Analysis and Modeling of Power Transmitting Systems for Advanced Marine Vehicles

by

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B.E. in Mechanical Engineering
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Submitted to the Department of Ocean Engineering in partial fulfillment of the requirements for the degrees of

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Abstract

In this thesis, a new engine is considered for applications aboard mid-size navy surface combatants. The so called Air-Bottoming Cycle (ABC) is analyzed thermodynamically and a preliminary design was completed. The ABC is an air turbine which is coupled to an existing gas turbine, which in this example was the General Electric LM2500 marine modulus. The working fluid of the new, second, turbine is unvitiated air, meaning air that its oxygen has not been subjected to a combustion process. The new engine is designed based on radial rotating components, having two stages of intercooling, each one appearing between two adjacent compression stages. A heater, between the last compressor and the high pressure turbine, is used to raise the temperature of the compressed air exiting the high pressure compressor by crossflowing the hot exchaust gases of the LM2500 gas-turbine. With a compression ratio of 2:1 at each stage, heated air passes through three turbines that drive the corresponding compressors and finally expands through a power turbine which delivers 9,700 HP or 34% additional horsepower to that of the LM2500. The additional power delivered, results to a drop of the average specific fuel consumption of the LM2500, from 0.793 to 0.503. This 36.6% improvement of the sfc brings substantial fuel savings over the 40-year life span of a Naval vessel. Two scenarios of potential applications for the engine were examined. In the first, the air-turbine provides shaft power to the propeller. Using the operatating profile of a DDG-51 destroyer of the U. S. Navy, such an application of the ABC generates 30% of savings in propulsion-fuel cost compared to the LM2500. In the second scenario, the air-turbine of the ABC is coupled both to the main transmission and to electric generators. Thus depending on the power requirement, it can either deliver shaft-power or generate electric power. The cost savings from such operation of the ABC drop the annual cost of propulsion fuel by 38.5% for a DDG-51. while the savings from the first scenario are 22.5%.

Thesis Supervisor: A. Douglas Carmichael
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To my beloved parents,	*Αφιερώνεται στούς Γονείς μου ,*
Thekla and Miltiadi	*Θέκλα καί Μιλτιάδη*
Kampani.	*Καμπάνη .*

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To say the least, all this work would not have been possible without the love, support and trust of my beloved parents, who have always been sharing my enthusiasm of my research at M.I.T. and who never hesitated to cross the oceans that I have been studying in the past four years in order to help me get through the difficult times that I ever encountered.

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Leonidas M. Th. Kambanis

May 22, 1995

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Chapter 1

Introduction

1.1 General

During the early stages of the ship design process, the selection of the propulsion and auxiliary system takes place. Not only these two systems along with the fuel account for a great percentage of the vessel's displacement (forty percent in a surface combatant), but also they have a pervasive influence on both acquisition and operating costs as well as the following stages and iterations of the design spiral.

In the past fifty years, the most revolutionary technological advancement for the propulsion of naval ships is perhaps the gas turbine. Navies worldwide have been exploring, developing and adopting the gas turbine as an efficient, cost-effective, reliable prime mover for most combatants. It is the preferred choice for propulsion on most mid-sized naval combatants; future estimates, suggest that by the 21st century almost all U.S. Navy ships will be powered by gas turbines [1].

A significant drawback of gas turbines in naval ships is their poor off-design efficiency, and navy ships spend a significant portion of their operating life at low power levels. So far, and to a large extent, the technology and its development depends on the ability to transfer and to apply processes from the aerospace industry. Nevertheless, gas turbine technology has matured so much, that it may not be cost effective to improve a simple cycle by either using exotic materials or advanced manufacturing methods. The gains are relatively small, while the investment can be very significant. On the other hand, a simple cycle can enjoy major improvements, in both performance and operating cost, at design and off-design, by rerouting the flow of the fluids driving the engine and introducing heat transfer. In the past ten years the U.S. Navy engineers have investigated applications of various modifications to the simple cycle, and configurations that will provide more flexibility, higher reliability, while they could be efficient and cost effective. Along the same lines, the Air-Bottoming thermodynamic cycle, patented by William Farrell of General Electric, is presented, analyzed and modeled, demonstrating that waste heat recovery can be an efficient and yet, not an expensive additional source of energy to that of a primary, simple Brayton cycle.

1.2 Background

In the mid-1970's when fuel cost got higher and a because very significant contributor to the life-cycle cost of a combatant. Various programs, that would lead to the design of a more cost effective power plant, were proposed for investigation and development. Along with the rising fuel prices, the commitment of the U.S. Navy to gas turbine propulsion made more attractive the conversion of waste heat into useful shaft power. For a simple gas-cycle engine such as the General Electric LM2500, which is the main propulsor of most mid-sized surface combatants, such a development program was the design of the RACER (Rankine Cycle Energy Recovery) system.

In December 1979, the Naval Sea Systems Command awarded contracts to develop the conceptual design to provide 25% minimum improvement in ship propulsion fuel consumption and increase the ship's operating range by 1,000 nautical miles [2]. It was found that a considerable amount of energy could be recovered

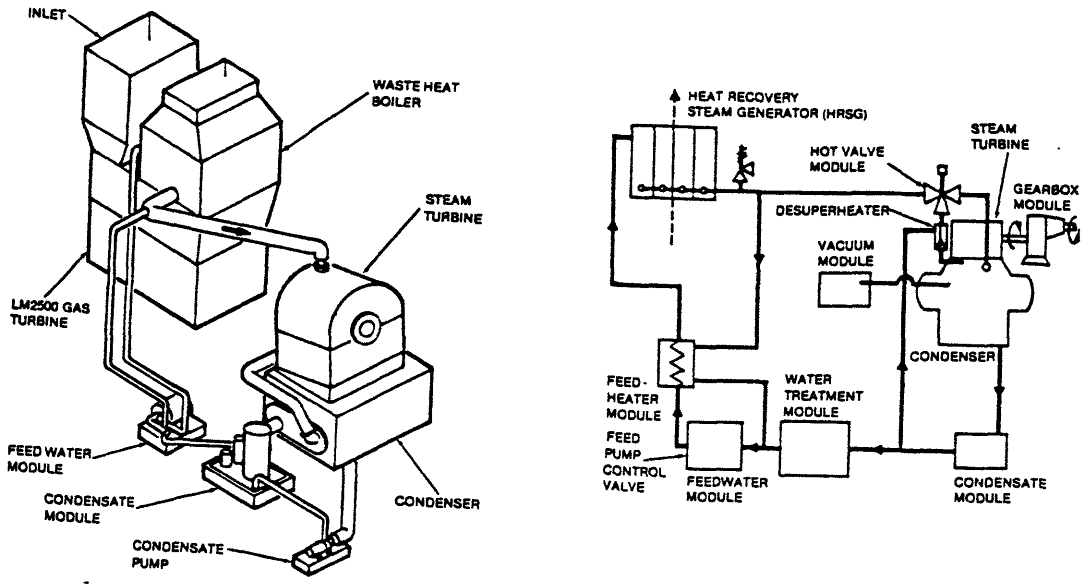


Figure 1-1: RACER modular components and baseline schematic.

and re-used by a waste recovery system using the hot exhaust gases (1000 F). The RACER system was an unfired waste heat recovery system that used the exhaust gases to generate medium pressure steam that can be used as an additional source of energy to the main propulsion unit of the LM2500. Figure 1-1 shows both the schematic and the modular components of the system. By reducing the load on the main propulsor, this configuration was operating as a nonmission critical system and could drive fuel consumption down by 20%. Nevertheless, a steam and gas turbine combined cycle has a number of drawbacks. Steam is a medium much different from gas, to contain and handle. The size and technology of the appropriate components that would guarantee the performance and reliability of the cycle, and the purity of the steam increase the costs related to the development, acquisition and operation of such a cycle. Also the fact that heat recovery steam generator is much less responsive to required changes in output than the gas turbine, contributes to the loss of even the most attractive characteristics of the RACER engine. Since the Navy requirements asked for minimum weight, compact design and simplicity of the system, non of the existing components was found appropriate for the new engine. Although new components incorporating innovative design were developed, the additional weight and volume of this so called COGAS (Combined Gas and Steam) arrangement, along with the high production costs associated with the high technology components used, did not take the RACER engine much further than the testing facility.

Since the demand for higher efficiency and lower operating cost continues growing, the U.S. Navy is now looking into other configurations that may have better chance of success than the RACER program and may finally see the light of production. The unattractive features of RACER along with the low efficiencies demonstrated by the simple gas turbine cycle at part-load conditions, have led towards the development of an intercooled, recuperative gas turbine engine. In this engine, heat from the exhaust gas stream is recovered by a gas-to-air heat exchanger (recuperator) and preheats the compressed air before it enters into the combustor. By

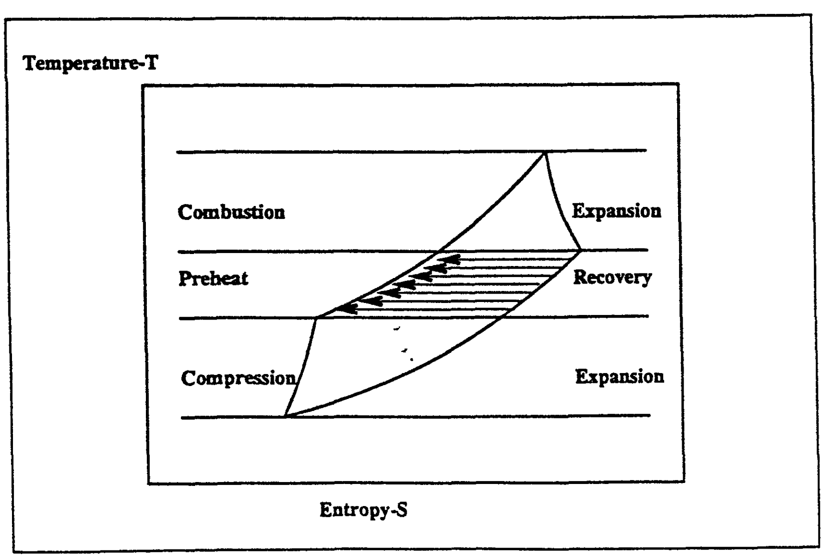


Figure 1-2: Regenerative cycle's T-S diagram of heat recovery potential.

doing this, less fuel is required to heat the gas to the turbine inlet temperature. Figure 1-2 shows how regeneration alone reduces the required fuel needed to raise the compressed-air temperature to the combustion inlet temperature. The improvement in thermal efficiency due to adding the recuperator to the simple cycle depends on the difference between the exhaust gas temperatures and the compressor exit air temperature, as well as the effectiveness and pressure drop of the recuperator. Lately, the trend in high end commercial gas turbines is towards higher compression ratios. As compression ratio increases to 18:1 and over, the above temperature

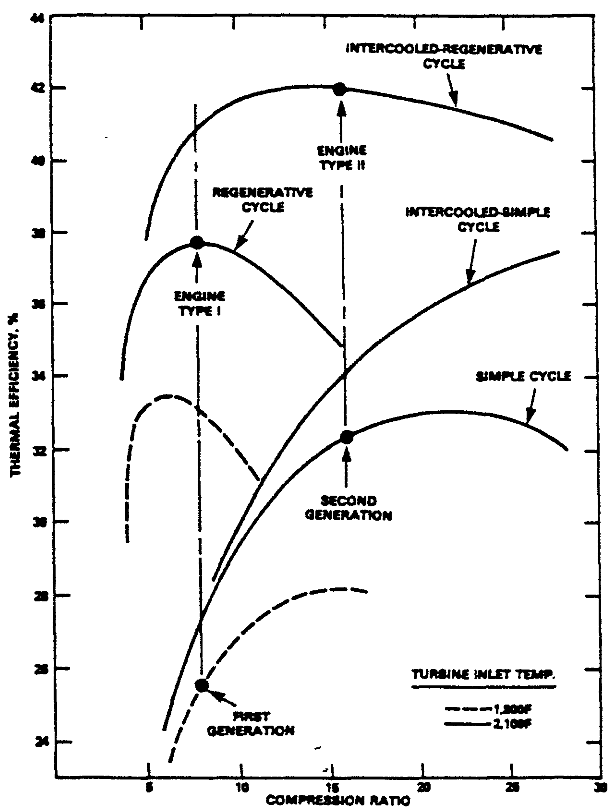


Figure 1-3: Thermal Efficiency Improvements from Intercooling and Regeneration. Source: Bowen and Groghan [3].

difference approaches zero and heat transfer is no longer possible. To overcome this obstacle, air-to-air or air-to-water heat exchange is introduced at an intermediate stage of the compression process. This process is called intercooling. Thus, not only the compressive work is reduced, but also the overall temperature rise across the compressors is lowered. The savings in compressive work can be significant to the performance of a simple cycle engine since two out of every three horsepower generated by the turbine is used in the compression [1]. Recuperating a first-generation marine, simple cycle, gas-turbine can increase the thermal efficiency from the current 26% to almost 38%, depending on pressure ratio and other factors, while just intercooling can improve the same quantity of the second generation engines from 32% to 34% for the same compression ratio without requiring advances in the basic core engine technology. Intercooling and regeneration together, can take the thermal efficiency of the existing marine gas turbines up to 42%. A graphic representation of all this is shown in Figure 1-3. The extra weight of the intercooler and regenerator can be off-set a little from the smaller engine-core that an ICR engine requires. The combination of the 30% fuel savings that such an advanced cycle can provide, along with the weight savings associated with the intakes and uptakes, can raise the cycle's power by 35%. The greatest fuel savings come from the off-design fuel consumption improvement, which consumption in the simple cycle is higher by as much as 40% from that of the advanced cycle.

Cost has always been a driving factor of the development of a new engine. In the case of gas turbines the highest cost is that of developing the rotating parts. Much effort was put on adapting existing simple-cycle turbomachine to the regenerative-intercooled configuration. As a result, three alternative core engines were identified which could yield regenerative or intercooled-regenerative gas turbines suitable for propulsion of mid sized surface combatants. These distinct alternatives included:

- Modifying the Rolls Royce Marine Spey which has a dual-spool gas generator to include both an inter-cooler and recuperator.
- Modifying the General Electric F404 dual-spool military turbofan to include both an intercooler and recuperator and incorporating a new-design power

turbine to obtain a marine propulsion engine.

- Modifying the high-pressure spool taken from the Pratt & Whitney PW2037 or PW4056 commercial turbofans to include a recuperator and incorporating a new design power turbine to obtain a marine propulsion engine [3].

Finally, in 1983 contractors studies were initiated and in 1992 Westinghouse Electric Company was awarded the contract as the overall ICR-engine development program coordinator. The core of the new engine is based on the Roll Royce Marine Spey, while Allied Signal Air-Research Company was awarded the development of the intercoolers and the recuperetors of the engine.

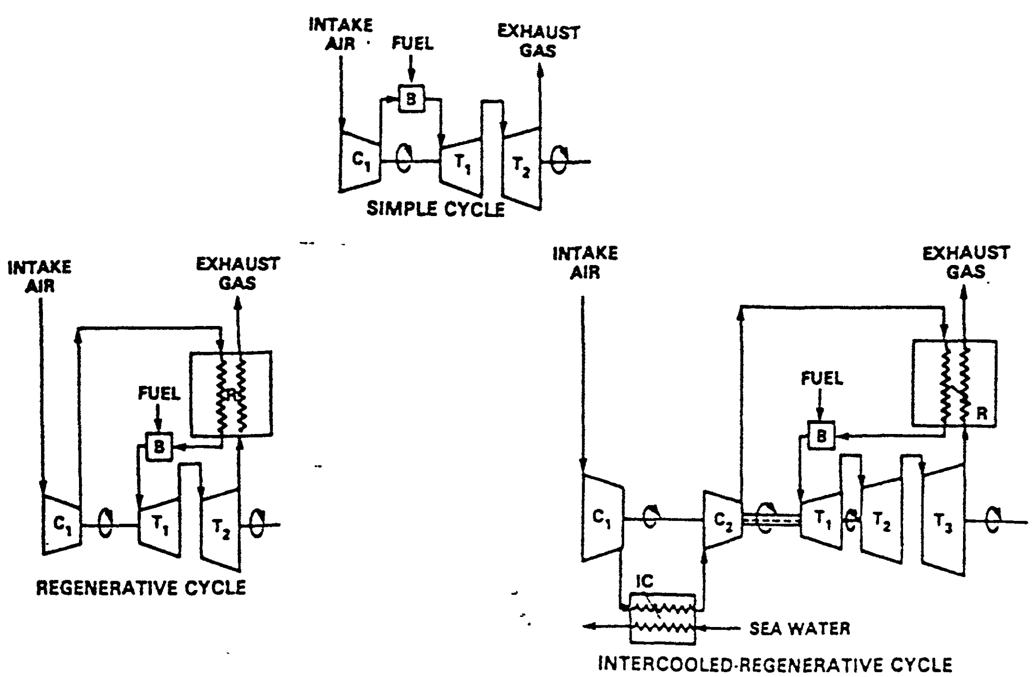


Figure 1-4: Schematic Diagram of Simple, Regenerative and Intercooled-Regenerative Cycle Configurations.

