 35. CROSS-SECTIONAL SKETCH OF THE LPR GAS-TURBINE ENGINE.

CONCLUSIONS

We have made a case that the low-pressure-ratio highly regenerated gas turbine has particular advantages for marine use. The low blade speeds required would enable nonmetallic materials to be used with advantage, although the virtues of the cycle are not dependent on the use of nonmetals. The design-point fuel consumption should be exceptionally good, and part-load consumption down to at least 25-percent power should be better than that of any 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 nonmetallic materials, particularly in ceramics and ceramic-shielded graphite fibers, would, if initial good reports of the resistance of ceramic coatings to sulfidation attack are further confirmed, make the engine even more attractive for marine use.

A university group cannot do much more than point out advantages and disadvantages of different technologies. We are funded by public money to stimulate change. We hope that some engine manufacturers will study this apparently attractive engine and produce some version of it for marine propulsion and particularly for US and overseas fishing fleets.

CHAPTER 6

CONVERSION OF DECOMMISSIONED MILITARY GAS TURBINES TO HIGH-EFFICIENCY MARINE UNITS

ABSTRACT

We examined the possibility of converting decommissioned helicopter and other military engines to the LPR cycle, as a potential short cut around the large capital costs required to develop a new engine. We believe that some engines are amenable to such conversion. Typically the high-pressure compressor and high-pressure turbine would be removed, and shaft and ducting modifications would be made to introduce high-effectiveness rotary ceramic-honeycomb regenerators. In one case examined the low-pressure turbine could be used with slight modification; in others the turbine stages would have to be rebladed and new shrouds manufactured. In another case the existing combustor could be used with little modification; in others, new combustors would be required. Despite the extent of the modifications, the resulting high-efficiency performance over a large part of the power range, and the presumably relatively low capital and development costs, could make conversion an attractive concept.

BACKGROUND

While we believe that the LPR engine should have relatively low development costs, because a large component of gas-turbine-engine design and development effort seems always to be devoted to the realization of an efficient high-pressure compressor and its associated starting and part-load problems, all of which would be absent in a machine of 3:1 pressure ratio, the development of any new engine is an extremely expensive undertaking. No manufacturer would embark upon this task with a market as small as that represented by the US, or even the international, fishing fleet as its only prospect. It is likely, of course, that if or when engines of the type proposed are developed, they will find applications in many areas besides that of fishing-vessel propulsion.

The present study is of a possible short cut to arrive at an LPR engine. Our reasoning is as follows. Virtually all existing gas-turbine engines have pressure ratios higher, sometimes considerably higher, than 3:1. Axial-flow compressors have low-pressure stages that are designed in almost exactly the same manner whether they are part of a high- or a low-pressure-ratio unit. The only compromise is likely to be that "early" stages of high-pressure-ratio compressors should be designed to be resistant to rotating and other stall phenomena, and may sacrifice a point of efficiency on this account (ref. 1). But they would thereby be more reliable and more tolerant of distortions in inlet velocities, of dirt or salt buildup on airfoils, and so forth, and be fully acceptable if they could be incorporated into a marine engine. Our first consideration was, therefore, that we could use the low-pressure stages of the axial-flow compressors of some existing engines to serve as the entire compressor of an LPR engine. It seemed possible also, though less likely, that the low-pressure compressor of an engine with a two-stage centrifugal compressor might be usable.

Our second consideration was that there are certification requirements that result in many engines, particularly those from helicopters and other aircraft, being decommissioned in a condition in which a long additional life in either a de-rated or a rebuilt form could be confidently expected.

Given these two considerations - that decommissioned engines are often available at little above salvage cost, and that in some cases the low-pressure compressor stages could be used as a starting point for an LPR engine - the purpose of the present study was to determine if the large amount of additional modifications required were worth the benefits obtained.

Our preliminary conclusions are mixed. We can be fairly certain that some engines present difficulties too great to make conversion worth while. On the other hand, we found one or two engines that seemed to promise simpler conversion possibilities and the potential of considerable cost savings. Whether or not these more optimistic findings can be substantiated will require more detailed study. They seemed to be of sufficient interest to present them here.

Previous work

One previous study of the conversion of gas-turbine engines has been reported: that by Bowen and Groghan (ref. 48). The type of conversion considered was similar to that proposed here: the use of, typically, a jet engine at a lower-than-design pressure ratio in conjunction with an added heat exchanger. The conclusions were generally favorable. However, the details of the engineering of conversion were not reported. Some rather difficult problems present themselves to the potential engine-conversion engineer. The principal problems encountered in the present study are discussed below. Bowen and Groghan proposed the intercooling of multi-spool engines, which would additionally present the problem of mismatch between compressor spools.

PROBLEM AREAS

Turbine

It is reasonable at first sight to assume that, if we dispense with the high-pressure compressor stages, we could similarly remove the high-pressure turbine stages to produce a flow-area and torque balance.

This simple concept does not work for two reasons.

1. There may be four times as many compressor stages as turbine stages. Removing a finite number of compressor stages may not be balanced by the removal of a finite number of turbine stages.

2. Even if it is possible to remove a finite number of both compressor and turbine stages and thereby to achieve area matching, this can be possible only if the turbine entry temperature is held to the same level in the expansion as was the same turbine stage in the original machine. This temperature will be considerably lower than the original turbine-inlet temperature, and the thermal efficiency of the engine will suffer drastically.

Therefore for these two reasons it is necessary to install new or modified turbines that can operate at least at the original turbine-inlet temperature. If the original first-stage compressor stage is retained, the design mass flow will be close to that of the original engine. If it is attempted to retain the use of an intermediate-pressure turbine disk rebladed with high-temperature blades, these blades will be longer than those replaced (because the mass flow and pressure will be similar to the original, while the temperature will be higher). They will therefore add stress to the disk, perhaps requiring a new disk. A new shroud will also be needed. If the blades are cooled, it is probable that an increased mass flow over that required for the original first-stage blades will be required.

There are also some favorable aspects to the conversion. The gas-to-blade heat-transfer coefficient will be reduced because of the lower gas density and higher viscosity. And any cooling air required will be supplied at a lower pressure than in the original, and therefore at a lower energy cost to the cycle.

Heat exchanger

The essential nature of the proposed conversion is that the energy transfer from the post-combustion phase to the pre-combustion phase, required in all heat-engine cycles, will be effected largely through the heat exchanger, rather than being entirely through the shaft as in a so-called "simple cycle" gas turbine. The cycle pressure ratio decreases, and at the same time the heat-exchanger effectiveness increases.

This process intrinsically eliminates all but the ceramic rotary regenerator from consideration.

The reason is as follows. The maximum turbine-inlet temperature permissible for metallic uncooled turbines is about 980C, 1800F. Air-cooled turbines have turbine-inlet temperatures from 1100-1500C (2000-2750F), depending on size (the larger blades are easier to cool) and state of development (aircraft gas turbines being the most highly advanced) (figure 1). Experimental uncooled ceramic turbines aim at using temperatures in the 1200-1400C (2200-2500F) range. Turbines of low pressure ratio (eg 3:1) have low temperature drops. At any of these turbine-inlet temperatures the outlet temperature of a low-pressure-ratio turbine will be higher than the maximum temperature presently contemplated for a metallic heat exchanger: 760C (1400F). Therefore metal heat exchangers must be ruled out.

On the other hand, ceramic rotary regenerators would seem to be particularly suitable. They have achieved steadily increasing design-point effectivenesses (figure 4). They have been indicted for excessive seal leakage at pressure ratios above 6:1. The present application, with pressure ratios generally below 3:1, promises to incur low seal leakage.

This does not mean that ceramic regenerators will be universally accepted. They have had a varied introduction. One showed its toughness when fitted to an experimental Rover-BRM sports-racing car in a running of the Le Mans 24-hour race. At a point well into the race the engine showed a drop in power. Mechanics in the pits found that a bolt had fallen into a duct and had become embedded in the rotating ceramic matrix. The decision was made to continue the race without undertaking repairs, and the car did in fact complete the 24 hours. Possibly influenced by this successful demonstration, Ford engineers decided to specify ceramic matrices for a new truck turbine. The engine and regenerator passed extensive development with flying colors, and the engine was marketed. Soon thereafter ceramic failures began occurring in the rotary regenerator. The failures were traced to an unsuspected sensitivity to sulfur that was found in only some diesel fuels. The problem was cured by the substitution of another ceramic.

At present we know of only one production gas-turbine engine equipped with ceramic regenerators: the Allison GT404 of about 300 kW (400 hp). The effectiveness is about 0.95, which is in the right range for an LPR cycle. Our fall-back position with regard to heat exchangers was to specify the use of these production ceramic cores, if permission were granted to do so, together with the seals and drive, and to combine as many as would be required in a new housing and drive system. Considerably larger units were made for the Ford truck engine and, if available, would give greater flexibility in designing for conversion.