General Electric, having spent the energy in putting together a competitive bid for the ICR engine, decided that they would keep developing their own version of an advanced gas-cycle. The result of this is the regenerative LM2500. It is an engine that as its name suggests, is going to be based on the so-far standard engine for the mid-sized surface combatants, but with regeneration from the exhaust gases pre-heating the compressed air before entering the combustion chamber.

Figure 1-4 shows the conceptual differences between the simple gas turbine cycle, the regenerative cycle and the intercooled-regenerative cycle.

Another potential improvement to gas turbine propulsion of Navy ships is the Air-Bottoming Cycle (ABC), a cycle using air as the working fluid to develop power from the waste heat of the gas turbine. This cycle will be investigated to determine whether it could provide a viable alternative to the ICR engine.

1.3 Thesis Objective

William Farrell of General Electric's Industrial Gas Turbine Division, patented a new conceptual idea of a combined (or advanced) gas turbine cycle. This is the Air-Bottoming Cycle (ABC) or the Farrell Cycle, named after its inventor.

The ABC is a thermodynamic gas-turbine cycle, using unvitiated air as its working fluid, meaning air that it does not go through a combustion proccess, which is coupled to a simple gas-turbine cycle. The cycle consists of a multi-stage compression proccess where the air passes through stages of compression and cooling. The latter is achieved by the use of intercoolers between two adjacent compression stages. The air before it expands in the turbines, it passes through a heat exchanger that raises the fluid's temperature using the waste heat from the gas turbine. Comparing this new cycle to its main competitor for the propulsion of Navy ships, the ICR engines it is clear that both engines share similar technologies in respect that they both use intercooling and regeneration of waste heat from a gas turbine.

A study was initiated to investigate the potential of this new cycle in finding applications in the surface combatants. As an initial step towards understanding the benefits of the ABC configuration, the components composing the cycle had to be outlined, while the advantages and improvements to the simple cycle, as well as compared to the rest of the combined gas-turbine cycles had to be identified.

This thesis focuses on evaluating the size, cost, and performance of the ABC system. First, the examination of the thermodynamics will help analyze the Farrell Cycle, and will lead to the selection of a design point. This point will set the basic conditions around which component design will be considered. Modeling the design process and its performance evaluation is one of the tasks that will facilitate future alterations and further investigation of the cycle. The analysis and design process is centered around three computer programs and two spreadsheets. The overall design then gives an efficiency prediction, which along with the off-design performance defines the overall efficiency and SFC operating envelop of the new engine. This envelop is then coupled to the operating profile of a mid-size surface combatant of the U.S. Navy is used to develop the operating profile of the engine. Finally, the cost of acquiring the engine will be considered, while the operating costs of the unit in a representative Navy ship, such as the DDG-51, will be estimated assuming the ABC is coupled to an LM2500 gas turbine. As a result of the above analysis, it will be determined, whether such a configuration will be not only applicable and efficient, but also cost effective. Starting in the following chapter, the Air-Bottoming cycle is presented and its major components are identified.

As a closing point, this thesis will recommend future areas of investigation in order for the design of the ABC to become more accurate and detailed. To the extent of a first attempt for the design of this new engine, two scenarios are going to be investigated for possible applications in a Navy ship. Further examination of the applicability of the cycle, such as when coupled to an electric drive, will be proposed keeping in mind the weight and volume implications imposed by the Navy standards.

Chapter 2

The Air-Bottoming Cycle

2.1 An Overview

There are some thermodynamic conversion, or chemical, processes that produce large quantities of hot byproduct gases containing sufficient heat energy to make it economically attractive to collect and use the heat energy [4]. In the open Brayton cycle's case, this chemical process is the burning of fuel and air in the combustor's chamber, the product is the torque produced, and the byproduct is the heat lost through the exit of the hot exhaust gases. Two out of every three horsepower produced are used to drive the compressors, while the remaining power is available on an output shaft.

The Air-Bottoming cycle provides an air-cycle thermodynamic conversion system in which air is compressed in a multi-stage compression process. Intercoolers are used between each adjacent pair of compressors in order to keep the temperature of the compressed gas almost as low as the ambient temperature before each succeeding compression and make the compression work more efficient. The compressed gas is passed through a heater, where its temperature is raised by the waste heat of the gas turbine's exhaust gas flowing in the counterflow. As one of the main advantages and objectives of this combined cycle, the temperature gradient between the heated compressed gas and the cooled exhaust gas is minimized by establishing the two flows such that they both have about equal heat capacities. This minimum gradient is necessary for maximum thermodynamic efficiency of the cycle. Further heat-capture may make use of the effluent heated medium from the intercoolers as well as the exhaust from the turbine [4]. A substantial part of the power produced is used to run the compressors, while any additional expansion of the now hot compressed gas, through a power turbine, can generate power that can be used directly.

There are three main objectives of the Air-Bottoming cycle:

- To provide a combined thermodynamic-cycle system that is coupled to a gas turbine and which has a greater thermodynamic efficiency than the gas turbine alone.
- To retain the operational flexibility and reliability of the gas turbine.
- To employ an air-bottoming cycle for recovering heat energy in a useful form from a stream of heated gas [4].
- To simplify the process for the bottoming cycle.

There are certain functional similarities between the ABC and the ICR, which may be, along with the regenerative gas turbine of General Electric, the main competitor of the new engine. Both the ABC and the ICR are concepts conceived to improve the current performance of the gas turbine which is the main propulsion engine for most mid-sized surface combatants. Both new engines are based on and designed around existing gas-turbine parts. The ABC in this example will be designed around the LM2500, while the ICR's core is based on the Rolls-Royce RB211-535. This is done in order to reduce the development, design and production costs of both new engines. Both engines will use cooling elements (intercoolers) and heat exchangers in order make the power generation more efficient. On the other hand there are some significant differences between the two engines. The combined cycle consists of two turbine modules; the LM2500 unit and the ABC which is the one examined later in this report. The ICR engine is a single integrated unit. Also, the ICR's turbine consists of parts that exists in the industry, while the ABC's air cycle since it is a new part gives some freedom in its design. Thus, in this report radial components have been selected, while William Farrell has approached the design with axial-flow compressors.

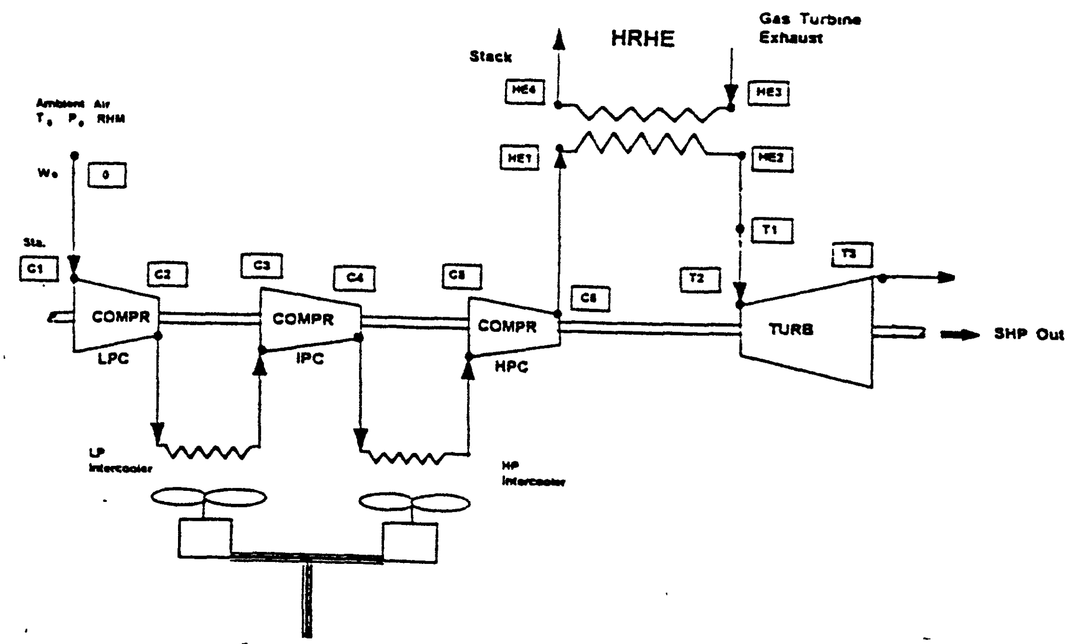


Figure 2-1: Air-Bottoming Cycle baseline schematic. The fans Represent the Cooling Effect of Water on the Compressed Air

For the purposes of this study and for a potential application of this new combined-cycle engine aboard medium-size surface combatants, the General Electric Aircraft Derivative LM2500 Marine Gas Turbine is considered as the source of the hot exhaust gas. In such case, the exhaust gas exits the gas turbine at a temperature of about 1000 degrees F. Furthermore, in the preferred embodiment, the working gas in the compression process, intercooling and expansion through the turbine is atmospheric air. A schematic of this new conceptual combined gas turbine cycle is shown in Figure 2-1.

In the following chapter, some of the thermodynamic principles and the corresponding analysis of the Air-Bottoming Cycle (ABC) will put the base on which the design of the cycle will be developed in Chapter 4.

Chapter 3

Thermodynamic Analysis

3.1 Introduction

The thermodynamic analysis of the Air-Bottoming Cycle (ABC) is based on the analysis of the gas-turbine or Brayton cycle, meaning that the same principles and properties of the working fluids that apply to a simple cycle are used here also. The difference in analyzing the ABC is the addition of three heat exhangers, i.e. two intercoolers and a heater.

The working fluid in the Air-Bottoming cycle is air. More specifically, the air used in the Farrell Cycle is atmospheric air whose oxygen has not been subjected to a combustion process. As described later in this chapter, the combustion process that in most conventional gas-turbine cycles is responsible for raising the thermal energy of the working fluid, is replaced in the ABC by a heating process. Thus, since there is no fuel mixing, it is a valid and a very convenient assumption to treat air as a perfect gas in an initial approach to analyze the cycle.

In the ABC, the compressed air is passed through a heater, where its temperature is raised by the waste heat of the gas-turbine's exhaust gas flowing in the counterflow. For the purpose of this thesis, the General Electric's aircraft-derivative, LM2500, marine gas-turbine, is used as the source, providing the hot exhaust gas. The

LM2500 is chosen for several reasons. Of those, the most dominant ones are the following two:

- William Farrell, who was the inventor and the first to analyze the ABC, used the LM2500 as his waste heat source. Thus, there is a basis for comparing the results of this research.
- The LM2500 is the most widely used gas-turbine in the U.S. Navy, used as the main propulsor for mid-size surface combatants. Thus, there is a basis for comparing engine performance based on the operational profile of a U.S. destroyer.

The exit conditions of the hot exhaust gas coming out of the LM2500 are used for the optimization of the Air-Bottoming cycle. Since optimum and off-design performances are well established for this engine when operating on a DDG-51 U.S. Navy destroyer, the use of the LM2500 facilitates the prediction of the target and off-design performance of the ABC.

3.2 Thermodynamic Analysis of the ABC

The Air-Bottoming cycle provides an air-cycle thermodynamic conversion system in which air is compressed in a multi-stage compression process. Starting from the first compression stage, ambient air is compressed by a pre-determined ratio (2:1 will be used later). With compression, the temperature of the air is increased along the first near vertical line of the T-s diagram describing the thermodynamics of the cycle, shown in Figure 3-1. The compression process is nearly adiabatic. Intercoolers are used between each adjacent pair of compressors, in order to keep the temperature of the compressed air low before each succeeding compression, and make the compression-work more efficient. The compressed air is passed through a heater, where its temperature is raised by the waste heat recovered from the hot gas-turbine's exhaust gas flowing in the heat exchanger's counterflow. The hot compressed air is then expanded in the turbine to develop the power of the cycle.

As one of the main advantages and objectives of this combined cycle, the temperature difference between the heated compressed gas and the cooled exhaust gas is maintained in minimum by establishing the two flows such that they both have about equal heat capacities. This minimum gradient is necessary for maximum thermodynamic efficiency of the cycle. Further heat-capture may make use of the heated coolant from the intercoolers as well as the exhaust from the turbine [4]. A substantial part of the power produced is used to run the compressors, while any additional expansion of the now hot compressed gas, through a power turbine, can generate power that can be used directly.

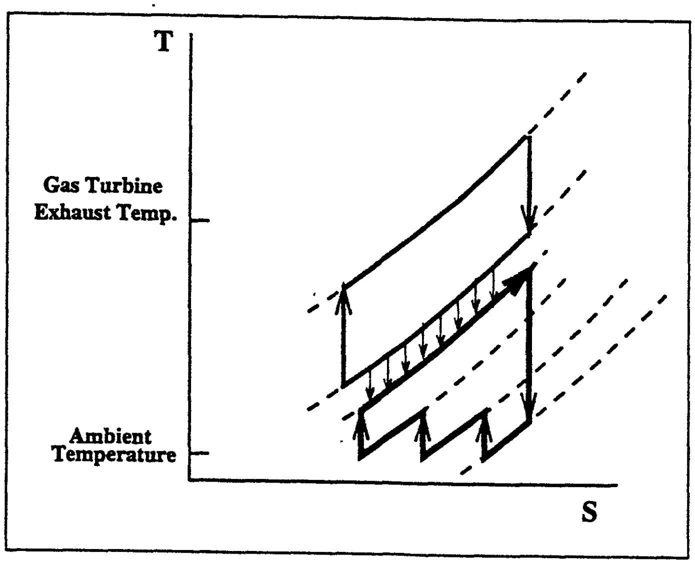


Figure 3-1: T-S diagram of the Air-Bottoming Cycle.