While rotary ceramic regenerators have not yet been fully accepted as fully developed and reliable components of production gas turbines, research and development work is accelerating. The duty required in an LPR engine should, because of its low pressure ratio, be less demanding than for a higher pressure ratio, but this advantage is counterbalanced somewhat by the higher inlet temperature that results from the low turbine temperature drop. We confidently believe that the substantial advantages given by the rotary ceramic regenerator will assure its full development very soon.

Combustor

In general, a high-pressure-ratio engine when converted to an LPR cycle will require a larger combustor, simply because of the lower density of the gas flow. In one of the cases discussed below, the combustor of a helicopter engine seemed sufficiently oversized to allow for a range of service operation that it could possibly be satisfactory in the LPR cycle.

PRELIMINARY INVESTIGATION OF CANDIDATE ENGINES

AGT 1500 tank engine

A large number of AGT 1500 gas-turbine engines are being manufacturered by Avco Lycoming for the XM1 tank. It is a three-spool engine, the gas generator having two counter-rotating spools (figure 36). The free power turbine delivers 1500 hp at design point. A plate-fin two-pass cross-flow heat exchanger of about 0.68 effectiveness is fitted, and the design-point pressure ratio is 14.5:1.

This seemed to be a good candidate for conversion. The low-pressure compressor is a five-stage axial unit of high efficiency, and a pressure ratio of 3.3. A preliminary design study of a new heat exchanger showed that twin rotary regenerators of 0.95 thermal effectiveness would be slightly more compact than the existing plate-fin heat exchanger of 0.68 effectiveness.

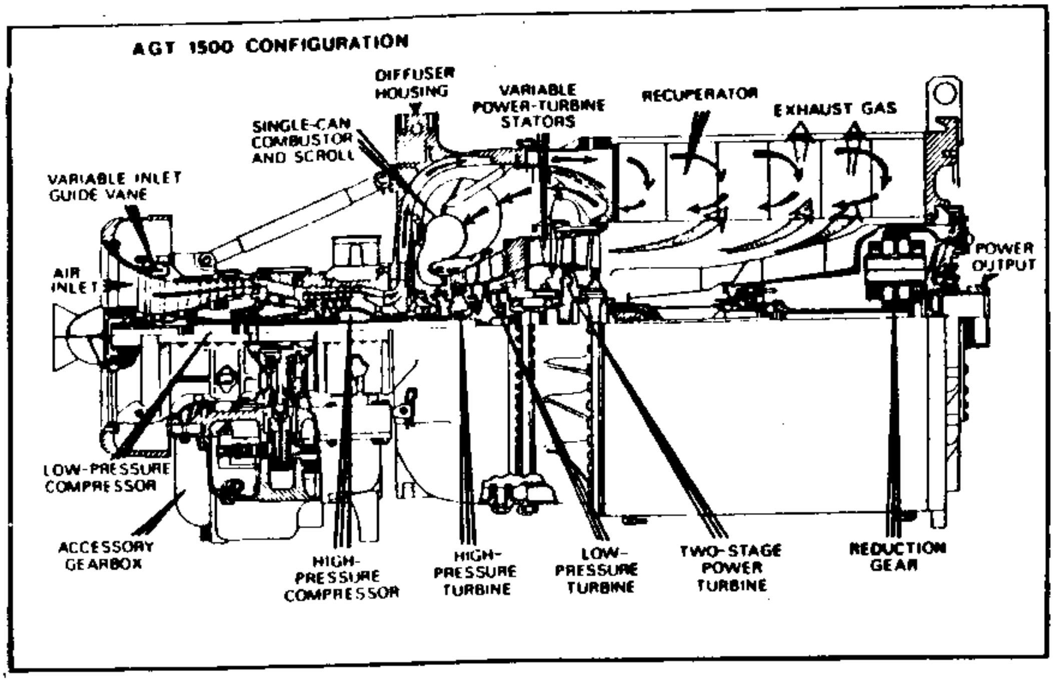


FIGURE 36. AGT-1500 TANK ENGINE: CROSS-SECTION THROUGH ENGINE.

The study, by Baglione. is reported in ref. 8. It was carried out under a self-imposed rule that a new power turbine would not be specified. Various modifications of the low-pressure gasifier turbine were studied, including machined cut-back of the nozzle and rotor blades, and reblading with 50-percent-reaction blades. Analyses of these configurations were carried out with NEPII, the Navy Engine Performance Computer Program, ref. 46.

Under the restrictive conditions of the study, the configuration showing greatest promise was that with the rebladed LP turbine. The cycle results are shown in figure 37. The design-point power level is reduced to about 700 kW (940 hp) and the thermal efficiency is better than that of an equivalent diesel engine for part loads down to about 300 kW (400 hp).

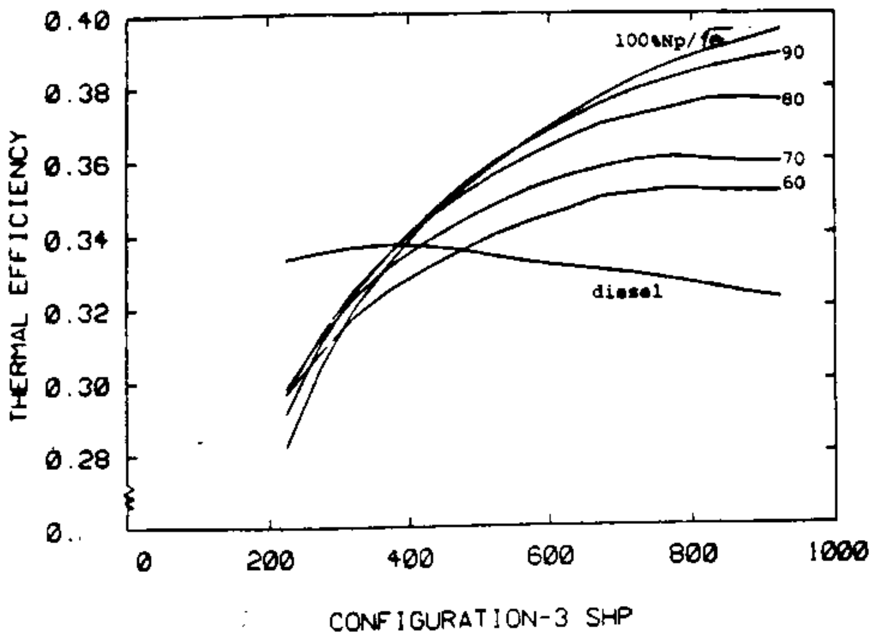


FIGURE 37. CONFIGURATION-3 (REDESIGNED LP TURBINE) PERFORMANCE COMPARISON WITH THE 920-shp CATERPILLAR D398-SERIES DIESEL ENGINE.

General electric T64 engine

The T64 engine is a two-spool turboshaft gas turbine used to power helicopters and turboprop aircraft (figure 38). A single 14-stage axial-flow compressor is driven by a two-stage high-temperature air-cooled turbine. The first five low-pressure compressor stages have variable-angle stators. The power turbine has two stages and is uncooled. The design-point pressure ratio is 14:1, and the power output is 3.7 MW (5000 hp). The relatively low compressor efficiency of 83 percent isentropic led to a higher optimum pressure ratio (4:1) than for other LPR studies being chosen by Schonewald, who performed this study (ref. 11). Details of the mechanical modifications that would be required were studied by Bernard (ref. 14).

The pressure ratio would be given by using eight stages, stages 2 through 9 of the original engine. The first high-pressure-turbine stage was eliminated. The temperature at inlet to the second stage was held to 865C, 1590F, which is its design inlet temperature in normal operation.