In the thermodynamic analysis of the ABC there are some assumptions that need to be understood. These assumptions are necessary in order to take into account differences in the performance of real cycles from that of ideal ones. In performing cycle calculations, values of both the compressor and the turbine efficiencies need to be assumed. Because turbomachines are essentially adiabatic, the ideal process is isentropic. Thus, isentropic efficiencies may be used. Nevertheless when performing cycle calculations covering a range of pressure ratios, in order to determine the optimum pressure ratio for a particular application, polytropic efficiency, η_{pc}, should be used. It is found that compressor isentropic efficiency tends to decrease and turbine

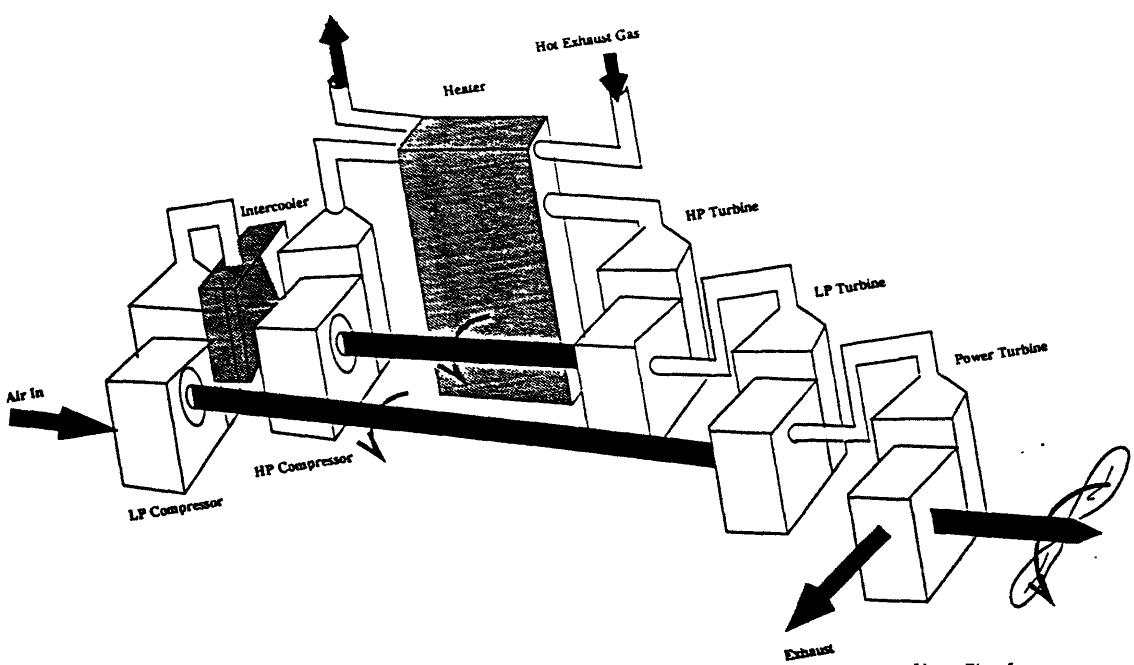


Figure 3-2: Air-Bottoming Cycle Arrangement Diagram.Notice the LM2500 represented by the flow of hot exhaust gas entering the hot side of the heater.

isentropic efficiency to increase as the pressure ratio, for which the compressor and the turbine are designed, increases [5]. Figure 3-3 shows how turbine and compressor isentropic efficiency varies with pressure ratio for a given polytropic efficiency. Assuming that compression and expansion consist of (n) stages, whose pressure difference tends each to zero, the polytropic efficiency is the isentropic efficiency of an element stage in the process, such that it is constant throughout the whole process. For this analysis of the ABC, the polytropic efficiency was set to 0.88 for the compression and 0.92 for the expansion. These values were used because it is assumed the compressors would be large and well designed.

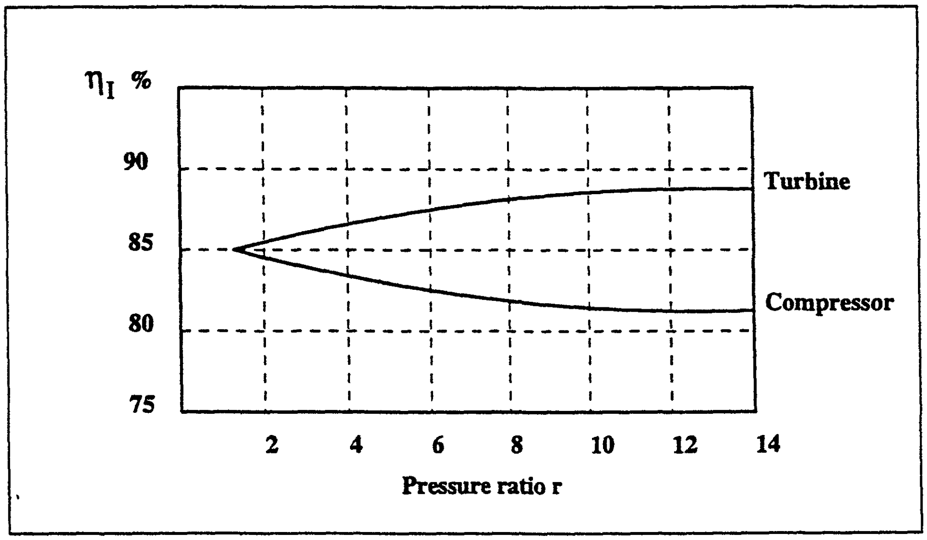


Figure 3-3: Variation of turbine and compressor isentropic efficiency with pressure ratio for polytropic efficiency of 0.85 in each process.

In this cycle analysis both (c_p) and (h) are calculated based on the Chapell and Cochshutt [1]approximation. Knowing that (c_p) is a function of temperature, they were able to approximate its value by using the following polynomial:

[1]In 1974, Chapell and Cockshutt developed these sets of polynomials and their coefficients for generating thermodynamic data by computational methods, as part of a work initiated by the National Research Council of Canada. The data and the polynomials are based on the treatment of dry air and combustion products as semi-perfect gases, so that the specific heat, enthalpy and entropy are dependent solely on temperature and are independent of pressure. Wilson got permission to reproduce the values of these coefficients and with the help of R. Bjorge he presented them in SI units in his book, from which they are extracted and presented in Table 3-1 of this thesis.

$$C_p(T) = \sum_{i=0}^{\infty} C_i \cdot T^i \tag{3.1}$$

The coefficients of the polynomial are constants, with units of (J/kg) and their values are shown in Table 3.1.[2] Enthalpy of dry air, at T-Kelvin, is approximated by a similar polynomial,

$$h_{air,T} = \sum_{i=0}^{\infty} \frac{C_i \cdot T^{i+1}}{(i+1)} + CH \tag{3.2}$$

In the latter polynomial, CH is another constant coefficient whose value also appears in Table 3.1.

	Temperature Range	
T (K) Symbol	200-800 (J/kg)	800-2200 (J/kg)
C_0	+1.0189134E+03	+7.9865509E+02
C_1	-1.3783636E-01	+5.3392159E-01
C_2	+1.9843397E-04	-2.2881694E-04
C_3	+4.2399242E-07	+3.7420857E-08
C_4	-3.7632489E-10	0.00E+00
C_5	0.0	0.0
CH	-1.6984633E+03	+4.7384653E+04
CF	+3.2050096E+00	+7.0344726E+00

Table 3.1: Coefficients for the C_p and $h_{air,T}$ numerical approximations in SI units, reproduced from Wilson [5].

When the air exits a compressor, it enters an intercooler after one or more compression stages or the heater after the final compression stage. The air goes through an almost isobaric temperature change. The inlet temperatures of the compressed air are known for all stages of heat exchanging; either that of cooling (two stages of intercooling used in this thesis) or that of heating. Also, the inlet temperatures of

[2]Wilson D. G., The Design of High-Efficiency Turbomachinery and Gas Turbines, The MIT Press, June 1992.

the cooling water in the intercoolers and the hot gases in the heat exchangers are known. The only temperatures to be determined are the exit temperatures of the compressed air coming out from the intercoolers and before it enters the turbine. To determine these temperatures the heater's effectiveness is defined as,

$$\epsilon_{heater} = \frac{(T_{air,out} - T_{air,in})}{(T_{exhaust,in} - T_{air,in})} \tag{3.3}$$

where the temperatures with the subscript ($_{air}$), indicate the (in)let and (out)let temperatures of the cold, compressed air. In the same manner, the subscript ($_{exhaust}$), indicates the corresponding temperatures of the hot gases exiting the gas turbine (in this case the LM2500). For the intercoolers, the effectiveness is given a coresponding equation:

$$\epsilon_{intercooler} = \frac{(T_{air,in} - T_{air,out})}{(T_{air,in} - T_{water,in})} \tag{3.4}$$

For the analysis of the ABC, the effectiveness of the intercoolers is 77.1%, while the effectiveness of the heater is 91.1%.

Table 3.2 summarizes all the assumed values of the key parameters used in the analysis of the ABC, as well as the initial conditions.

The need for size minimization, in terms of both volume and weight, that is required by any application developed for a Naval Combatant, led to the optimization of the heater's size without any sacrifice in performance. Campbell [6], in 1989, worked on the Soland modification of the Keys and London way of presenting heat exchanger performance. This new method minimizes the volume of the core of a plate-finned heat exchanger. A detailed description of this method, that led to the final design of the heater, is presented in the following chapter.

Compactness is one of the major reasons that made radial machines attractive for the design of the ABC-engine in this thesis. Radial-flow compressors can be designed for a higher per stage pressure ratio, enthalpy rise or size of head developed, than their equivalent axial machines. Nevertheless, the efficiency of radial-flow machines is relatively lower than that of an equivalent axial-flow machine. Flow separation, resulting from the compex three-dimensional flow in the rotor, the fact that much of the enthalpy rise can be attributed to the velocity head, relatively low radial-diffuser pressure-rise coefficients, and higher friction developed from the larger wetted surfaces are the four dominant reasons for the difference in the efficiency between radial and axial machines.

Symbol	Compressors	Turbines
η_{pc}	0.88	0.92
	Intercoolers	Heater
ϵ	0.771	0.911
	Air In	Water In
T (K)	288.15	288.15
P (kPa)	100.33	

Table 3.2: Values of key parameters, and initial conditions as used in the thermodynamic analysis of the Air-Bottoming Cycle.

One other factor that contributed to the selection of radial components for the design of the Air-Bottoming Cycle is the required ducting necessary to collect and pass the flow to the intercoolers and the heater. Radial machines, have already incorporated in their design the flow collection and carry the appropriate ducting. While this complication in the design of radial machines may seem to be disadvantegeous (due to the pressure losses) and cost ineffective for any other application, in the case of this intercooled-regenerative cycle it turns out to be very attractive. The additional ducting and flow collecting components for an axial machine will tend to increase both size and cost further. It should be clear that the requirement of lower volume, weight and cost place some very rigid design constraints.

Finally, the fact that William Farrell was approaching the design of the ABC engine using axial components for industrial applications, made the design of a radial version of the engine attractive for comparison.

Thus, all the key parameters necessary for the thermodynamic analysis of the cycle have been determined. In the following section the Air-Bottoming Cycle's analysis results are presented with a simple sensitivity study that will support the reasoning behind the selection of the engine's design point.

3.3 Air-Bottoming Cycle Analysis Results

Three computer programs were developed in order to model and solve both the thermodynamics and the design of the Air-Bottoming Cycle. The programs are kept simple and are based on the theory and design principles outlined in the previous sections, as well as in the following chapter. All programs are interactive in the sense that they give the user the freedom to define most key parameters. Many con-

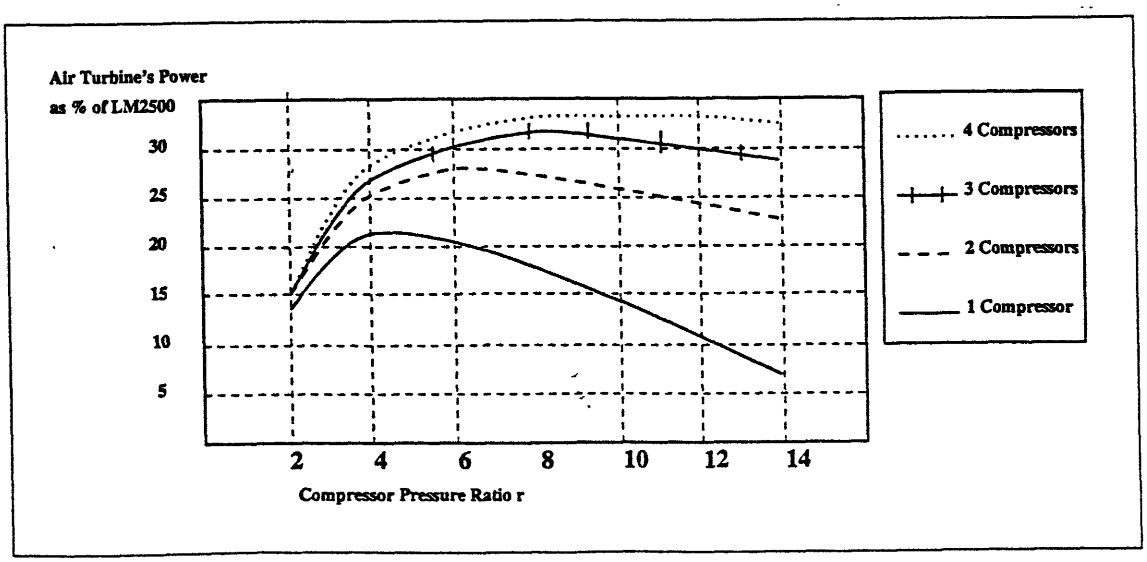


Figure 3-4: Air-Bottoming Cycle sensitivity, of percentage of LM2500 power generated, on compression ratio.

figurations can be investigated and the new air cycle can consist of various number of compressors, intercoolers and turbines with different performance characteristics. In this manner, one compressor with a compression ratio of 8:1, can be substituted by three compressors each having a compression ratio of 2:1. The selection of the optimum configuration and design point was performed in such an iterative fashion. The sensitivity of the performance of the ABC to parameters like the number of the compressors and their individual compression ratios is shown in Figures 3-4 and 3-5. Furthermore, the user/designer can change the inlet conditions to the heater model, which means changing the source of the hot exhaust gas. The air side of the Air-Bottoming Cycle can be re-designed and coupled to various gas-turbines. Thus, many more potential applications of the new cycle can be found, since not all possible applications use the General Electric LM2500 gas turbine.

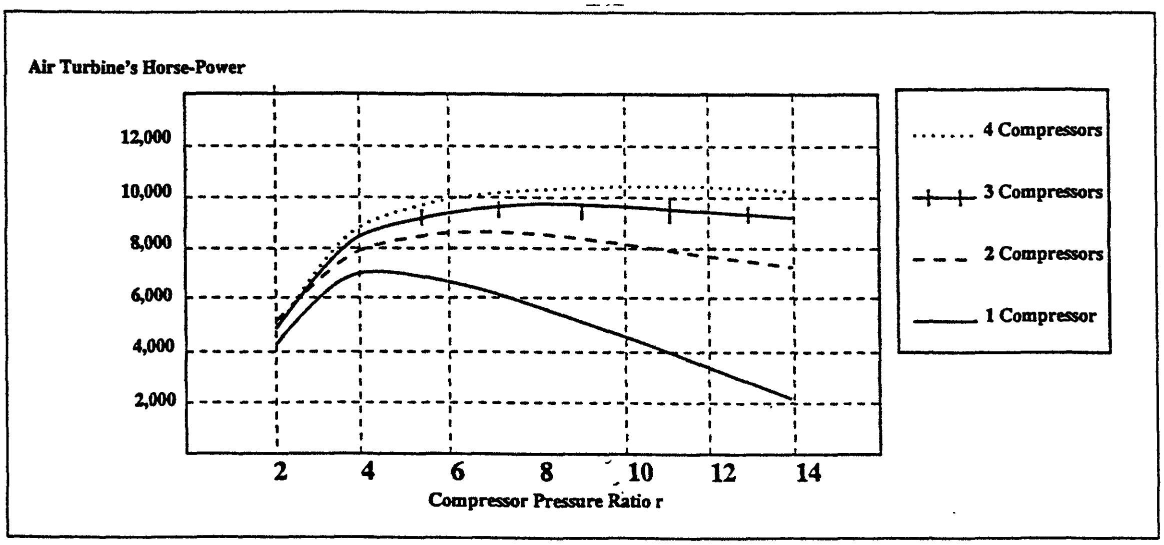


Figure 3-5: Air-Bottoming Cycle Horsepower generated at different compression ratios.

Two of the computer programs used for the analysis of the Air-Bottoming Cycle (ABC) are concerned with the design parameters and will be addressed after the design principles have been outlined in the next chapter. The first program provides the thermodynamic analysis. The model can run with either the desired temperatures of each stage, or the pressure ratios being known. Although only the second option of the code is the one used for the purpose of this thesis, freedom to repeat the overall process in a much easier and faster way was the philosophy behind the development of the code. Whatever path the user decides to follow, the program will use the input parameters to determine the remaining design conditions that describe each stage sufficiently. The program will run for as many compressor stages as specified in the thermodynamics modeling. For the purpose of this thesis, a variation of both the number of compressors in the air-cycle, and the compression ratio of each, was analyzed, in order to decide the final cycle configuration. A configuration of three compressors at a pressure ratio of 2.0 each, is a valid trade-off design point. Although slightly better performance could be achieved with four compressors with an overall pressure ratio of ten, Figure 3-4 shows that there would be an small gain of an additional one or two percent in horsepower. While the horsepower of the selected design point is 9,700 HP, the horsepower of the more complicated engine would only rise to 10,400 HP. Such an engine would be much bigger, heavier and more expensive. Figures 3-4 and 3-5, present the sensitivity study that led to the design-point selection with a configuration of three compressors at a 2:1 compression ratio each. As shown in both graphs, the air-cycle's performance, when coupled to an LM2500 operating at design point, is close to the maximum. As shown in Table 3.3, the initial conditions are $T_1 = 288.15$ K and $p_1 = 100.33$ kPa. The air, with a mass flow rate of 69.8 Kg/sec, after passing through the cycle will deliver 9,698.2 HP.

LP Compressor	Inlet Conditions	Exit Conditions
T (K)	288.15	359.5
P (kPa)	100.33	200.66
Air-Flow Rates	69.8 kg/sec	119,777 cfm
Pressure Ratio r		2:1
1_{st} Intercooler	Inlet Conditions	Exit Conditions
T (K)	359.5	304.5
P (kPa)	200.66	196.7
MP Compressor	Inlet Conditions	Exit Conditions
T (K)	304.5	379.8
P (kPa)	196.7	393.3
Mass Flow Rate	69.8 kg/sec	59,888.5 cfm
Pressure Ratio r		2:1
2_{nd} Intercooler	Inlet Conditions	Exit Conditions
T (K)	379.8	309.1
P (kPa)	393.3	385.4
HP Compressor	Inlet Conditions	Exit Conditions
T (K)	309.1	385.6
P (kPa)	385.4	770.9
Mass Flow Rate	69.8 kg/sec	29,944.25 cfm
Pressure Ratio r		2:1
Heater	Inlet Conditions	Exit Conditions
T_{Air} (K)	385.6	797.8
P_{Air} (kPa)	770.9	740.3
HP Turbine	Inlet Conditions	Exit Conditions
T (K)	797.8	720.8
P (kPa)	740.3	542.4
MP Turbine	Inlet Conditions	Exit Conditions
T (K)	720.8	645.
P (kPa)	542.4	385.7
LP Turbine	Inlet Conditions	Exit Conditions
T (K)	645.	573.3
P (kPa)	385.7	268.7
Power Turbine	Inlet Conditions	Exit Conditions
T (K)	573.3	485.3
P (kPa)	268.7	101.2
Power Turbine's Power Generated	9,698.2 HP	

Table 3.3: Design point conditions and configuration parameters.

3.4 Off-Design Performance

Power-plants aboard Naval combatants are usually selected based on a maximum horsepower requirement. Nevertheless, surface combatants spend most of their cruising time at power levels significantly lower than maximum. The representative operating profile of a DDG-51, is shown in Figure 3-6 [7]. The operating profile indicates the need for an engine that not only can meet the maximum speed requirement, but can also perform efficiently at lower speeds. As shown in Figure 3-7, the engines

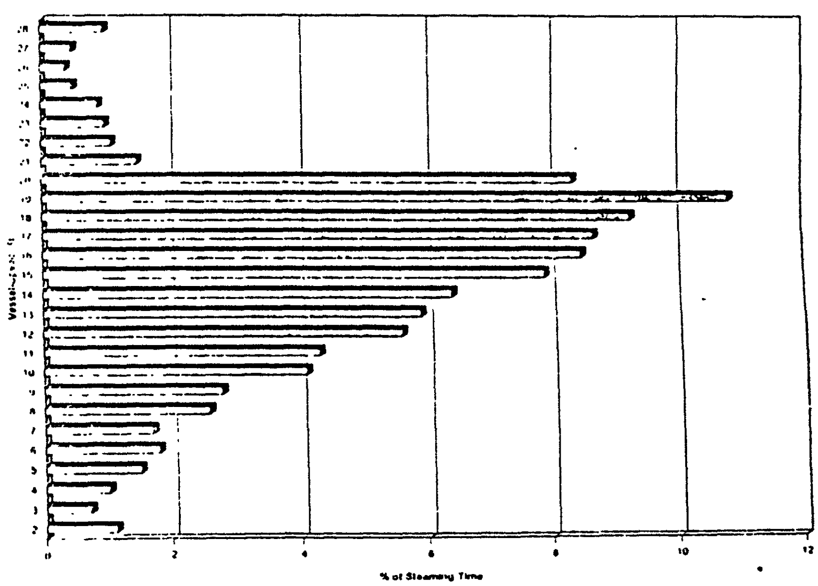


Figure 3-6: Typical U.S. Navy Destroyer Operating Profile. The figure demonstratesthe percentage of time spent by a DDG-51 destroyer at various speeds.

aboard a surface combatant operate a very broad power range. A significant drawback of gas turbines in Navy ships is their poor off-design performance, increasing their already high, operating cost. Although several modifications have improved the simple-cycle's performance, gas-turbine technology has matured so much, that it may not be cost effective to improve a simple-cycle engine by either using exotic materials or advanced manufacturing methods. Still, gas turbines perform poorly, when operating at off-design power levels Figure 3-8.

Rerouting the flow is the only way that can bring improvements to the performance of a simple gas turbine at a reasonable cost. Cost considerations will be addressed in a later chapter of this thesis. Figure 3-8 shows that by using the exhaust gas of an LM2500 in the Air-Bottoming Cycle, the off-design performance of the combined cycle is considerably better. As Figure 3-9 shows, the off-design

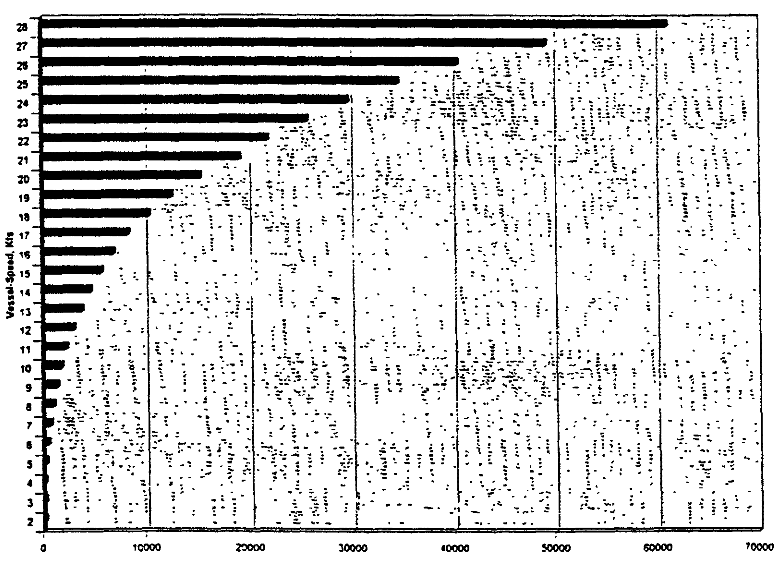


Figure 3-7: Typical U.S. Navy Destroyer Power Requirement at Different Speeds.

performance of the new cycle [8] demonstrates even greater contributions to the power of the LM2500. Especially at the lower rating levels, where gas turbines run very inefficiently, there is significant improvement in the power generated. There is 100% additional power provided by the air-turbine when the LM2500 generates 1,000 HP. At the 3,000 HP level of the LM2500, the ABC generates 1,554 HP or an additional 50%. The improvement becomes almost constant after the 10,000 HP

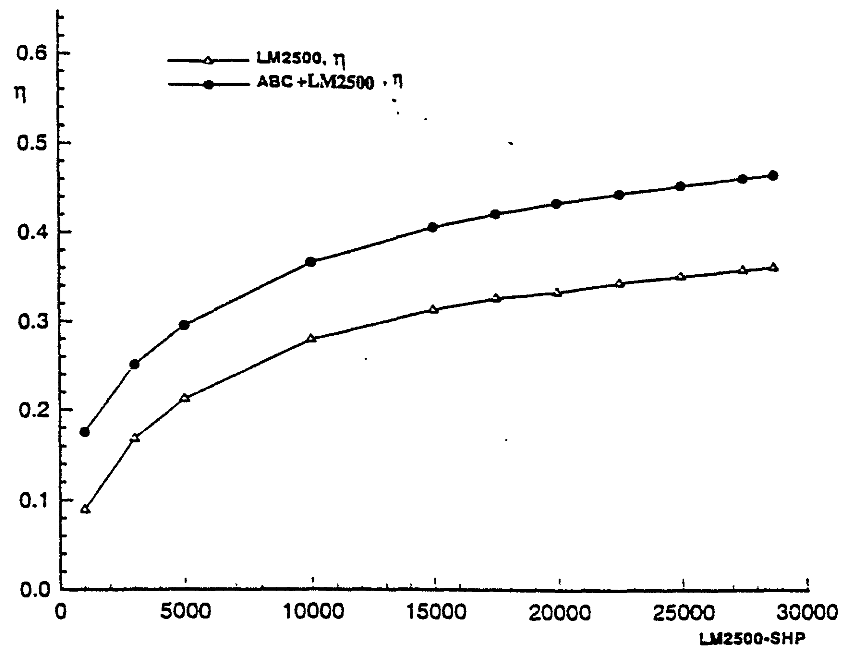


Figure 3-8: Air-Bottoming Cycle Efficiency Improvement to the LM2500.

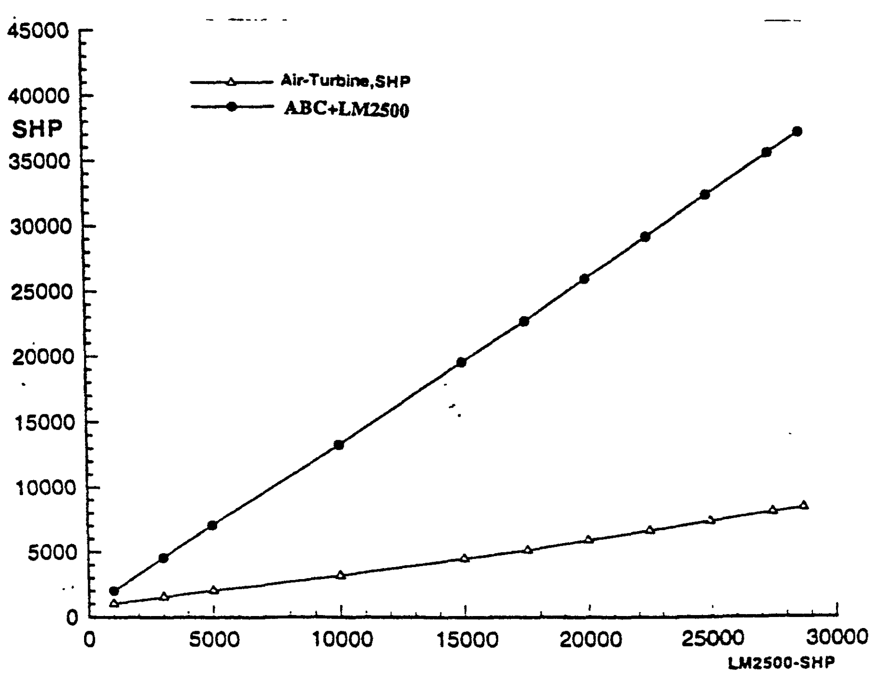


Figure 3-9: Air-Bottoming Cycle Off-Design Performance.

rating of the LM2500, with an average of 32% additional power. Thus, the operat-

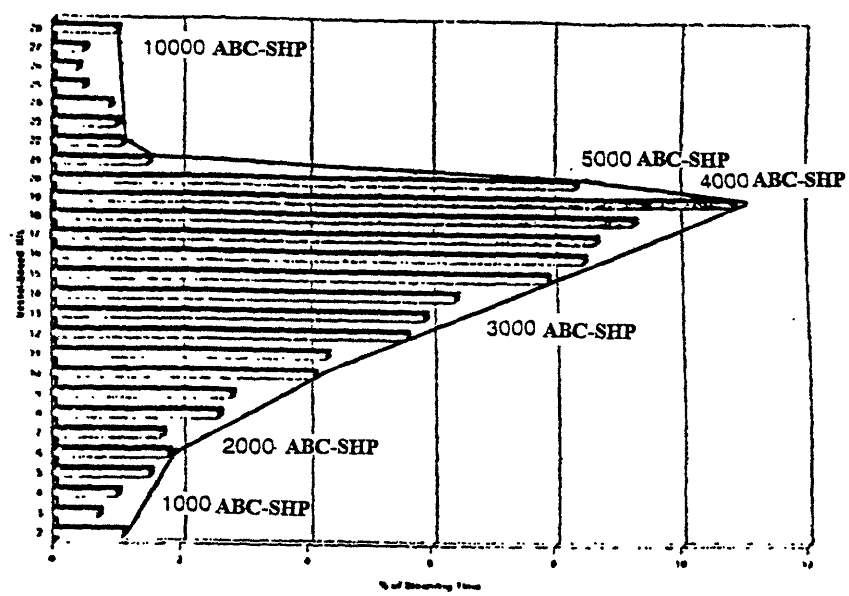


Figure 3-10: The ABC's operation, fit in a DDG-51 operating profile. There is a significant contribution in the overall power from the air-turbine.

ing profile of the LM2500 will change significantly as shown in Figure 3-10, where the power levels of the ABC are shown against the operating speed of a U.S. Navy DDG-51.

From Figure 3-11 it can be seen that the air-turbine's SHP contribution to the overall ABC's shaft-horsepower is significant. The power required by the LM2500 is now lowered, and can be described as a function of the total power generated by the combined (ABC) cycle as follows:

$$SHP_{(LM2500)} = 0.78904 \cdot SHP_{(LM2500+ABC)} - 514.134 \tag{3.5}$$

This relationship gives the magnitude of the power improvement, as the LM2500's SHP contribution to the overall combined cycle's SHP, at the various horsepower requirements. The significance of such a change in power generated by the LM2500 is demonstrated in Figure 3-10, from where it is clear that for a large amount of the cruising time of a surface combatant, the air-turbine will contribute a major part of the overall power. There are similar improvements in the efficiency of the new combined-cycle engine. Both in trends and in magnitude, the efficiency of the Air-Bottoming cycle improves the performance of the new combined engine. Improvement ranges between 97% at the lower LM2500 ratings to 29% at 29,000 HP. The new efficiency curve, as shown in Figure 3-8, ranges between 18% and 47%.

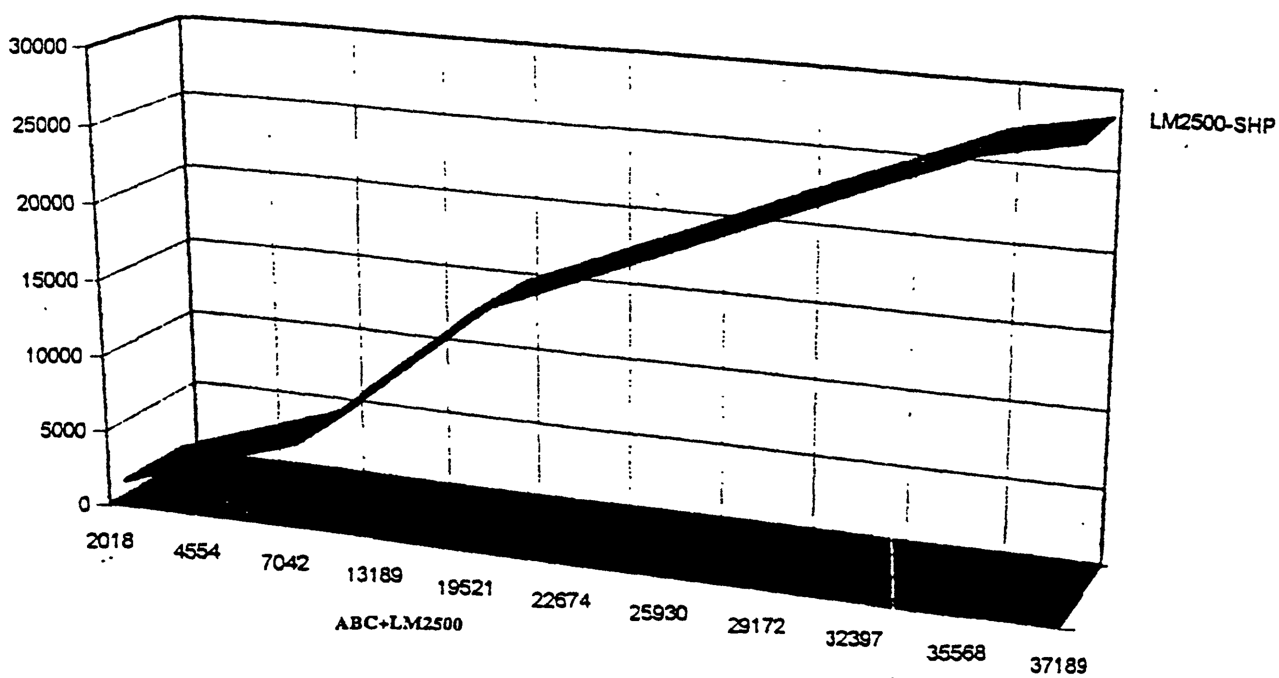


Figure 3-11: LM2500 Power Supplied as a Function of the Total Power Generated by the Combined ABC.