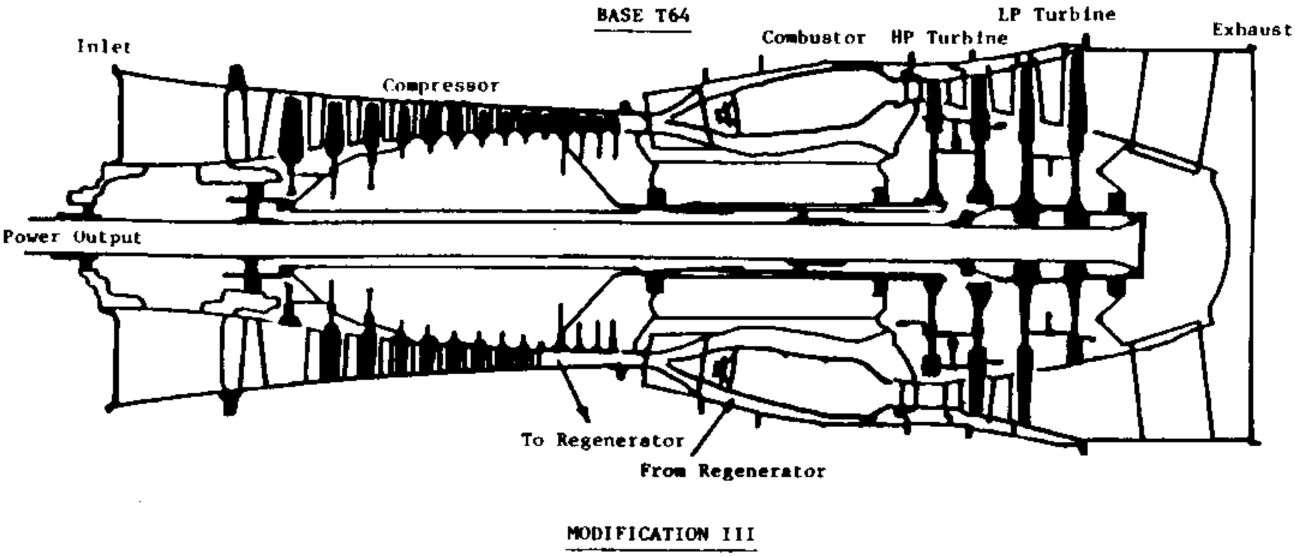
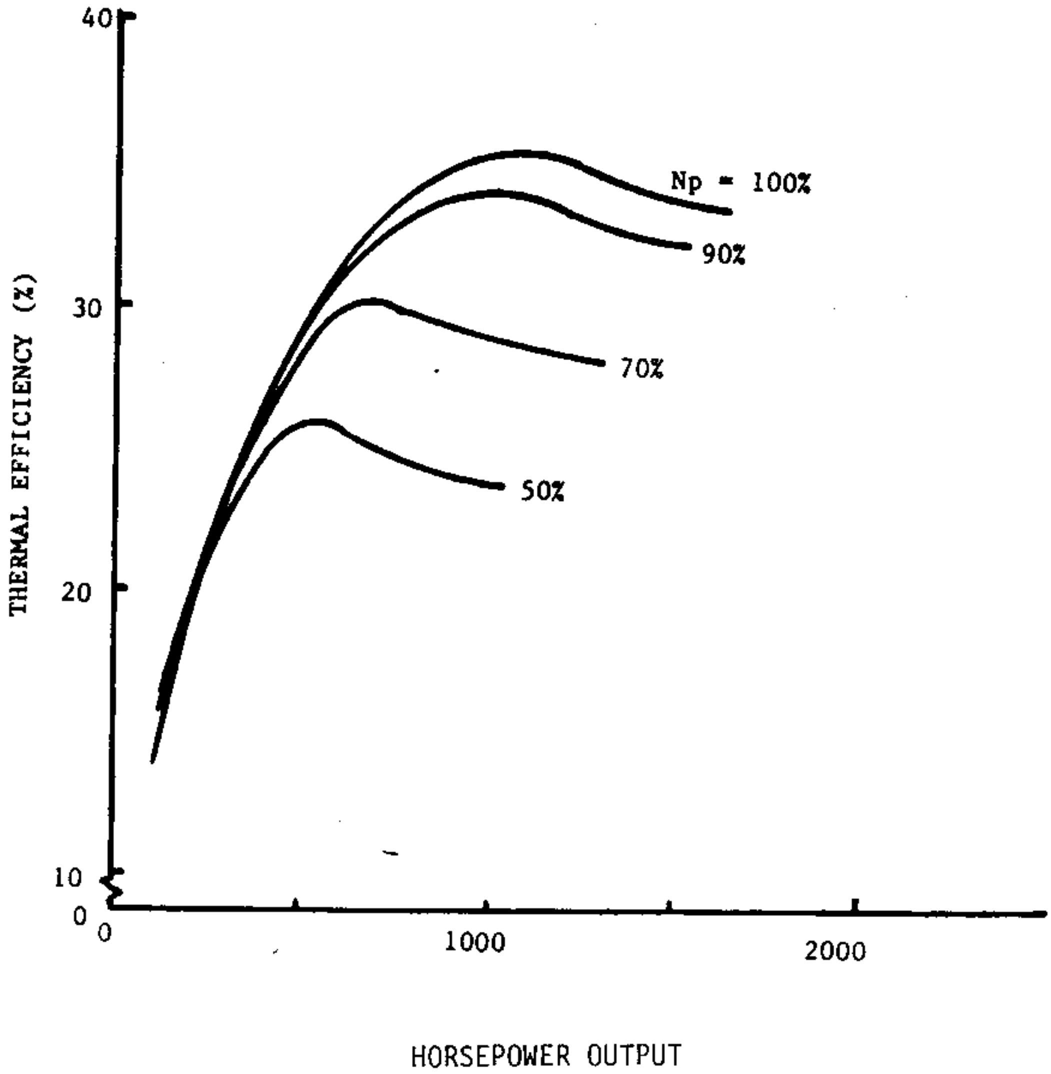


FIGURE 38. T-64 ENGINE CROSS-SECTIONS, ORIGINAL AND MODIFIED.

This severe limitation in peak cycle temperature made the predicted results unattractive (figure 39). The power output was reduced to 1.2 MW (1600 hp). Schonewald's conclusions were that conversion studies should be undertaken only if a high turbine-inlet temperature could be used, through blade cooling or the use of ceramic turbines.

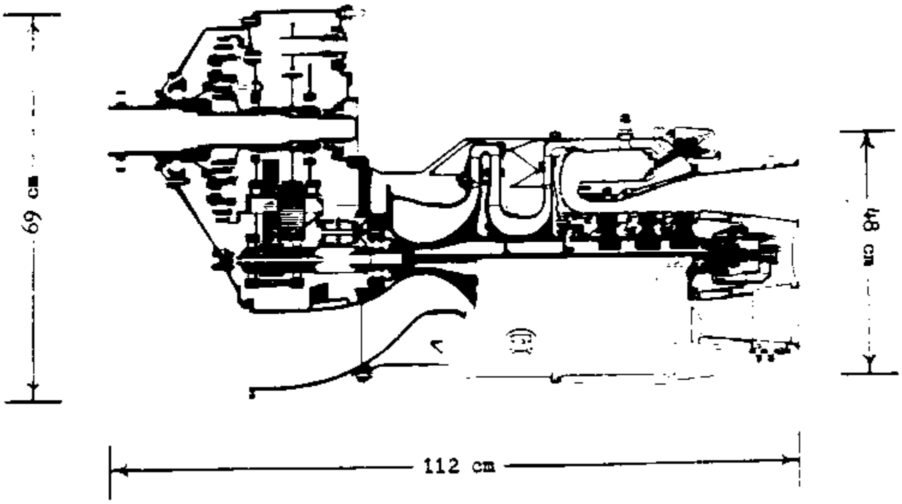
Garrett T76

This is a single-shaft engine with a two-stage centrifugal compressor, reverse-flow combustor, and a cooled three-stage axial-flow turbine (figure 40). The design-point output power is 800 kW (1070 hp). A preliminary study of the possibility of modifying this engine to the LPR cycle was made by King (ref. 6). He looked at five possible configurations. The fifth configuration involved using the first, low-pressure, compressor stage; a rotary ceramic regenerator of 0.95 effectiveness; and a redesigned turbine. The performance of this configuration is shown as line "V" in figure 41.



Lines of constant physical output shaft speed (Np) in percent.

FIGURE 39. MODIFICATION-III THERMAL EFFICIENCY: LOW-PRESSURE-RATIO T64.



The TPE331 is a single-shaft design with a two-stage centrifugal compressor, reverse-flow combustor, and a cooled three-stage axial turbine. At the design point, output power is 800 kW (1070 hp).

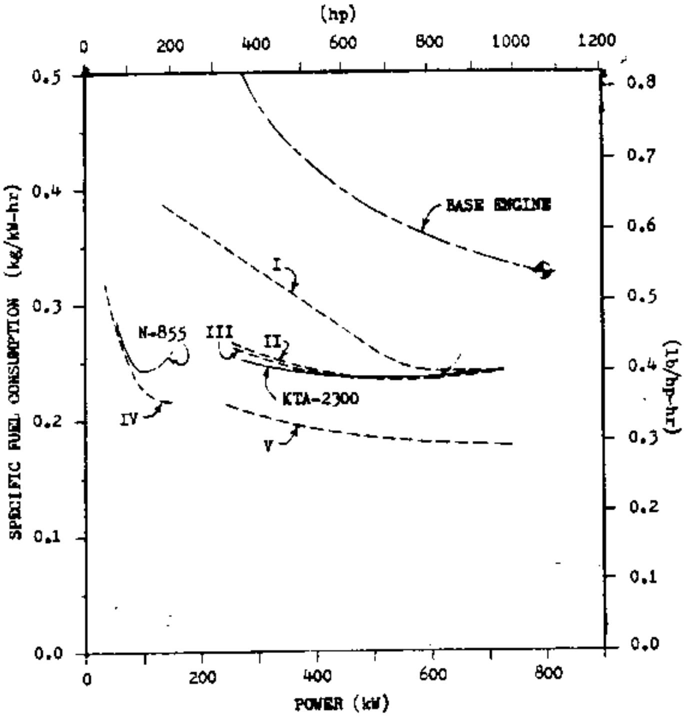
FIGURE 40. GENERAL CONFIGURATION OF BASE ENGINE T76.