In the next chapter the design principles will incorporate the thermodynamics analysis results, in order to generate a design of the new components of the Air-Bottoming Cycle (ABC) and the results of the computer-code, modeling the design, will appear in the final section.

Chapter 4

Design Process

4.1 Introduction

The Air-Bottoming Cycle (ABC) was described in the previous chapter and the selected cycle had three compressors of 2:1 pressure ratio each. Also, between each pair of adjacent compressors, intercoolers reduce the temperature of the compressed air. The compressed air exiting the third compressor enters a heater, where the compressed air is heated from the waste heat recovered from thew exhaust gas of an LM2500. The air leaving the heater passes through three turbines, each one coupled to a corresponding compressor, before it expands in a power turbine which delivers the power from the ABC. The design goal is to optimize each component (stage) of the new cycle for maximum output and efficiency and minimum cost. Understanding these constraints and knowing the conditions of the working fluids at all stages, one can move toward a preliminary design of the components of the Air-Bottoming Cycle.

In this chapter the goal is to match the engine-components to the operating conditions and the power requirements determined by the thermodynamic analysis of the cycle. There were two areas that the design focused on:

- The preliminary design of all rotating parts in two groups of components:
  - The design of the compressors.
  - The design of the turbines.
- The design of the heat exchangers:
  - The design of the intercoolers.
  - The design of the heater.

A new engine is attractive for production, not only when it generates significant performance improvement compared to the existing ones, but also when it is cost effective compared to the competition. Both performance and cost characteristics have to be taken into consideration, in order to evaluate the potential of the combined gas turbine and Air-Bottoming Cycle to replace simple gas turbines aboard Navy vessels, such as the LM2500, and to compete with other engines under development, such as the ICR engine. The performance characteristics were presented in the previous chapter, while cost will be addressed in the following one. This chapter serves as the transition stage from the performance characteristics of the air-turbine to the cost evaluation of the combined gas turbine engine and Air-Bottoming Cycle through the preliminary design of the air-turbine's components. The design-characteristics of the componentsof the new engine will facilitate the estimation of the acquisition cost of the air-turbine and thus the combined ABC, while the performance characteristics will be used to address the issue of operating and life-cycle costs.

The input parameters to the design process have been partially determined in the thermodynamic analysis as it was shown in Table 3.3. Table 4.1 summarizes the setup parameters of this design. Each component is designed to match and perform according to these values and some key assumptions, which are being outlined in

Table 4.2. Radial components are used throughout this design, for the reasons outlined in the foregoing chapter.

Configuration Parameters			
Number of Compressors	3	Stage Pressure Ratio r	2:1
Number of Intercoolers	2	Intercooler Effectiveness ϵ	0.771
Number of Heaters	1	Heater Effectiveness ϵ	0.911
Ambient-Air T (K)	288.15	Gas-Turbine Exhaust Gas T (K)	838.1

Table 4.1: Input Parameters to the Design Process, determined in the Thermodynamic Analysis of the ABC. The hot gas conditions used here were the exit conditions of the exhaust gas exiting an LM2500.

4.2 The Design of the Rotating Components

In the Air-Bottoming Cycle the use of the intercoolers and the heater require the collection of the flow after it exits each compressor. The advantage of using radial components versus using their axial equivalent is that the former collect the flow in their non-rotating parts, while the latter need additional ducting which in turn produce further losses.

The preliminary design of all rotating parts of the Air-Bottoming Cycle was therefore based on design principles of radial turbomachines. The methodology used, breaks the compression and the expansion processes into smaller stages corresponding to the various parts of the compressor or turbine. Figure 4-1 shows in a schematic form the stages through which the air passes during the compression process. First it enters the first impeller at ambient conditions (Stage 1). Before the design of the impeller could start, the rotational speed will have to be determined.

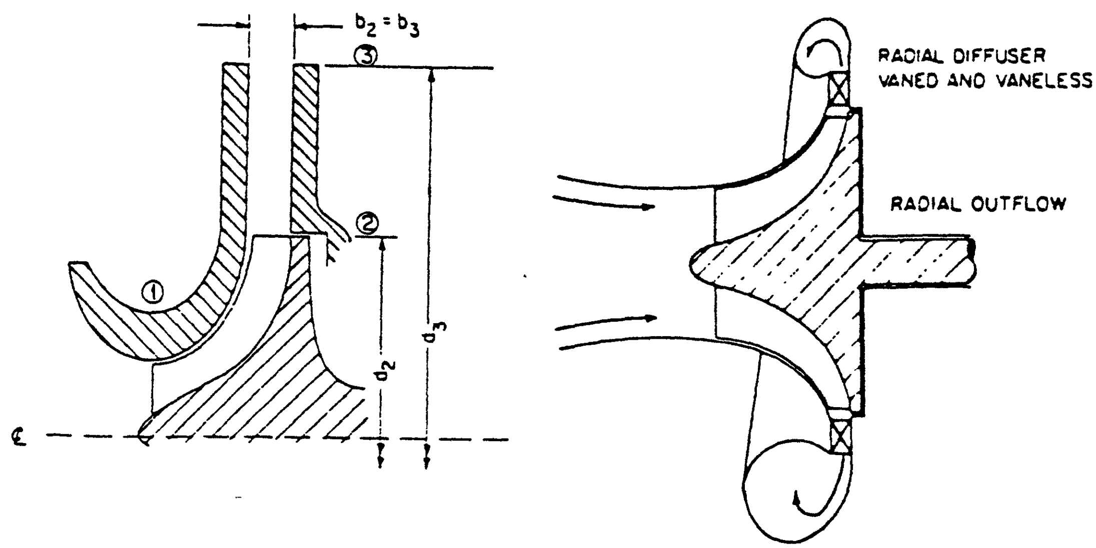


Figure 4-1: Schematic Diagram of a Compressor's Stages.

The rotational speed of the impeller can be calculated by assuming a value of the specific speed. Specific speed is a non diamensional group based on the rotational speed, the volumetric flow rate and the enthalpy rise developed. The volumetric flow rate can be determined from the mass flow rate:

$$Q = \frac{m}{\rho_{o1}} \tag{4.1}$$

For the purpose of this design, the mass flow rate was assumed to be 69.8 $\frac{kg}{sec}$ as determined in Chapter 3. It should be noted that the subscript ($_o$) indicates stagnation conditions. Specific speed for radial turbomachinary is usually between 0.08 and 0.17. The selection of the appropriate value for the specific speed of a machine is important, since there is a dependence of the machine's efficiency on the specific speed of the impeller. In this design, the specific speed of the compressor was selected to be 0.12. Thus, by using the following equation the rotational speed of each stage compressor (turbine) can be calculated:

$$N = \frac{N_s \cdot 60 \cdot (h_{o2} - h_{o1})^{0.75}}{Q^{0.5}} \tag{4.2}$$

The selection of specific speed in this case is a compromise between N_s for the compressor and the turbine on the same shaft. In addition, the same specific speed was selected for the three compressors.

Table 4.2 shows all the parameters used in the design of the turbomachinery for the Air-Bottoming Cycle.

The approximate dimension of the complete compressor, i.e. maximum diameter, was estimated to be twice the impeller diameter.

Symbol	Compressor		Turbine
Mass-Flow Rate (Kg/sec)	69.8		69.8
Specific Speed N_s	0.08		Calculated 0.10-0.15
Polytropic Efficiency η_{PC}	0.88		0.92
Impeller-Blades Z_C	12		18
Isentropic Efficiency η_I	0.87		0.92

Table 4.2: Assumed Parameters used in the Design Process.

4.3 The Design of the Heat Exchangers

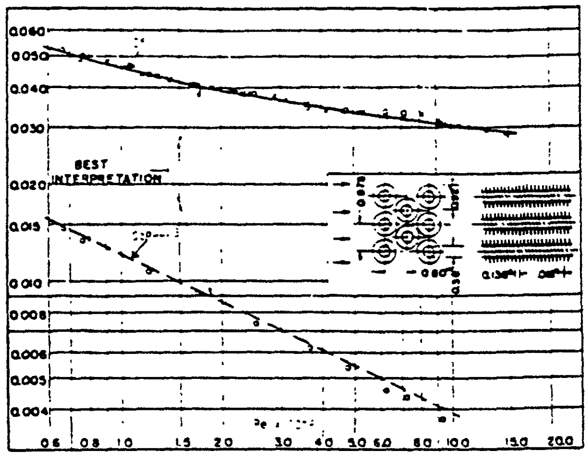
For the design of the intercoolers the NTU method of designing one- pass cross-flow heat exchangers was employed. The intercoolers were treated as finned-tube heat exchangers with water flowing on the cold side and hot, compressed air on the hot side. After determining the inlet and exit temperatures, the Reynold's and Stanton numbers need to be calculated. The friction factors now can be found, which in turn are used to determine the the heat transfer coefficients. Then, the overall heat transfer coefficient can be calculated. To find the heat exchange areas of the intercoolers, Tables 4.4 and 4.5 were used for the air and the water side respectively. The j and f factors were found from Figure 4-4, while Figure 4-3 was used to get the value of NTU. In order to incorporate the appearance of the fins, Figure 4-2 was used for the correction of the overall heat transfer coefficient. The constants and factors used in this design were assumed at mean fluid conditions. The values of the key input parameters to the design of the intercoolers was presented in early tables, while Table 4.6 shows all the resulting parameters and the intercoolers' physical dimensions.

For the design of the heater an attempt to apply the same principles of the NTU method led to a design that was not attractive, due to excessive volume and weight. The final design process for the heat exchanger was based on the Soland modification of the Keys and London way of presenting heat exchanger performance. It is an optimization of volume approach. This new method minimizes the volume of the core of a plate- finned heat exchanger. Initially the selection of the two surfaces for the hot and the cold sides of the heat exchanger describes the performance of the materials used. The selection process is based on the minimum plate-spacing commensurate for the high pressure side, and on the minimum hydraulic diameter for the use of the low pressure side. The heat exchanger size can be defined in terms of the allowable pressure drops as,

$$P = \frac{(X \cdot Y)^{(2-s)}}{Z} \tag{4.3}$$

where X, Y and Z are the physical dimensions of the heat exchanger, while P is the pressure drop. Also, the core size can be expressed in terms of heat transfer,

$$Q = \frac{(X \cdot Y)^{s}}{Z}. \tag{4.4}$$



Tube outside diameter = 0.38 in.
Fin pitch = 7.34 per in.
Flow passage hydraulic diameter, = 0.0154 ft
Fin thickness (average)* = 0.018 in., aluminum
Free-flow area/frontal area, = 0.538
Heat transfer area/total volume, = 140 ft^2/ft^3
Fin area/total area = 0.892

Figure 4-2: The effect of finned tubes on the overall heat-transfer coefficient used in the design of the intercoolers.

where Q is the heat transfer and s is the average slope of all best fit lines through j_n and f_n vs. Re_n data points like the one appearing in Figure 4-5. The equations 4.3 and 4.4 when solved as a system of two equations for the two unknown quantities $(X \cdot Y)$ and Z, yield the closed-form solution for the heat exchanger core volume:

$$V = P^{(\frac{1-s}{2})} \cdot Q^{(\frac{3-s}{2})} \tag{4.5}$$

Equations (4.3), (4.4) and (4.5) suggest that given a set of operating conditions and with one surface specified and thus its material's properties, then the volume of the heat exchanger can be expressed as:

$$V = f[\frac{b_2}{b_1}, \frac{K_{f2}}{K_{f1}}, [\frac{K_{j2}}{K_{j1}}]^{-1}, \text{operating conditions}] \tag{4.6}$$

where the first ratio in equation 4.6 is the ratio of plate spacing at surfaces one and two, while the ratios $\frac{K_{f2}}{K_{f1}}$ and $\frac{K_{j2}}{K_{j1}}$ are ratios of y-intercepts when fitting straight

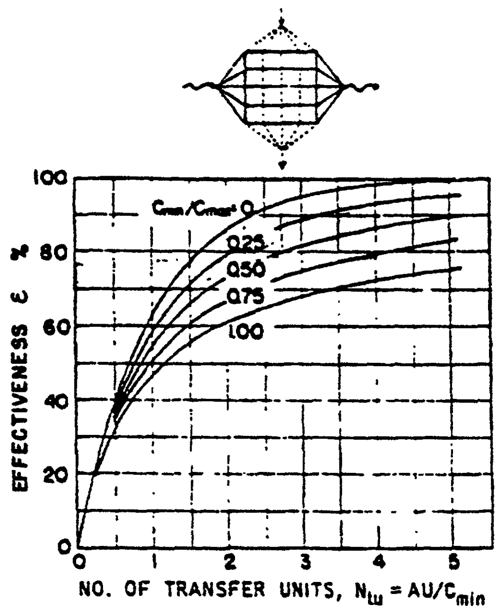
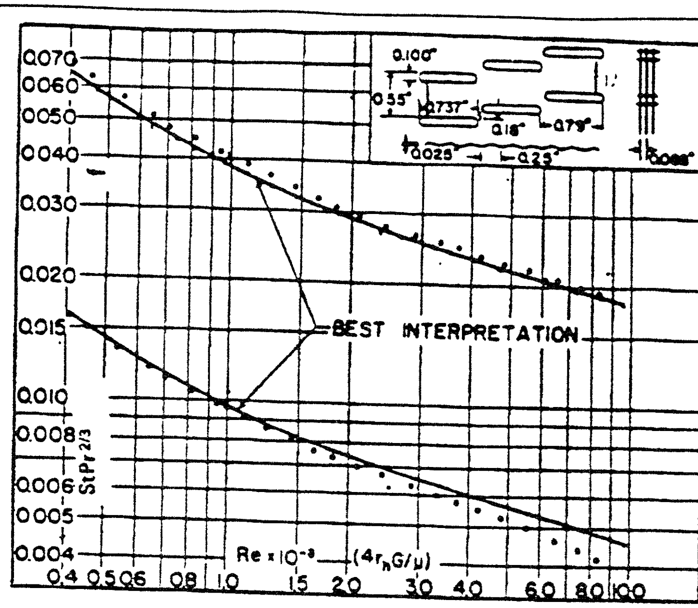


Figure 4-3: Heat transfer effectiveness as a function of NTU and capacity-rate ratio for crossflow exchanger with fluids unmixed.

lines through the modified Keys and London data points, j_n and f_n vs. R_{e_n} as found in Campbell [6]. These values are listed in Table 4.3 for selected surfaces. Volumes in this table are grouped by nominal ratio $\frac{b_2}{b_1}$ and then lines of constant $\frac{K_{j2}}{K_{j1}}$ are plotted as function of $\frac{K_{f2}}{K_{f1}}$. From these figures, [6] figures 6a-6e, one can select the surfaces that provide the minimum volume for the operating conditions, which is achieved when $\frac{K_{j2}}{K_{j1}}$ and $\frac{K_{f2}}{K_{f1}}$ are minimized and the ratio $\frac{b_2}{b_1}$ gets closer to a value of one. Table 4.7 shows all the assumed values of the key parameters used for the design of the heater, as well as the resulting heater's physical characteristics.