It was King's study that stimulated us, and possibly others, to examine the possibilities of conversion in more detail. Poole and Owens undertook a closer study of the T76 engine (ref. 10). Their calculations of cycle thermal efficiency and specific power versus pressure and temperature ratios for the losses to be expected in a conversion of the T76 are plotted in figure 42. The temperature ratio T' for this engine at design point is 4.8 in normal conditions, leading to an expectation of a thermal efficiency near 0.50 if an optimum pressure ratio of a little under 3:1 could be used.

Again, several alternative conversion options were examined. The high-cost option included redesign of the first and second turbine stages, including new coolant-flow channels for the first-stage nozzle and rotor. Medium-cost options include a new first stage with the existing coolant channels, new second-stage nozzles and a modified original-configuration third-stage rotor. The low-cost option also required a new first stage with the existing cooling pattern and modified original-configuration third-stage nozzles and rotors. The high-cost configuration with new first and second turbine stages is predicted to give a power output of 700 kW, 940 hp, at a thermal efficiency of 0.49. The original combustor could be used with only slight modifications. A cross-section of the modified engine is shown in figure 43.

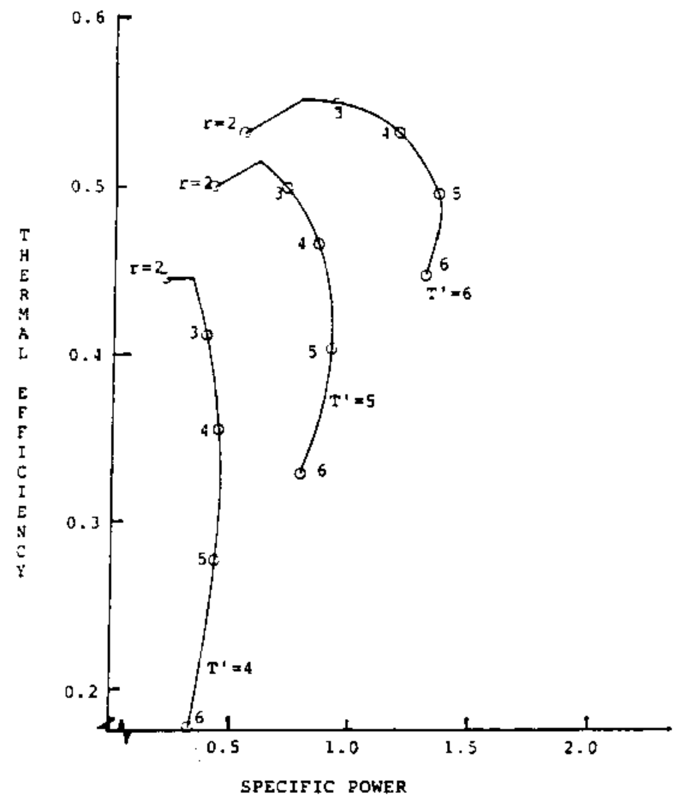
CONCLUSIONS

There appears to be good potential for the conversion of existing higher-pressure-ratio gas turbines to a low-pressure-ratio highly regenerated cycle giving greatly improved thermal efficiencies.



Lines shown for configurations II through V indicate operation at maximum turbine inlet temperature. Information for Cummins N-855 and KTA-2300 marine diesels obtained from company brochure.

FIGURE 41. SUMMARY OF MINIMUM OBTAINABLE SPECIFIC FUEL CONSUMPTION.



$$r \equiv \frac{\text{Compressor-outlet total pressure}}{\text{Compressor-inlet total pressure}}$$

$$T' \equiv \frac{\text{Turbine-inlet total temperature}}{\text{Compressor-inlet total temperature}}$$

FIGURE 42. THERMAL EFFICIENCY VERSUS SPECIFIC POWER FOR A REGENERATIVE CYCLE.

REDESIGNED ENGINE CROSS-SECTION

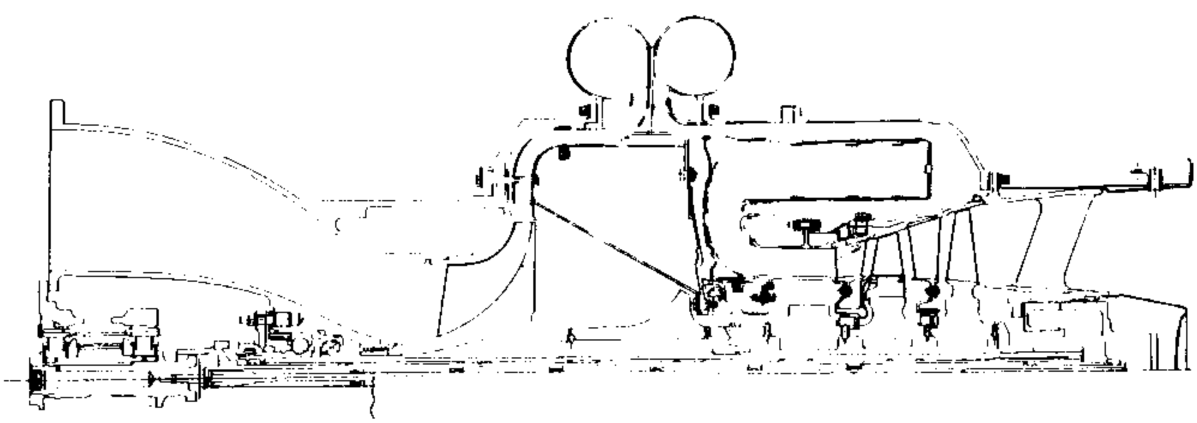


FIGURE 43. CROSS-SECTION OF T76 ENGINE CONVERTED TO LPR CYCLE.

To be a good candidate for such a conversion, an engine should have either an axial compressor or a two-stage centrifugal compressor such that a pressure ratio of about 3:1 could be reached in a modified engine.

Normally the high-pressure stage(s) of the compressor would be discarded, together with the high-pressure stage of the turbine and at least the low-pressure turbine blading and shrouds. New low-pressure turbine blades and shrouds and possibly disks would be required.

Rotary ceramic regenerators would be used. The matrix cores could possibly be those from a production engine, but the housings would necessarily be of the stand-alone type. External ducting would thereby be required, but this could be designed to give lower pressure drops and better flow distributions than those presently penalizing some production engines.

CHAPTER 7

EFFECTS ON SHIP DESIGN OF USING GAS-TURBINE ENGINES

In comparison with diesel engines of the type fitted in fishing vessels, gas turbines of the type discussed here would be considerably smaller and much lighter; they would also be quieter and the exhaust gases would be less noxious. There would be much less vibration transmitted through the vessel. Fuel usage would be less, so that less tankage would be required. The lubricating-oil requirements would be greatly reduced. There would be need only for very small cooling-water intakes and discharges (for the oil coolers for the engine and gear reducers.) The waste heat would be more available because it would all be in the exhaust flow. (In the diesel it is in both the exhaust and the jacket water.)

The scheduled maintenance required is much reduced. If a major component, such as the turbine, compressor, combustor or heat exchanger, requires service, it is usual to remove and replace the entire engine. In naval vessels this is frequently accomplished through the exhaust stack.

A gas-turbine engine of the type discussed here will have more than twice the air and exhaust flow of a diesel of the same power. The turbine engine is also more sensitive to pressure drops in the air and exhaust ducts. Therefore the cross-sectional areas of the inlet and exhaust ducts are likely to be three or four times those of a diesel of the same power. However, when we studied a particular design on the baseline fishing boat (a stern trawler) and discussed our findings with its designer, Mr. J. W. Gilbert, we found that the ducts are still small compared with those for below-decks ventilation air, and that they could be accommodated into available spaces without significant cost or compromise.

STUDY OF ALTERNATIVE ENGINE LOCATIONS

We felt that the aspects of the gas-turbine engine summarized above could lead to more freedom in locating it in a ship, and that there might be advantages in placing it somewhere other than in the traditional amidships area. Several possibilities were studied by Hwang (ref. 13). She first looked at hydrostatic, hydrodynamic and electric transmission systems. All failed their candidacies because of losses of ten percent or greater. In addition, they tended to be heavy and expensive.