Fin pitch = 11.32 per in = 446 per m
Flow passage hydraulic diameter, $4r_h$ = 0.01152 ft = 3.510×10^{-3} m
Fin metal thickness = 0.004 in, copper = 0.102×10^{-3} m
Free-flow area/frontal area, σ = 0.780
Total heat transfer area/total volume, α = 270 ft²/ft³ = 886 m²/m³
Fin area/total area = 0.845

Figure 4-4: Finned flat tubes, surface 11.32-0.737-SR.

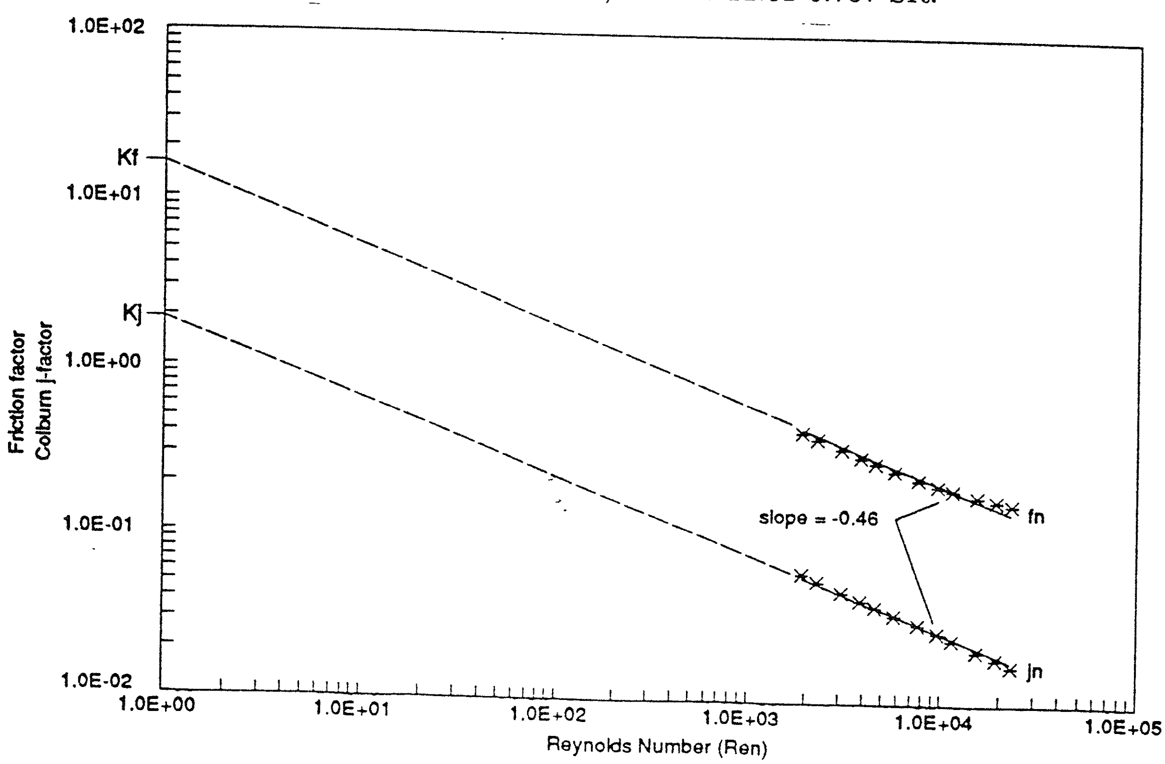


Figure 4-5: "New Best-Fit" Line of Surface 3/16-11.1 with Slope s=0.46 was Used for the Design of the Heater.

Side One: 3/16-11.1					
Surface Designation	$4r_h$ (ft)	Volume (ft^3)	b_2/b_1	K_{h2}/K_{h1}	K_{f2}/K_{f1}
1/10-19.74	0.00400	436.7	0.2	0.35	0.27
1/10-19.35	0.00460	356.4	0.3	0.57	0.36
1/9-24.12	0.00397	375.8	0.3	0.70	0.59
3/8(b)-11.1	0.01012	377.9	1.0	0.93	0.78
3/4-11.1	0.01012	400.5	1.0	0.75	0.55
3/8-11.1	0.01012	376.2	1.0	0.95	0.80
3/16-11.1	0.01012	385.8	1.0	1.00	1.00
11.1	0.01012	460.7	1.0	0.50	0.28
1/2-11.1	0.01012	394.1	1.0	0.81	0.64
3/4(b)-11.1	0.01012	404.0	1.0	0.74	0.55
1/8-16.12T	0.00514	362.4	1.3	2.37	3.56
14.77	0.00848	439.1	1.3	0.86	0.70
30.33T	0.00401	322.1	1.4	2.30	2.03
1/6-12.18D	0.00885	374.4	1.4	1.42	1.27
6.2	0.01820	809.1	1.6	0.37	0.21
17.8-3/8W	0.00696	394.7	1.7	2.16	3.09
11.44-3/8W	0.0106	435.3	1.7	1.61	2.31
1/8-15.2	0.00868	429.0	1.7	1.91	3.22
15.08	0.00876	502.2	1.7	0.77	0.48
5.3	0.02016	648.5	1.9	0.53	0.22
11.11(a)	0.01153	439.2	1.9	1.09	0.60
3/32-12.22	0.01120	509.3	1.9	1.55	2.78
3.97	0.02820	916.8	3.0	0.68	0.38
2.0	0.04740	1321.9	3.0	0.42	0.18
3.01	0.03546	1119.3	3.0	0.54	0.30
9.03	0.01522	942.0	3.3	0.87	0.82

Table 4.3: Values for the coefficients in equation 4.43 for side one of surface 3/16-11

4.4 Methodology and Results

4.4.1 Methodology

As a result of the sensitivity analysis of the cycle, to both the number of the compressors and their pressure ratio, a configuration of three compressors having intercoolers between them (a total of two intercoolers are used), a heat exchanger after the third compressor and three turbine stages coupled to the corresponding compressors, followed by a power turbine stage delivering the power, was selected. As

j_h vs. Re_h						
Surface name from K-L [1]	Name used in Fig. 6	Plate Spacing, b	Nominal Plate Spacing	slope	r	K_j
1/10-19.74	S27	0.0042		-0.440	0.995	0.656
1/10-19.35	S29	0.0063	0.006	-0.484	0.999	1.069
1/9-24.12	S28	0.0063		-0.434	0.993	1.312
3/8(b)-11.1	L19	0.0208		-0.486	0.999	1.746
3/4-11.1	L21	0.0208		-0.484	0.999	1.396
3/8-11.1	L18	0.0208		-0.506	0.999	1.773
3/16-11.1	L15	0.0208	0.021	-0.497	0.999	1.872
11.1	P04	0.0208		-0.395	0.983	0.932
1/2-11.1	L20	0.0208		-0.473	0.997	1.514
3/4(b)-11.1	L22	0.0208		-0.459	0.999	1.380
1/8-16.12T	S31	0.0261		-0.607	0.998	4.440
14.77	P06	0.0275	0.028	-0.503	0.996	1.615
30.33T	P10	0.0287		-0.790	0.999	4.314
1/6-12.18D	S30	0.0294		-0.641	0.999	2.656
6.2	P02	0.0337		-0.306	0.938	0.686
17.8-3/8W	W27	0.0345		-0.612	0.999	4.046
11.44-3/8W	W26	0.0345	0.034	-0.549	0.999	3.014
1/8-15.2	S25	0.0346		-0.489	0.999	3.575
15.08	P07	0.0348		-0.621	0.986	1.447
5.3	P01	0.0392		-0.403	0.996	0.987
11.11(a)	P05	0.0400	0.040	-0.396	0.936	2.046
3/32-12.22	S24	0.0404		-0.584	0.999	2.899
3.97	P14	0.0625		-0.331	0.989	1.274
2.0	P12	0.0625	0.064	-0.313	0.997	0.790
3.01	P13	0.0625		-0.339	0.987	1.007
9.03	P03	0.0686		-0.537	0.987	1.625

Note: The intercept K_j is determined for the average slope equal to -0.46.

Table 4.4: Reproduced from Table 2 of Campbell [6] incorporating Table A-1 of Kays and London.

mentioned in a previous chapter, three computer programs were developed in order

f_s vs. Re_s						
Surface name from K-L [1]	Name used in Fig. 6	Plate Spacing, b	Nominal Plate Spacing	slope	r	K_f
1/10-19.74	S27	0.0042		-0.457	0.977	3.919
1/10-19.35	S29	0.0063	0.006	-0.522	0.992	5.265
1/9-24.12	S28	0.0063		-0.493	0.988	8.737
3/8(b)-11.1	L19	0.0208		-0.376	0.980	11.561
3/4-11.1	L21	0.0208		-0.454	0.982	8.060
3/8-11.1	L18	0.0208		-0.388	0.982	11.806
3/16-11.1	L15	0.0208	0.021	-0.395	0.995	14.766
11.1	P04	0.0208		-0.441	0.960	4.141
1/2-11.1	L20	0.0208		-0.375	0.971	9.436
3/4(b)-11.1	L22	0.0208		-0.429	0.982	8.068
1/8-16.12T	S31	0.0261		-0.378	0.965	52.575
14.77	P06	0.0275	0.028	-0.493	0.972	10.369
30.33T	P10	0.0287		-0.807	0.991	29.905
1/6-12.18D	S30	0.0294		-0.498	0.985	18.694
6.2	P02	0.0337		-0.371	0.943	3.092
17.8-3/8W	W27	0.0345		-0.435	0.999	45.642
11.44-3/8W	W26	0.0345	0.034	-0.391	1.000	34.107
1/8-15.2	S25	0.0346		-0.335	0.969	47.617
15.08	P07	0.0348		-0.640	0.969	7.032
5.3	P01	0.0392		-0.458	0.972	3.226
11.11(a)	P05	0.0400	0.040	-0.516	0.959	8.833
3/32-12.22	S24	0.0404		-0.385	0.980	41.038
3.97	P14	0.0625		-0.260	0.969	5.607
2.0	P12	0.0625	0.064	-0.212	0.976	2.648
3.01	P13	0.0625		-0.264	0.968	4.448
9.03	P03	0.0686		-0.418	0.946	12.057

Note: The intercept K_f is determined for the average slope equal to -0.46.

Table 4.5: Reproduced from Table 3 of Campbell [6] incorporating Table A-8 of Kays and London.

LP Intercooler			
Symbol	Air Side		Water Side
Re	9,812		5,080
f	0.019		0.038
h_t	319.5		7,540
m	192.2		20,431.3
U_w	187.2		795.5
NTU	*2.5*		
Heat-Transfer Area	928.3		218.4
Height	3.8	Length	0.42
Width	0.65	Number of Tubes	1,482
HP Intercooler			
Symbol	Air Side		Water Side
Re	9,544		5,080
f	0.019		0.038
h_t	325.3		7,540
m	193.9		20,431.3
U_w	206.0		795.5
NTU	*2.5*		
Heat-Transfer Area	854.0		218.4
Height	3.8	Length	0.42
Width	0.60	Number of Tubes	1,482

Table 4.6: The Resulting Parameters, and Physical Characteristics of the Intercoolers.

to model the Air-Bottoming Cycle (ABC). The first program takes care of all the thermodynamics. After all the characteristic parameters of the cycle have been determined, the second program is used. This is a tool for the preliminary design of a centrifugal compressor. The program assumes an initial condition, and proceeds iteratively to calculate the value of the specific variable. The design process is divided to five different sub-stages, as many as the individual parts of the compressor. The design model, deals with each such stage separately, and moves to the next stage after the previous stage has been configured. Both vector diagrams and geometric parameters are determined. At the end of this routine the shape characteristics and the performance parameters should be known. The program will run for as many compressor stages as specified in the thermodynamic modeling.

Between two compressors there are intercoolers that are used to lower the temperature of the compressed hot air and so, to make the following compression stage more efficient. A spreadsheet, that was used for the design of the intercoolers, used input values from the exit characteristics of the flow from each compressor. The regenerator was treated in a similar manner.

The third program is a derivative of the second. It is a tool that this time is used for the design of the turbine. The original program is switched, and treats the turbine as an inverted compressor. In other words the program is converted to design a compressor operatating with the fluid entering at the exit conditions of the turbine, while the compressed fluid would exit the compressor at the inlet conditions of the fluid entering the turbine.

4.4.2 Results

The initial conditions are $T_1 = 288.15$ K and $p_1 = 100.33$ kPa. The first compressor has shroud radius of 0.42 m, and an outer impeller radius of 0.65 m. The compressor runs at 4,152 RPM. The air then enters the first intercooler at a speed of 52.8 m/s, with the same mass flow rate of 69.8 kg/s. In order to decrease the air's temperature from 359.48 K to 304.48 K, the intercooler is designed to be 3.81m in length, 0.65m in width and 0.42m in height, with the air running through the 0.65 m side and the water through the 3.81 m. The core of the intercooler will consist of 1,482 fined-tubes of $11.32 - 737 - SR$ surface. The second compressor, runs at 5,890 RPM. Its outer impeller radius is reduced to 0.47 m, while its shroud radius is 0.31 m. The exit temperature is now 379.8 K, with the air running at a speed of 54.0 m/s, compressed at a pressure of 393.3 kPa. The second intercooler is going to cool the air to 309.0 K. It consists of the same core as the first one. Its outer dimensions change to 3.81m in length, 0.6m in width and 0.42m in height. The high

Heater's Parameters			
Symbol	Air Side		Waste-Gas Side
Surface	1/9-24.12		1/10-19.35
T_{in} (K)	385.6		838.1
P_{in} (kPa)	770.9		104.3
Pressure Loss (%)	2.0		5.3
K_j	1.312		1.069
K_f	8.737		5.265
b	0.006		0.026
Model-Calculated Parameters			
P	9,759	Q	5.3
V (m^3)	2.97	XY (m^2)	17.64
X=Y (m)	4.2	Z (m)	0.17

Table 4.7: The Assumed Values for Key Parameters, Input to the Heater's design and the Resulting Heater's Characteristics.

pressure compressor runs at a speed of 8,276 RPM and its dimensions are 0.23 m shroud radius and 0.34 m outer impeller radius. The air leaving the volute of the compressor is going to be at a temperature of 386 K and a pressure of 770 kPa.

A plate-fined counterflow air heater was chosen to heat the compressed air to a temperature of 798 K. Due to losses the pressure will drop to 740 kPa. The size of the heat exchanger was optimized to a minimum volume of 100 cubic feet. Its height came out to be 0.44 ft, while its other dimensions can be squared to about 15 ft long each. The surfaces chosen were $1/9 - 24.12$ and $1/10 - 19.35$ picked from Table 4.3 of reference [6]. The last part of the design consists of three turbines that will power the three compressors, and a power turbine that will supply the additional power to the cycle. The first of the four turbines, the high pressure one, will power the third compressor. It will run at the same RPM as the compressor, but its specific speed is 0.154 with a shroud radius of 0.26 m, and an outer impeller radius of 0.34 m. The temperature and pressure will drop to 721 K and 542 kPa respectively. The second turbine, running at the same speed as the second compressor, will be

Air Mass-Flow Rate	*69.8 (kg/sec)*				
LP Compressor					
R_{Shroud} (m)	0.42	$R_{Impeller}$	0.65	RPM	4,152
MP Compressor					
R_{Shroud} (m)	0.31	$R_{Impeller}$	0.47	RPM	5,890
HP Compressor					
R_{Shroud} (m)	0.23	$R_{Impeller}$	0.34	RPM	8,276
LP Intercooler					
T_{in} (K)	359.5	T_{out} (K)	304.5	Number of Finned Tubes	1,482
L_{Air} (m)	0.65	L_{Water} (m)	3.8	Height (m)	0.42
HP Intercooler					
T_{in} (K)	379.8	T_{out} (K)	309.0	Number of Finned Tubes	1,482
L_{Air} (m)	0.61	L_{Water} (m)	3.8	Height (m)	0.42
Heater					
Volume (m^3)	2.97	X=Y (m)	4.2	Z (m)	0.17
HP Turbine				N_S	0.154
R_{Shroud} (m)	0.26	$R_{Impeller}$	0.34	RPM	8,276
MP Turbine				N_S	0.124
R_{Shroud} (m)	0.31	$R_{Impeller}$	0.47	RPM	5,890
LP Turbine				N_S	0.103
R_{Shroud} (m)	0.37	$R_{Impeller}$	0.65	RPM	4,152
Power Turbine					
R_{Shroud} (m)	0.48	$R_{Impeller}$	0.64	RPM	4,701

Table 4.8: The Physical Characteristics of the Air-Bottoming Cycle.

of 0.47 m outer impeller radius and of a specific speed of 0.124. The shroud radius of the impeller is 0.31 m. The exit conditions are going to be a temperature of 645 K and a pressure of 386 kPa. The third turbine, powering the first compressor, has an outer impeller radius of 0.65 m and a specific speed of 0.103. The air will now enter the power turbine at a speed of 43.76 m/sec, with temperature and pressure of 573 K and 269 kPa. This turbine is of 0.48 m shroud radius and of 0.64 m outer impeller radius running at 4,701 RPM. The air expands to the pressure of 101 kPa and a temperature of 485 K, supplying an additional power of 9,698.2 HP. The total length of the Air-Bottoming Cycle is estimated to be 4.8 m, which is significantly smaller from the 8.1 m that the LM2500 gas turbine occupies. All the calculated values of the design process are summarized in Table 4.8.