Hwang therefore confined her attention to mechanical drives, and looked at two alternatives to the traditional engine location. One was to have the engine at the stern on one side of the trawl way. The confined space and limited draught for a propeller-shaft gear made this location unattractive. In addition the air intake would be in a high-spray region. The second alternative was to install the engine on the 0-1 deck on the pilot-house roof with the shaft vertical and the compressor uppermost. The engine would incorporate a single epicyclic reduction to a shaft that would run vertically down to a final reduction bevel-gear right-angle drive in the conventional engine-room location to the propeller shaft, also in the conventional place. This arrangement seemed to offer the advantages of a very accessible engine; a short intake duct to a filter high up and away from most spray; a short exhaust duct that could connect to the heat exchanger of an absorption freezer; a saving of space for intake and exhaust ducts; a saving of engine-room space; and reduced ship noise, odor and fire danger. However. she concluded that the advantages of this location were probably not great enough to justify such a radical departure from present practices. and that initially at least the gas turbine should be placed in the conventional engine-room position.

POSSIBILITY FOR ABSORPTION CHILLERS

The most efficient internal-combustion engines have thermal efficiencies between 50 and 55 percent. The first law of thermodynamics requires that almost half of the energy input be rejected as heat. Recovery of this wasted energy is often problematic. Only 40 to 50 percent of a diesel engine's waste heat is found in the exhaust. The remainder is low-temperature heat that appears in the cooling water, lubricating oil etc. Additionally, the larger diesel engines that attain the highest levels of thermal efficiency do so with the use of turbochargers that reduce the temperature of the exhaust to levels near the condensation point of sulfuric acid, thus rendering further waste-heat recovery impractical due to the dangers of corrosion. On the other hand a gas-turbine rejects nearly all of its waste heat in the exhaust, which is intrinsically cleaner than the exhaust of diesel engines.

Kellen (ref. 9) studied the following refrigeration cycles that have already been used before in marine applications:

vapor-compression cycle;
Rankine-driven vapor compression;
vapor absorption:
vapor jet;
air cycle; and
thermoelectric cycle.

Having narrowed the choice to between Rankine-driven vapor compression and vapor-absorption cycles, Kellen compared the size, maintenance and reliability, cost and availability. On the basis of its superior volume, maintenance and reliability, and cost and availability, the ammonia-water vapor-absorption system was chosen as the most attractive of the two cycles. Although this refrigeration system is not presently available in standard models, standard designs exist and several U.S. manufacturers are capable of custom-building suitable units. For the baseline fishing boat powered by the LPR engine, such a unit could be directly heated by exhaust gases, and would include a provision for auxiliary oil firing.

CHAPTER 8

EXAMINATION OF VARIOUS COSTS

COMPARISON OF THE LPR ENGINE AND A DIESEL ENGINE

PERFORMANCE COMPARISON

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

$$sfc = 2545 /(\eta_{th}.HVF) \quad \text{(in units of lbm/(shp.hr))}$$

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

Also, sfc [lbm/(shp.hr)] x 0.6082773 = sfc [kg/(kW.hr)]

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

The baseline fishing boat is specified as being powered by the Caterpillar 3516 medium-speed diesel engine, which is rated at 1051 kW (1410 hp) at 1600 rpm. The thermal efficiency of this diesel engine, shown in figure 44, was compiled from data supplied by Mr. J. W. Gilbert and data included in reference 49.

By direct comparison of the values indicated in figures 33, 34 and 36 we conclude that:

with flow fractions of 0.03325 from the compressor and 0.0005 from each side of the heat exchanger, the design-point thermal efficiency of the LPR engine is 55%;

when the leakage-flow fraction and cooling-flow fraction of the LPR engine are increased to 400% of the above flow-fraction figures, which correspond to leakage-flow fraction and cooling-flow fraction of 0.133 from the compressor and 0.002 from each side of the regenerator, the design-point thermal efficiency maintains values in excess of 80% of the above design-point thermal-efficiency value, which corresponds to a design-point thermal-efficiency of about 44%;

even when the leakage-flow fractions and the cooling-flow fractions are increased to higher than 13% of the main flow fraction, the

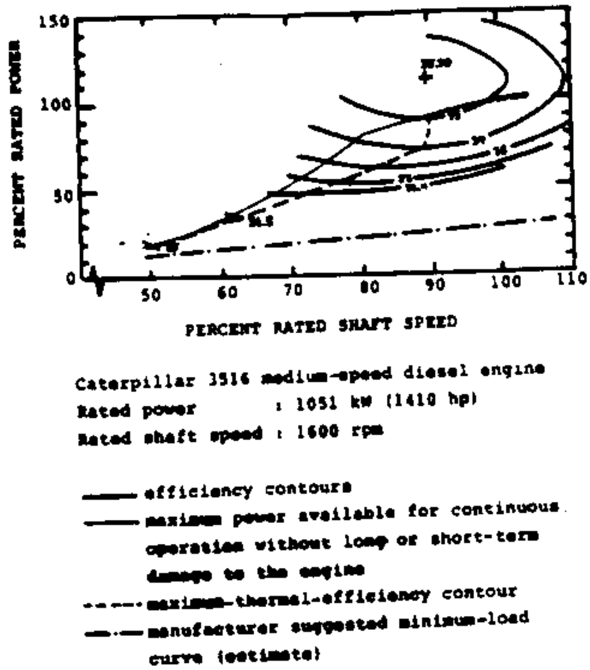


FIGURE 44. THERMAL EFFICIENCY OF THE DIESEL ENGINE AS FUNCTION OF PERCENT RATED POWER AND SPEED.

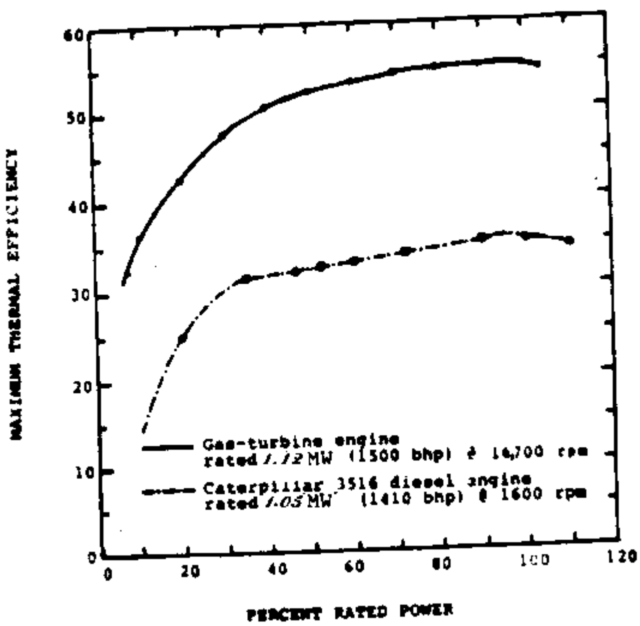


FIGURE 45. PART-LOAD EFFICIENCIES OF GAS-TURBINE AND DIESEL ENGINES.

design-point thermal efficiency of the LPR engine is predicted higher than the design-point thermal efficiency of the medium-speed diesel engine, which is 35.7%.

Comparing the thermal-efficiency contours of the LPR engine and the diesel engine (shown in figures 33 and 36 respectively) the LPR engine is found to be more efficient than the diesel engine. For a given power the thermal efficiency of the LPR engine is highest near the compressor surge line. To avoid compressor surge, we specify the maximum-thermal-efficiency line as shown in figure 33. The maximum-thermal-efficiency contour of the diesel engine is shown in figure 44. Figure 45 compares the maximum attainable thermal efficiency of each engine as a function of percent rated power. The new gas- turbine engine maintains a thermal efficiency advantage over the diesel engine at all powers.

Annual fuel savings themselves will be affected by the particular operating profile of the vessel. To convert the above fuel efficiencies to fuel savings we used data from the baseline fishing boat. Although Mr. J. W. Gilbert generously provided much data, the actual speed-power characteristics of the vessel were not available (partly because once the vessel is delivered, few boat owners are willing to invest the time required for trials). To obtain the speed-power characteristic of the vessel we assumed constant wake and thrust deduction (defined in reference 50), for all vessel speeds, and a typical propeller law (expressed in a cubic horsepower-versus-speed characteristic). This results in a linear relation between vessel speed (in knots) and propeller speed (rpm) for any prime mover (ref. 50, page 93). Although this is a crude approach it is a sufficiently accurate representation for the baseline fishing boat, since the actual characteristics of individual fishing boats vary widely.

The operating profile of chapter 2 for 28 trips, (each of ten-day duration followed by a two-day port call), results in the following annual allocation of time:

2020 hrs, or 23% of the duration, cruising to and from port;
1500 hrs, or 17% of the duration, cruising at fishing grounds;
1800 hrs, or 21% of the duration trawling;
200 hrs, or 2% of the duration hauling nets;
1200 hrs, or 14% of the duration, idling;
1340 hrs, or 15% of the duration, at port; and
700 hrs, or 8% of the duration, at repairs (a value we hope will be conservative for gas-turbine engines).

(Total 8760 hrs-(365 days), 100% of the duration).