In the following chapter cost considerations are addressed. The physical characteristics developed in this chapter are used for the calculation of the acquisition cost, while the performance characteristics determined in the previous chapter are use for the calculation of the operating and life-cycle costs.

Chapter 5

Cost Analysis

5.1 Introduction

The past forty years have seen major advances in naval propulsion technologies. Although advancements have occurred in all three areas of propulsion (engines, transmissions and propulsors) the development of the gas turbine was by far the most astonishing. Gas turbine technology has matured so much, that it may not be cost effective to improve a simple cycle by either using exotic materials or advanced manufacturing methods. While the gains of doing so would admittedly be small, the investment could be very significant. On the other hand, a simple cycle could enjoy major improvements in both performance and operating cost by rerouting the flow of the fluids driving the engine with a reasonable initial investment. It would be reasonable then to improve the simple gas-turbine cycle, which, apart from the defense systems, contributes the most to a Naval combatant's acquisition and operating costs. Apart from the Air-Bottoming Cycle's (ABC) exceptional performance, which is a significant improvement on the LM2500, there are cost considerations that must be examined. The performance of a new engine describes its efficiency in powering a vessel, while its acquisition and operation costs describe its effectiveness in reducing the overall life cycle cost of a vessel. Both efficiency and cost considerations must be taken into account before replacing an engine. The ABC on the other hand does not replace the LM2500. Rather, it is added to it.

Cost has always been a major driving factor in the development of a new engine. There are several components that constitute the cost function of an engine. The most significant ones cover the acquisition cost and financing, the fuel oil costs, the maintenance and repair costs, the lubricating costs along with the crew-related cost and insurance [9]. For the purpose of this study, only the first two constituents were considered. Although all the foregoing variables change from one engine to another, it was assumed that the engines compared would only differ in acquisition and fuel costs. This may be a crude assumption, especially for an engine such as the ABC. The fact that there is no combustor in the system, eliminates part of the maintenance cost which appears in all the engine components that run in very high temperatures. This also eliminates the need for very exotic materials that are usually used to withstand combustion temperatures. On the other hand, the existence of the heater adds some maintenance cost, which, nevertheless, should be lower than that of a combustor. Thus, there is going to be some cost benefit from the fact that the new part of the ABC is an unvitiated-air turbine. Finally, the life cycle cost of the Air-Bottoming Cycle will provide some important figures about the life-cycle savings that the Navy would make if the ABC is installed aboard the Navy combatants. In the estimation of the acquisition cost, the physical characteristics determined in the previous chapter were used in relating each component of the new engine to an existing one. The operating and life-cycle cost estimates were based on the performance predictions of the thermodynamic analysis of the Air-Bottoming Cycle.

5.2 Acquisition Cost

Estimating the acquisition cost of the Air-cycle was the most difficult task throughout this cost analysis. It was not the complexity of the system that the task so difficult. Rather, it was the proprietary nature of the gas-turbine technology, that kept useful data, covering the turbine components' prices, unavailable. The possibility of using a state of the art cost model, that the U.S. Air-force has developed for the estimation of aircraft engines' costs [10], was investigated for the ABC's pricing, but it failed to give any reasonable results. This is believed to be due to the much higher temperatures developed in these engines, as well as the complicated design and fabrication of the jets usually attached to a fighter's jet-engine. Thus, after a series of discussions with experts in the gas turbo-machinery industry, the acquisition cost was developed from three methodologies. The first is based on fitting the new engine's components to the production line of a radial compressor's manufacturer. The second is based on estimates extracted from discussions with industry experts on General Electric's estimates and the third is based on General Electric's estimates.

5.2.1 Fitting the Designed Components into the Product-Curves of a Radial-Compressor Manufacturer

In the previous chapter all the physical characteristics of the ideal components for the air-cycle of the Air-Bottoming Cycle were determined. In this first attempt to estimate the acquisition cost of the rotating parts of the air turbine, the physical characteristics and the operating conditions were used to fit each compressor and turbine designed, to the line of products of a major radial compressor manufacturer [11], [12].

The inputs to this methodology are the volumetric flow, the inlet temperature , the inlet pressure and the pressure ratio of each compressor or turbine. Table 5.1

summarizes all the input quantities to the methodology used for fitting the designed

Quantity	LP Compressor	MP Compressor	HP Compressor
Inlet Temperature (Deg R)	518.7	548.2	556.2
Mass Flow (lbs/sec)	153.9	153.9	153.9
Volumetric Flow (CFM)	119,777	59,888.5	29,944.25
Pressure Ratio	2:1	2:1	2:1
Air-Mole Weight ($\frac{Lb}{Lb\cdot Mole}$)	28.97	28.97	28.97
γ	1.4	1.4	1.4
Speed (RPM)	4,153	5,891	8,277

Table 5.1: The Input Quantities to the Model Used for Fitting the Designed Rotating Components to the Line of Products of a Radial Turbomachinery Manufacturer

rotating components of the ABC into the line of products of the manufacturer.

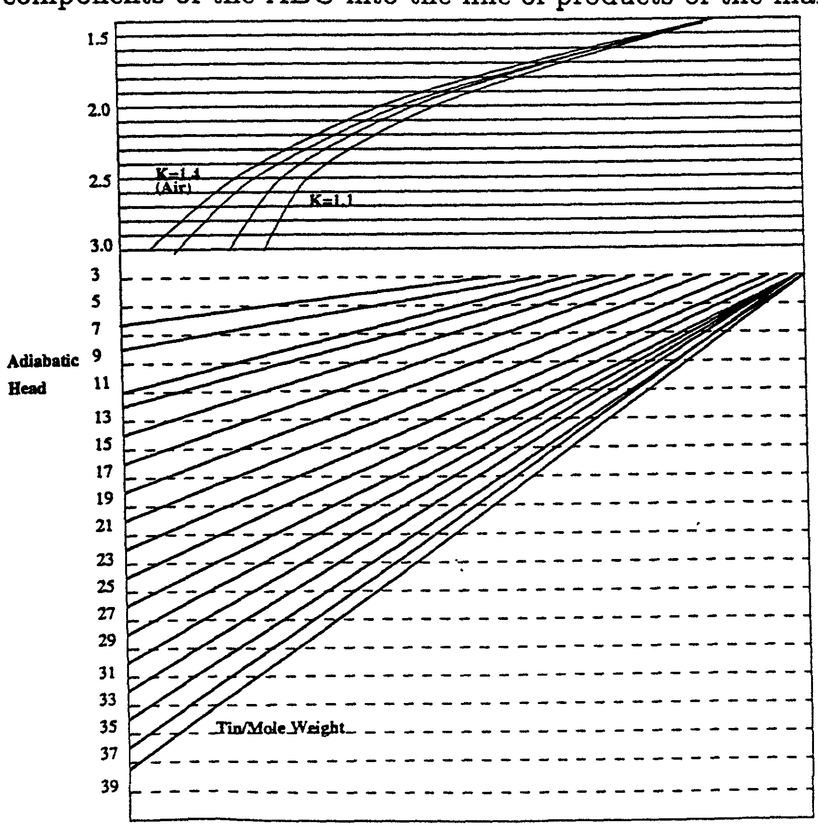


Figure 5-1: Adiabatic Head Chart. Compressibility Factor Z=1.0.

In order to find the appropriate frame size corresponding to each compressor or turbine, a series of calculations were performed, necessary to determine quantities used by the model.

Since in the case of the ABC the gas mix is air, some of the assumption made are that the molecular weight of the gas mix is 28.97 lbs/mol and the K is 1.4. The adiabatic head developed in each compressor was determined by using Figure 5-1 with the assumed or calculated values of pressure ratio, K and the ratio of the inlet temperature to the molecular weight of the gas mix. With the adiabatic head in thousands of feet, the rotating speed in thousands RPM and the volumetric flow rate in cubic feet per minute (cfm), Figure 5-2, [12] was used to select the appropriate frame for each designed compressor. Thus, for the low-pressure compressor model 90P of the P-Line series of the manufacturer is the best fit. Similarly for the medium-

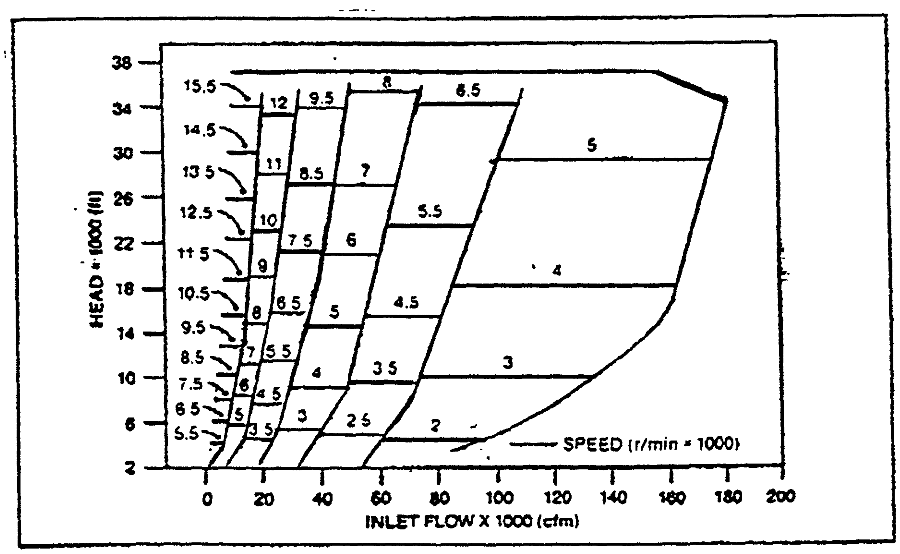


Figure 5-2: Frame Selection Chart.

pressure compressor the best fit would be the 70P, while for the high-pressure one model 60P would best fit the size and performance characteristics.

Table 5.2 summarizes the results of the methodology. Table 5.2 also includes the prices of the models which were empirically calculated by linearly relating the frame size and the model price of the compressors. The formula used is:

$$SellingPrice = \$1.15 \cdot 10^4 \cdot Frame - Size \tag{5.1}$$

So for the low-pressure compressor the selling price would be: $SellingPrice = \$1.15 \cdot 10^4 \cdot 90 = \$1,035,000$ The total acquisition cost of all the compressors of the air side of the Air-Bottoming Cycle was estimated \$2.53 million. The acquisition cost of the turbines was based on the cost for the compressors since they are similar in size and the operate at the same RPM.

Thus the total acquisition cost of all the rotating components of the air side of the Air-Bottoming Cycle was estimated around \$6 million.

5.2.2 Industry Experts' Estimates

According to representatives of Allied Signal Air-Research [13], in order to get real values for acquisition costs, one should break down the development and production costs into their constituents. This requires estimating the cost of hardware, tooling, extruding and recurring. Also, some knowledge of the tests that have to be run is required. All this result in the man-power required to work on the development of the new engine before it is available for purchase. The man power would also be expressed in terms of man-hours. Thus, the acquisition cost would be a function of the foregoing variables, after taking into account how many engines of the kind will be purchased. Most of this work is usually derived from past experience with similar engines. However, since this is the first attempt to value such an engine, past experience is of little help, and can only be used as a guide to an initial

Quantity	LP Compressor	MP Compressor	HP Compressor
Isentropic Head ($\frac{Ft}{Lb \cdot Lbm}$)	21,209	23,000	23,600
Number of Stages	1	1	1
K	1.4	1.4	1.4
Frame Size (Model)	90P	70P	60P
Maximum Diameter (m)	4.0	3.0	2.6
Length (m)	1.8	1.5	1.3
Width Diameter (m)	4.2	2.9	2.4
Price (FY95 \$*million*)	1.035	0.805	0.69
	LP Turbine	MP Turbine	HP Turbine
Price (FY95 \$*million*)	0.69	0.805	1.035
Price (FY95 \$*million*)		Power Turbine	0.92

Table 5.2: The Calculated Quantities for Frame Selection of the Compressors and the Resulting Models Selected with their Prices. The Frame Size was Used fro the Purpose of estimating the Price and not to Estimate the Size. The Size of each Compressor is Shown here only as a Reference Point. The Prices of the Turbines were Based on the Prices of the Corresponding Compressors

rough estimate. A representative of the Westinghouse Electric Corporation [14], Sunnyville, CA, who works on the development of the ICR engine, outlined the difficulties and incorporated them into a first estimate of a totally new engine. Eventhough the ICR engine is based on existing components and although at the time of the personal conversation it was tested for only 20 hours, cost estimates are expected to change before the engine is in production. It was suggested that an engine of the size of the Air-Bottoming Cycle would cost somewhere between \$25 and \$30 million dollars to develop. It was estimated that the heat exchanger will cost \$500,000 dollars while the rotating components (including the three compressors, the three turbines and the power turbine), could cost close to \$1.5 million dollars.

5.2.3 General Electric's Estimates

The Air-Bottoming Cycle was originally conceived at General Electric by William Farrell. It is no surprise then, that General Electric has looked on the possible applications for such a new engine. Special interest has been expressed for marine applications of the ABC. Figure 5-3 shows the Air-Bottoming Cycle as it appears in General Electric's marine applications brochure. General Electric [8] that has looked into alternative designs for a replacement of the LM2500, that will compete with the ICR engine, comtributed in the development of the acquisition cost of the ABC cycle for this thesis.

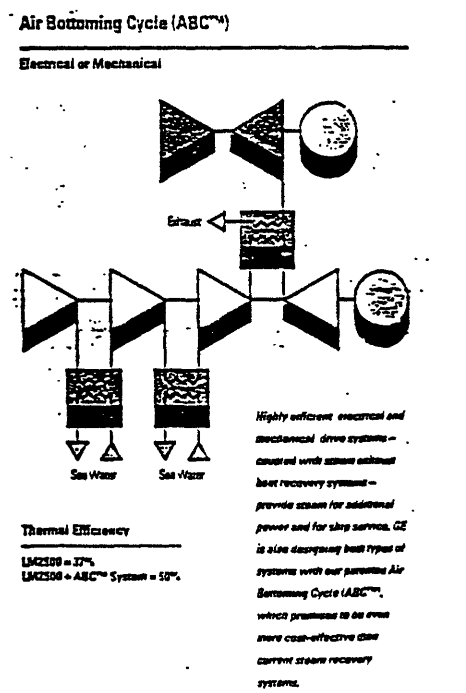


Figure 5-3: The Air-Bottoming Cycle has been Considered for Future Marine Aplictions by General Electric also.