FPPs for trawlers are sometimes designed for the cruising condition, sometimes designed for the trawling condition, but are frequently designed for a point in between. The speed-versus-power characteristic of the baseline fishing boat for the last condition is shown in figure 46. The boat operates off the design-point condition most of its life. If the propeller were designed for the trawling condition the cruising line would be below the "manufacturer suggested minimum load line" (shown in figure 44). If the propeller were designed for the cruising condition then the trawling condition would lie outside the operating envelope of the engines.

The only solution to this problem is to design a CPP, which can operate (at any speed and power combination) in the most efficient region of the engine-performance map. A CRPP can also provide for a means of reversing the vessel without reversing the rotation of the main shaft, thus saving some complications in the gearbox arrangements. If a CPP or a CRPP is installed then the fuel and speed controls of the engine would be adjusted to operate along the lines of maximum engine efficiency, shown in figure 47. The vessel-speed lines are not drawn in figure 47 because they would vary with the propeller pitch. The CPP would result in lower fuel consumption but would require higher acquisition and installation costs. A CRPP would be more expensive than a CPP but the resulting gearbox arrangements would be less complicated and less expensive. An economic study would determine the cost advantage or disadvantage of an FPP, a CPP or a CRPP. Typical costs for sizes applicable to the baseline fishing boat are compiled below:

- the overall shafting assembly for a CPP or a CRPP would cost about $ 0.120 million (including shafting, controls, stern-tube and all bearings);
- the single-input/single-output gearbox (including a disconnect clutch) required for a CRPP shafting system would cost about $ 0.100 million;
- the overall shafting assembly for an FPP would cost about $ 0.030 million (including shafting, controls, stern-tube and all bearings); and
- the single-input/single-output gearbox (including a disconnect clutch and a reversing mechanism) required for an FPP or a CPP shafting system would cost about $ 0.130 million.

Because there were not any CPP-performance data available to calculate the annual fuel consumption of the baseline fishing boat, we calculated the annual fuel consumption (based on the above operating profile) with an FPP and with a CPP (or a CRPP) with the following assumption: the CPP would require the same shaft horsepower as the FPP to power the boat at the same vessel speed, but it would deliver this power operating at the respective engine's optimum speed for maximum thermal efficiency. At design point an FPP could have higher open-water efficiency, (defined in reference 50), than a CPP because of the higher loss of circulation around the larger propeller hub that

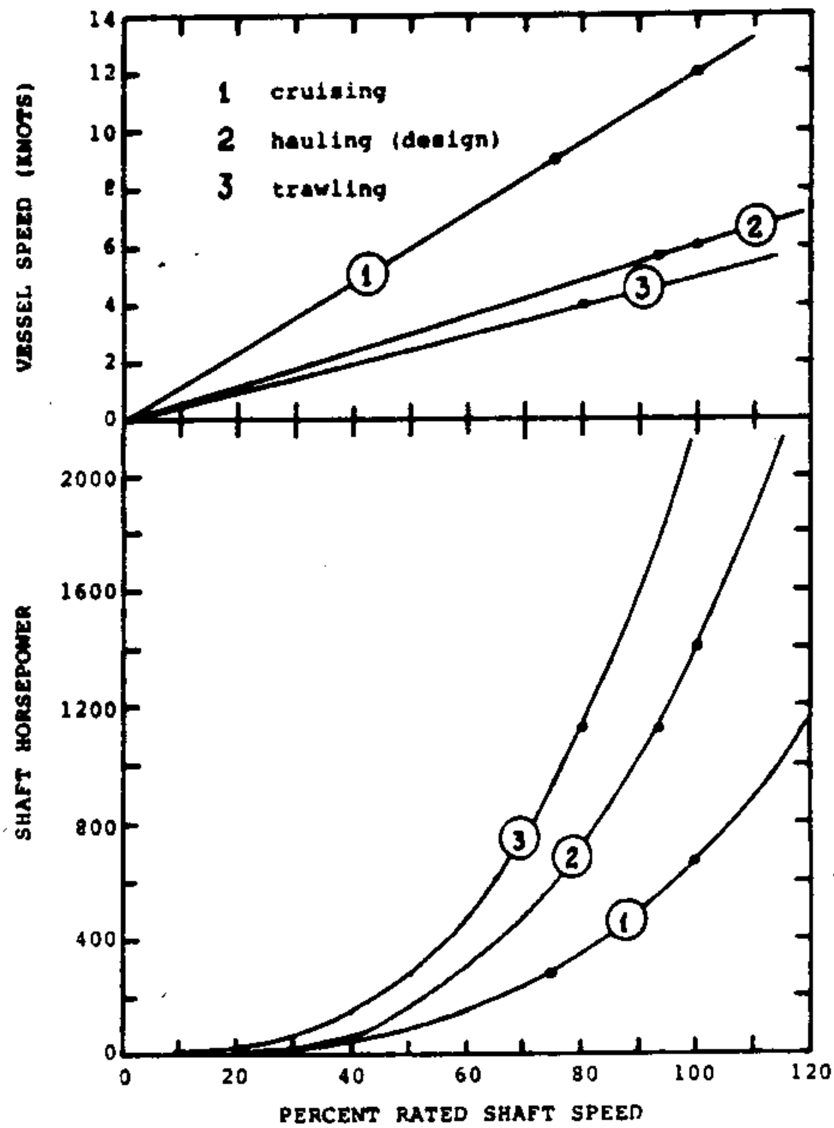


FIGURE 46. SPEED-POWER CURVES OF THE BASELINE FISHING VESSEL EQUIPPED WITH A FIXED-PITCH PROPELLER.

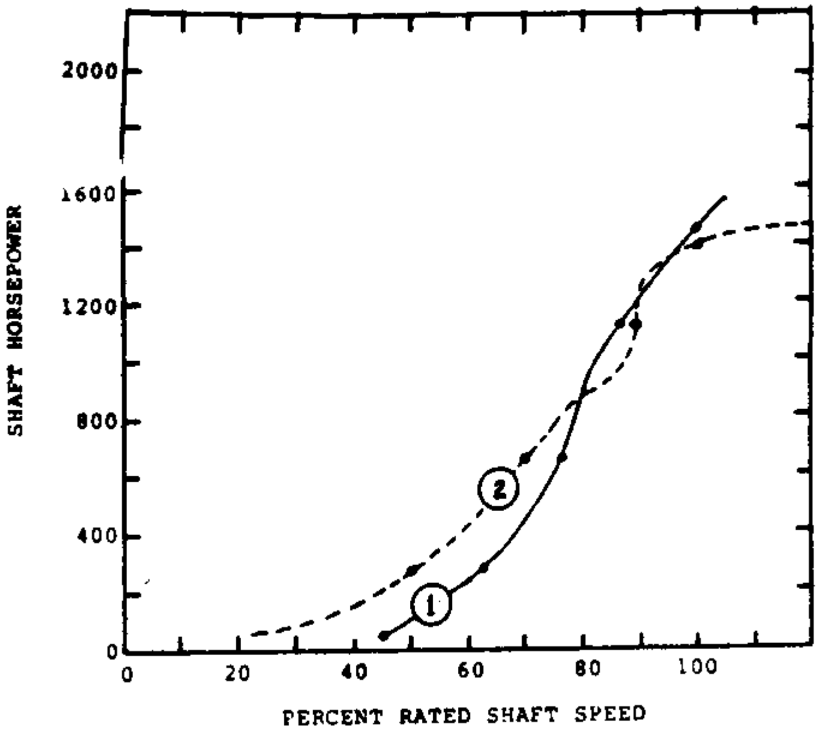


FIGURE 47. SPEED-POWER CURVES OF THE BASELINE FISHING VESSEL EQUIPPED WITH A CONTROLLABLE-PITCH PROPELLER.

CPPs require. At off-design points the CPP would have higher open-water efficiency than the FPP, because the pitch would be adjusted to the optimum value for that condition. In the above we have neglected the mechanical power-transmission efficiency of the shaft system (which for both engines and propeller types is about 98%, (ref. 52)), and we have not considered the possibility of propeller cavitation, (defined in reference 50), at any condition for two reasons. First because it would depend on the particular boat and propeller and second because there was insufficient information available. Overall, the above assumption (that the CPP requires the same shaft horsepower as the FPP to power the boat at the same vessel speed) largely overestimates the fuel consumption with a CPP.

The results of the annual fuel-consumption calculations, which are included in Appendix D of reference 12, are summarized below:

diesel engine with FPP: 263 thousand U.S. gallons / year (100.0 %);
diesel engine with CPP: 253 thousand U.S. gallons / year (96.2 %);
LPR engine with FPP: 172 thousand U.S. gallons / year (65.2 %); and
LPR engine with CPP: 158 thousand U.S. gallons / year (59.8 %).

The current price of fuel (the study was performed in 1984) is about $1.06 per U.S. gallon, but at the end of the year fishermen receive as rebate a small percentage of the cost of fuel they consumed. The final cost of fuel is about $1.00 per U.S. gallon. Therefore the above numbers can also represent the respective annual fuel costs (required for the fuel of the propulsion engine only) in 1984 dollars. For the above operations the LPR engine used as the main-propulsion device consumes only 60% to 65% of the annual fuel cost that the diesel engine would consume as the main propulsion device.

In the above calculations we did not account for the fuel required for auxiliary loads at sea and at port. These loads would be met in both cases with auxiliary diesel-generator sets. Since these loads need to be met with either propusion engine, the above results are unaffected.

COMPARISON OF SOME OTHER ASPECTS OF THE TWO ENGINES

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 space available in the vessel that can be allocated as hold space. In turn this would allow for larger catches per trip and longer trips, thus minimizing transit time between trips and resulting, in general, in more efficient operations. The longer trips would require means of refrigeration better than the simple storage in ice that is still frequently used; this in turn would improve the quality of the landings. The absorption refrigeration cycle (ref. 9) would be an advantage. Alternatively for the same catch the LPR engine would permit the design of smaller fishing boats that would have better arrangements and would require less propulsive power, thus economizing fuel twice.

Diesel engines cannot be started and run up to full power from a cold condition; they require a period, increasing with size, to warm up. The LPR engine could be started in minutes. The diesel engine is approximately 3.54 m long by 1.70 m wide by 2.13 m high and weighs dry 8081 kg. 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 disks 0.14 m thick and 1.75 m in diameter. (Recent advances in rotary-regenerator technology may permit a single disk of larger dimensions). The weight of the LPR engine would be between 1000 kg and 2000 kg. With this lighter propulsion engine the hydrostatic stability of the boat must be studied as the metacentric height, (defined in ref. 50), may be reduced to unacceptable levels).

The LPR engine would be quieter and would give less vibration than the diesel engine. This would result in improved living conditions and happier (and hence more efficient) crew. The exhaust from the LPR engine would be less noxious and would smell better than that of the diesel. Because the engine is smaller it could be installed on board the vessel in less valuable places than the present engine room, thus allowing more flexibility in design and improving the general arrangement of the boat.

The diesel engine requires smaller inlet and exhaust ducts than the LPR engine. Gas-turbine engines are sensitive to salt ingestion and therefore the air inlet must be protected from spray. With a propeller designed for 290 rpm the diesel engine requires a single-input/single-output gear unit of gear ratio 5.5:1, while the LPR engine requires a gear ratio of 58:1. This means that two gear sets are required for the LPR engine. Typically the problem would be solved with an epicyclic gear of ratio 12:1 (probably an integral part of the gas-turbine engine) followed by a single-input/ single-output gear of ratio 4.8:1. Special attention is required on the controls of the LPR engine to prevent overspeeding in sudden removals 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 reference 52.

The marine environment is severe for any engine. There is 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 fishing boats. Another undetermined factor is the reliability and maintenance requirements of the LPR engine. In a recent comparison of maintenance requirements of diesel engines and gas-turbine engines, (ref. 23), we concluded that gas-turbine-engine maintenance cost and time requirements are:

difficult to quantify;
comparable to maintenance costs and time required for diesel engines;
can be high with unfamiliar personnel; and
can be reduced to levels below those of diesel engines when the personnel becomes familiar with the engine (or with trained personnel).

It is known that few fishermen have maintenance schedules for their diesel engines. The maintenance they do carry out is usually limited to oil and filter changes. With this state of affairs in the fishing industry, the maintenance of the LPR engine remains an unanswered question. With a fully developed gas-turbine engine fishermen 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 spray with distilled water or rice injection.

Finally, we do not know the capital cost of the LPR engine. However we estimate that for mass production it will be comparable to or less than the capital cost of the diesel engine (figure 1.12 of reference 1), especially if the engine is designed with a reinforced-plastic compressor).

CHAPTER 9

THE RESPONSE OF THE NEW ENGLAND FISHING COMMUNITY

BACKGROUND

In the course of this study we undertook two surveys of the fishing industry. The purpose of the first survey was to identify the baseline fishing boat. The purpose of the second survey was to probe the response of the New England fishing industry to the LPR engine proposal. During these surveys we became increasingly aware of other not-so-obvious factors which influence the outcome of any proposition (such as the LPR engine) to the fishing industry. The reason for these "side effects" is rooted in the history of the New England fisheries. In the following we report the results of the second survey of the industry.

SOME FISHING-INDUSTRY RESPONSES TO THE LPR-ENGINE CONCEPT

We conducted a second survey of the New England fishing industry, with the intention of probing the response to the proposed LPR engine. We interviewed fishermen in Boston, New Bedford and Gloucester (Massachusetts) and Rockland (Maine). We initiated each interview with a series of statements about the LPR engine's efficiency, fuel savings, resulting larger holds, quieter running and the possibility of "free" refrigeration with the absorption chillers.

We continued with a series of questions of the following nature.

- Given the superior performance of the LPR engine, would they be interested in installing the new engine in one of their boats?
- How many days of additional downtime for engine maintenance, (in excess to those required for the diesel), would they be willing to accept for the LPR engine in order to save 35% of the fuel consumed annually by the propulsion engine?
- If they would consider installing the LPR engine as a first step, would they also consider as a second step absorption chillers heated by the engine exhaust energy?
- Under what conditions would they consider installing the LPR engine in one of their boats?
- What support, (possibly in the form of a subsidy), would they like to be offered before they considered installing the LPR engine?
- What maintenance support would they require for the LPR engine?

It was difficult to conduct the survey because the fishing boats are out of port most of the time. Skippers avoid bringing all the boats in port the same day; if they did, the market would be oversupplied and the prices would drop. As reported in most books on the subject, the majority of the fishermen are of Italian, Portugese, Norwegian etc. descent. Since they spend a large percentage of their lives at sea with other fishermen that speak their mother language, few of them speak English. But overcoming the language barrier was not sufficient to evoke a reasoned response to the survey questions. It was obvious from the beginning that while there are some fishermen and boat owners who understand well the many factors that produce efficient engine-propeller combinations, there are also many who make their decisions in this area more by instinct or by following the example of others. Typically there would be many fishermen who would ask: "What kind of engine is a gas turbine?". However, the survey did give us some answers and a better understanding of the U.S. fishing industry. The two distinct categories of fishermen gave distinct groups of answers.

THE FIRST CATEGORY OF FISHERMEN AND OWNERS

The first category of fishermen and owners verbalized little of their thoughts on the LPR engine. Few of them expressed enthusiasm for the proposed "new" engine. This in part is due to the calm, proud, self-reliant nature of the fishermen. They did not seem much concerned about the fuel savings, but they did mention that it was very important to them that the boat was operating and earning money by catching fish rather than staying idle in a repair facility. The skippers were not concerned very much with fuel costs and fuel savings since these costs are deducted from the crew's share. They pointed out that installation of the LPR engine in an existing boat would involve removal of the bridge and an expensive diesel engine. Both processes are time-consuming as well as expensive, and they would not favor restricting the boat from fishing during the engine change and reselling an expensive diesel that has served them well (and probably still has many years of useful life left). Only somebody that was building a new boat might consider installing the LPR engine. They did not know what absorption chillers are. However, some of them did ask about the price of the LPR engine, a question that we were unable to answer satisfactorily.

It was clear from the interviews that nobody wanted to be the "guinea pig" to try the engine first. If the engine did not perform as well as expected their fisherman's pride would be hurt too much. They would rather let somebody else try it first. For maintenance and repair support they would like to have available to them facilities that could undertake maintenance and repair of the LPR engine, like the support available for diesels. If they were to buy the LPR engine they would like it to be available "off the shelf" and thoroughly tested like the diesel engines. Of course if all the above were available then there would not be any reason not to prefer the LPR engine over the diesel. Although the fishermen would welcome any subsidy on the installation of the LPR engine, if all the above were available the government would not need to subsidize installation of the engine.

THE SECOND CATEGORY OF FISHERMEN AND OWNERS

Much information was supplied by the second category of fishermen and owners. They understood that fuel costs account for a large percentage of the price of landed fish and therefore expressed great interest in a fuel-efficient engine. Like the first category they did not want to be the first to experiment with the LPR engine, but their reasons were different. They pointed out that fishing is a marginally profitable business and therefore they could not afford the experiment because negative results would financially destroy them. However, they pointed out that the operating profiles of tugboats are similar to the operating profiles of fishing boats. Since most tugboat companies have a series of tugs, one of them could possibly be withdrawn from service for some time to test the engine. They also suggested testing the LPR engine on a land-based gas-turbine-generator installation. (Of course this, or some other similar test, would be done anyway.) If one of these two tests (or both of them) were carried out successfully first, then they might consider installing one LPR engine in one of their boats as a first step. Subsequent installation of the LPR engine in other boats would depend on the performance of the first.

Fishermen of the second category were also interested in other aspects of the engine. They were interested in the proposal to install the LPR engine with the shaft in a vertical position on the bridge and to allocate the vacated space in the engine room to holds. They did ask questions about the shafting arrangement, cooling and lubricating-oil systems and the ducting arrangements. We provided the following answers.

For the shafting system: an epicyclic gear of gear ratio 12:1 incorporated into the engine would reduce the shaft speed to 1391 rpm. A vertical shaft would transmit power to a horizontal shaft via a 4.8:1 bevel-gear set located near the keel giving 290 rpm at the propeller.

For the ducting: the ducts for the LPR engine are bigger than the ducts for the diesel engine. The intake duct must be protected from spray and salt ingestion. The performance of the engine is sensitive to duct pressure losses. By locating the LPR engine high on the deck, on the pilot house perhaps, duct length is minimized (which minimizes both duct pressure losses and duct cost) and the intake location would be in the area of the boat that is subjected to the least amount of spray and salt ingestion.

For the cooling and the lubricating-oil systems: unlike diesel engines, gas-turbine engines do not require cooling water. Some air cooling occurs internally in the engine (compressor delivery to turbine blades). The lubricating-oil system of the engine is self contained and the lubricating-oil consumption is negligible.

Some fishermen stated at this point that they liked this last feature of the LPR engine because it requires only fuel and air connections. In comparison, diesel engines in addition to the above require lubricating oil, sea-water and fresh-water connections. The cost of these connections constitutes about ten percent of the cost of the diesel engine installed.

To our questions about absorption chillers they replied that they have been used with success in Europe, but there is not enough support available in this country. However, if they were to install the LPR engine, and if the cost was reasonable, they were willing to experiment with installing an absorption chiller to save the extra fuel and to test the two units together.

Despite the additional payload that would be available if the LPR engine was installed, fishermen of the second category stated that they would not increase the duration of their trips above 8 to 9 days. They prefer offering fish that has been refrigerated at about 0 C (32 F). They stated that the quality of frozen fish is lower than the quality of refrigerated fish, and the quality of refrigerated fish deteriorates after the ninth day.

Finally, they stated that they would not consider installing the LPR engine unless it was at least as reliable as the diesel engine. They stressed that for less than 25 trips per year the operation becomes unprofitable. They were not willing to compromise present levels of reliability for any amount of fuel savings. One of the reasons they cited for this inflexible stance was that the crew can profit only when at sea. The crews would not want to work on a boat with an unreliable engine, and they would not hesitate to work for another owner that provided a more reliable engine.

FISHERMEN AND OWNERS IN GENERAL

Most fishermen have FPP installed in their boats. Few fishermen have realized the advantages of CPP (which were illustrated in chapter 8). These few are divided into fishermen who think that the fuel savings with a CPP are sufficiently big to justify the additional capital expenditure required for this kind of propeller, and fishermen who think exactly the opposite.

Most boatyards we visited could, (to some extent), repair and maintain diesel engines. Usually there would be a diesel mechanic nearby, or working in the yard. Of course, none of the mechanics in the boat yards we visited was qualified to or would undertake gas-turbine- engine repair.

Finally, in the presentation of reference 53 the audience responded with great enthusiasm to the LPR engine. They thought that the future potential of the engine is promising. However, the majority of attendees were U.S. Navy and Coast Guard officers, academics etc.

The conclusions of our survey are that fishing-boat owners have little incentive to install more-efficient engines in their boats because the fuel bill is paid by the crew. Since boat owners prefer to have the boat in operation rather than in a repair yard, owners would not consider re-engining existing boats. The existing-boat market is virtually eliminated for the LPR engine. Because of the risks associated with the LPR engine, because fishermen are afraid of the possibility of ridicule in their community and because (for good reasons) they are reluctant to change their ideas, it is unlikely that the LPR engine will be used in a new boat unless it is tried and proved elsewhere. It is not clear if a subsidy would induce fishermen to use the engine.

Fishermen and boat owners must be conservative because of the particular events that led the U.S fishing industry to its present marginally profitable condition. However, the industry needs an efficient engine. There are some ways by which the necessary support could be provided. The government could provide a subsidy in the form of a money-back guarantee if the installation of the engine proved unsuccessful and in the form of a bonus for the boat owner who first installs the LPR engine in his boat. If the first installation proved successful many other marine and land-based users would order more units; therefore the engine builder would benefit from the sales and the other users would benefit from having an efficient engine available. It seems that gas-turbine-engine manufacturers and potential users of the LPR engine should be approached for financial and technical support. There is the possibility that additional research funds may be allocated to actually build and test the LPR engine, or to modify an existing engine to a low-pressure-ratio, regenerative cycle and to test it to prove the feasibility of the LPR engine. The risk could be spread to a larger number of owners if a fishermen's union or a co-operative was approached with the idea. There is much foreign investment in the U.S. fishing industry of the West Coast. These foreign interests would also benefit from the introduction of a more-efficient engine in the fishing industry and therefore could be approached for financial support. Finally it is unlikely that any single one of the above proposals alone could withstand the technical and financial burden of implementing the LPR engine, but a combination of some of them could be successful.

CHAPTER 10

CONCLUSIONS AND RECOMMENDATIONS

We believe that the difficulties of implementing the LPR engine in the fishing industry will eventually be overcome, because in today's efficiency-conscious world there are many other engine users that would be interested in trying to implement in their industry a prime mover with low fuel consumption. The LPR engine also increases payload, which is important in all marine applications. Most Navy fast-attack craft, Coast Guard cutters, various patrol craft, tugboats, planing boats etc. require prime movers with power ratings near the rating of the LPR engine. There is a large sector of the marine market that would be interested in the LPR engine.

Uses of the LPR engine may also prove advantageous to land-based installations. Although payload is not a concern in stationary plants, fuel efficiency is very important. The thermal efficiency of the LPR engine is attractive for any application.

Low maintenance demands are welcomed in any application. Although we cannot quantify maintenance requirements for the LPR engine, the maintenance requirements of gas-turbine engines in marine-combat duty cycles have proven very low [47,51]. It is highly probable that future gas-turbine engines will require less maintenance than diesel engines.

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. In our opinion the lack of enthusiasm for the LPR engine encountered in the fishing industry will not be detrimental to the further development of the concept and the final implementation of LPR engines in various applications.

In this study we have considered the development and introduction of a high-efficiency prime mover for marine propulsion. In particular we based the study on the U.S. fishing industry which faces some efficiency problems and fierce foreign competition.

The prime mover is a low-pressure-ratio, highly-regenerative Brayton-cycle (gas-turbine) -LPR- engine. We calculated the performance of this engine at design point and at off-design points. At design point the thermal efficiency is about 55%. At off-design points the thermal efficiency remains higher than the respective thermal efficiency of diesel engines. The resulting performance is predicted to be better than the performance of the most efficient prime mover available for this application, the diesel engine.

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 energy at exhaust.

However the LPR engine has not yet been built; it therefore has two disadvantages: lack of tested hardware (which would prove the above claims), and a maintenance-effort requirement and a reliability that is impossible to predict.

The LPR engine appears advantageous for many marine and land-based installations. In this study we compared its performance with that of a diesel engine in fishing-boat propulsion. Indeed, it was estimated that for the operating profile of the baseline fishing vessel the LPR engine would consume less than 60% to 65% of the fuel the current diesel engine consumes. We would normally expect the fishing industry to respond favorably and actively support further development of the cycle and the engine. But there are a number of reasons which prevent the U.S. fishing-boat industry from supporting further development. These reasons became apparent after a study of the history of the New England fisheries and two recent surveys of the industry. Fishing boats operate with marginal profits and cannot afford experimentation. And because fishermen are paid by the lay system and the fuel costs are deducted from the fishermen's share, boat owners have little incentive to spend useful operating time and money to become more efficient. However, some fishermen would be willing to experiment with the engine if it is thoroughly tested in a tugboat or a land-based installation and shown to be reliable.

The predicted efficiency of the engine is very attractive - more favorable than all other prime movers available today. We recommend further development of these engines. We recommend work to continue to the next stages: refining the design and building a prototype engine for testing. In this case the fishing industry may be able to participate through the fishermen's union and/or fisheries co-operatives.

In the above we investigated a single-shaft LPR engine. We recommend a similar investigation of an LPR engine with two expanders of which the first will drive the compressor and the second the load. We also recommend actually building and testing the single-shaft or the two-shaft LPR engine, with a reinforced-plastic compressor, ceramic expander(s) and rotary regenerator(s). The above procedure should quantify reliability and maintenance schedules.

Many marine engineering and naval architecture aspects need to be studied before an LPR engine is installed in a fishing boat, including the arrangement, ducting, shafting, hydrostatic stability, propeller type etc. We recommend a detailed study of all the above before the first LPR engine is installed in any marine craft. No major problems seem likely to be encountered.

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