Based on axial turbine components, it was estimated that the compressor and the turbine would cost \$3.8 million dollars, which covers the cost of the three compressors, the three turbines coupled to them, as well as the cost of the power turbine. Each intercooler operating between two adjacent compressors would cost \$195,000 dollars, adding up to a total of $390,000 dollars for the two intercoolers needed. Finally, the heat exchanger was estimated to cost $480,000 dollars. One of the main factors that led to a radial design was the lower acquisition cost that these engines have, as compared to their axial-flow equivalents. This is due to the fact that fewer stages are required, and both tooling and machining the components is inexpensive. Thus a 30% deduction in the cost of the rotating parts of an equivalent axial engine [1] would bring the total acquisition cost of the air side of the ABC engine to $3.53 million dollars. Although the final price for the ABC engine will add $3.53M to the price of the LM2500 which is $4.5M [15] and which is 23% higher than the originally predicted price for the ICR engine, it is believed that on one hand the ABC+LM2500 will have a lower acquisition cost since it would be sold as one engine with the LM2500. On the other hand, the ICR was neither valued on the same basis with the ABC, nor is it at this point estimated to cost as much as the originally estimated price. Later in this thesis, a proposal for how the ABC will pay off its extra cost will be presented. Table 5.3 summarizes the total acquisition cost of each component of the new engine and the ABC as a whole, as it was estimated from the three different methodologies.

	Acquisition Cost (FY-95 $Million		
Engine Component	Manufacturer	Industry Experts	General Electric
Compressors	2.53	0.7	1.5
Intercoolers	-	-	0.39
Turbines	3.45	0.8	2.3
Heater	-	0.5	0.48
Assumed Total Acquisition Cost from GE			4.67

Table 5.3: The Acquisition Cost of all the Components of the Air Side of the Air-Bottoming Cycle, Estimated from Three Different Methodologies.

[1] These parts would be the compressors, the turbines and the power turbine, since their price is based on the cost of axial components which tend to be more expensive than radial ones [13]

5.3 Operating and Life Cycle Costs

The operating cost of a surface combatant's propulsion plant is the second largest constituent of the ship's life cycle cost. According to Haremeyer [14], the Navy spends an average of $1.5 million a year on propulsion fuel for each mid-size surface combatant. This fuel cost adds up to $60 million per vessel during its 40-year life.

Economic Assumptions			
Fuel Cost ($/Ton)	220.0	Discount Factor	6%
Avg. Operation (hrs/Yr)	2,700	Vessel's Life (Yrs.)	40
Configuration Assumptions			
Drive	Electric	Number of ABCs Aboard	1

Table 5.4: Some Economic and Propulsion Configuration Assumptions Made for the Calculation of Operating and Life-Cycle Costs

This is 15% to 22% of the ship's acquisition cost, depending on the type and the operating profile of the vessel. Thus, savings in the area of propulsion fuel could make the difference in the development of a new engine or even a new type of hull. It has been the trend of the more recent naval development strategies not only to built more efficient and cost effective designs, but also to integrate more than one type of naval combatant into one design philosophy. This is a result of the rise in oil prices as well as the end of the cold war and the change in world politics that came along with it.

For the purpose of estimating the ABC's operating cost there were a few assumptions made that simplified the overall task. First, the only part of the operating cost considered was the propulsion fuel cost. Second, the fuel cost is based on the operating profile and the power requirements of the U.S. Navy destroyer DDG-51, Arliegh Burke. As shown in Figure 5-4, a representative operating profile of DDG-51 as the percentage of time spent at each speed [7]. Assuming a 2,700-hour annual operating time that is the average for a surface combatant, and using the required power

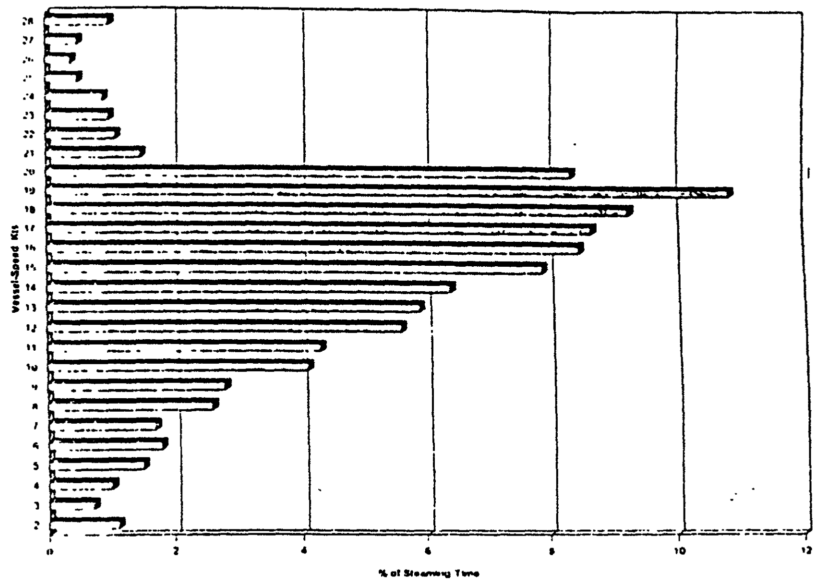


Figure 5-4: Typical U.S. Navy Destroyer Operating Profile.

rating as given by the U.S. Navy's naval-ship design program ASSET, Figure 5-6, two application scenarios were investigated for the ABC engine. Some economic assumptions were also made. Table 5.4 summarizes all the assumptions that the operating and life-cycle costs were based upon.

5.3.1 The LM2500+ABC Operating as Main Propulsor

In the first scenario, the air-cycle is coupled to the propeller shaft. The ABC is connected to the main transmission since it runs at almost the same RPM as the LM2500. In this case the ABC is responsible for the generation of a significant fraction of the total SHP required to move the vessel at the various speeds. Thus, the LM2500 will be working at lower power ratings. At these lower rating levels, where gas turbines run very inefficiently, there is significant improvement not only in the power generated, but also in the *sfc* of the turbine. For instance there is a 100% additional power provided by the air-turbine when the LM2500 generates 1,000 HP [8], while the *sfc* of the engine drops 51.1% from 1.545 to 0.789. At the 3,000 HP level of the LM2500, the ABC generates 1,554 HP or an additional 50%, while the *sfc* drops 33.1%. The improvement becomes almost constant after the 10,000 HP rating of the LM2500, with an average of 32% additional power and a 24% drop in the *sfc*. Figure 5-5 demonstrates the *sfc* of the ABC as compared to the *sfc* of the LM2500.

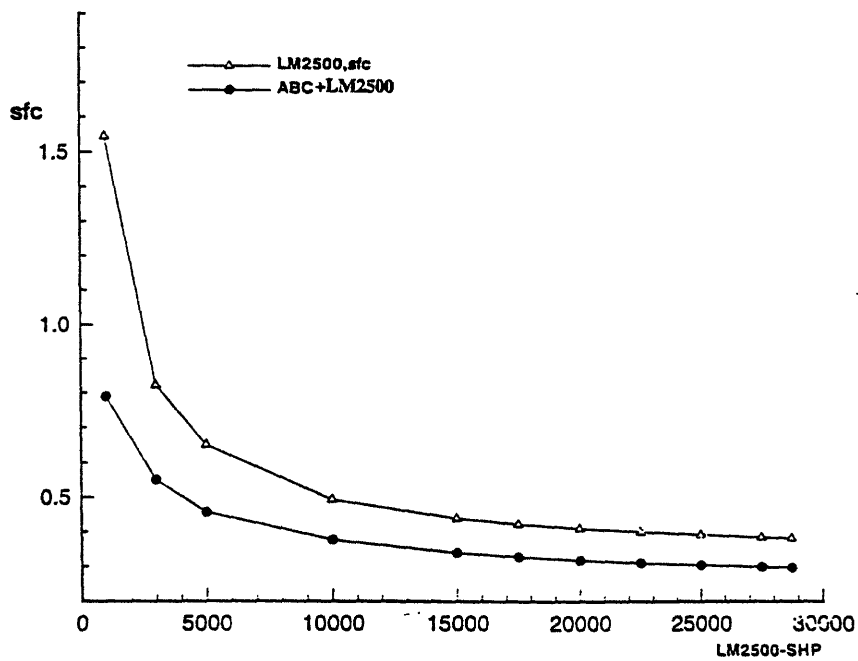


Figure 5-5: Air-Bottoming Cycle *sfc* Improvement to the LM2500.

As a basis for comparison, the LM2500's fuel consumption was estimated first. From discrete points of operation of the LM2500, where the *sfc* values were provided by the manufacturer [16], an equation was derived for the LM2500's sfc as a function of SHP:

$$sfc_{(LM2500)} = 21.219087 \cdot SHP^{-0.3981}{}_{(LM2500)} \quad (5.2)$$

By multiplying the sfc by the corresponding SHP and the percentage of time spent at the specific horsepower, the consumption can be calculated. Adding up

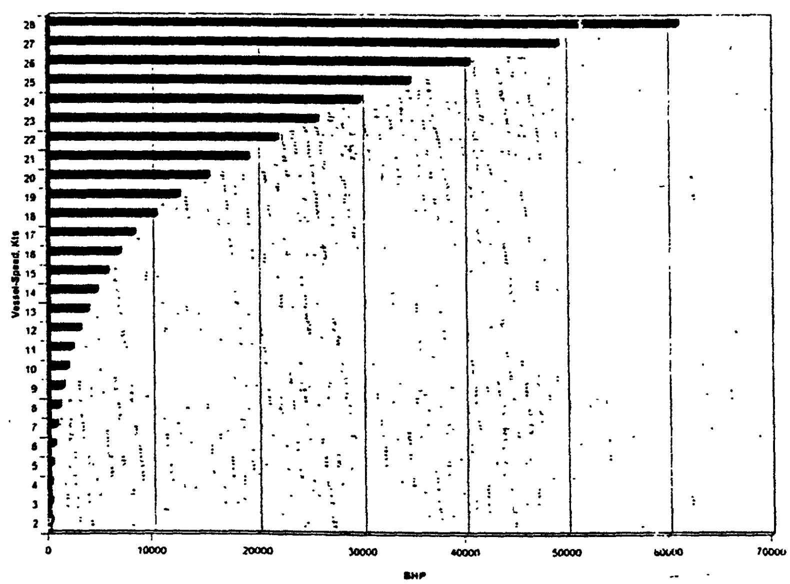


Figure 5-6: Typical U.S. Navy Destroyer Power Requirement at Different Speeds.

the various consumption levels, the average consumption of operation came to be 4,222.72 lbs/hr. Multiplying the consumption by 2,700 hours per year, the annual fuel consumption of an LM2500 was found to be 11,401,344 lbs of fuel per year, which cost about $1.2 million. From Figure 5-7 it was found that the LM2500's SHP as a function of the overall SHP, generated by the LM2500+ABC, is described by:

$$SHP_{(LM2500)} = 0.78904 \cdot SHP_{(LM2500+ABC)} - 514.134 \tag{5.3}$$

This relationship gives the LM2500's SHP contribution in the overall ABC at the various horsepower levels. One technique for estimating the fuel savings from the addition of the air-turbine is to calculate what the fuel burning-rate will be for the LM2500 if it provides only the fraction of the power required shown in Figure 5-7 and using again the LM2500's sfc. By applying the foregoing technique for the calculation of the fuel consumption and using equation 5.3 for the corresponding operating power levels of the LM2500, it was found that the average fuel consumption drops to 3,483.4 lbs/hr. This leads to an annual savings of 2.0 million lbs of propulsion fuel or $3.53 million life cycle savings from operating costs, for every LM2500+ABC combined engine aboard a DDG-51, using a 6% discount rate. Since the power, efficiency and sfc improvements are so encouraging, such a consumption improvement did not seem to represent the actual ABC performance.

In 1992, Stanko [15] provided a relationship of the ICR' sfc as a function of the percentage power generated.

$$sfc = 0.2353 \cdot \left\{\frac{BHP}{26400}\right\}^{-0.3485} + 0.1237 \cdot \left\{\frac{BHP}{26400}\right\}^{1.487} \tag{5.4}$$

Equation 5.4, when used for the calculation of the ICR's fuel consumption, provides a 2,802 lbs/hr average fuel consumption. This is in the vicinity of 5% lower than the fuel consumption of the LM2500+ABC. The difference between the two competing engines in the average sfc is 0.026 (0.503 for the ABC and 0.477 for the ICR) over the operating profile of a mid-size surface combatant is similar. Nevertheless, the extra 9,000 HP that a Navy destroyer will have available when using the ABC instead of the ICR should be considered.

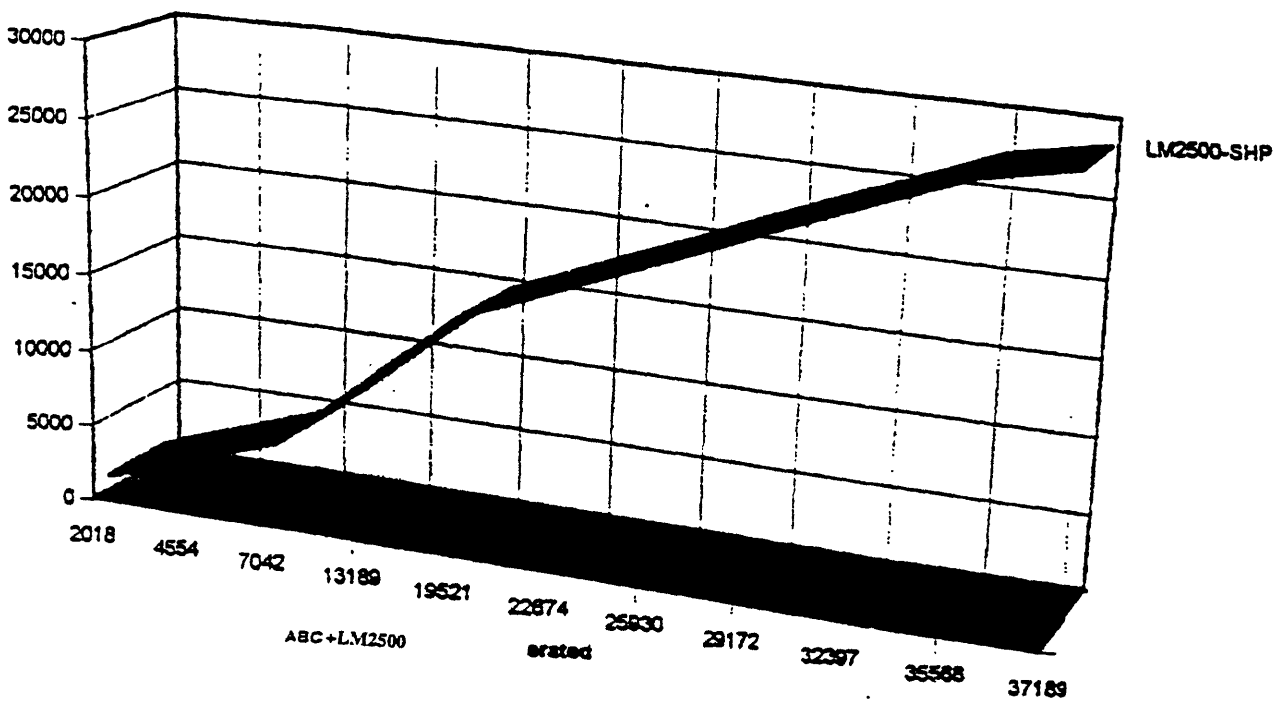


Figure 5-7: LM2500 Power Supplied as a Function of the Power Generated by the LM2500+ABC.

5.3.2 The ABC Operating as Electric Generator

In the second scenario, the applicability and the cost effectiveness of the air-turbine powering an electric generator was examined. It was assumed [15] that a DDG-51 vessel is required to carry three generators at 2,500 KW rating each. The average electric load for such a combatant is estimated to be 2,525 KW. Also, the average endurance power is 15,000 BHP and the electric load of 2,525 KW is equally split between two generators that operate continuously. For the purpose of this thesis the replacement of an Allison 501-K34 generator set was examined. In this scenario, the ABC is going to physically replace one 501-K34, while according to its power generation, and after the 2,525 KW rating not only it will shut down the second, but also its excess power is going to be forwarded to the propeller shaft. With the immediate replacement of one out of three engines, the Navy would cover $2.3 million of the ABC's acquisition cost, which is the savings from acquiring one Allison less. By using the ABC, onboard a surface combatant like the DDG-51 there would be two 501-K34s while the ABC will provide the power of the third. In estimating the savings from the fuel savings, Stanko [15] suggests that the average sfc of the 501-K34 supplying the required bleed air is 0.9 lbs-fuel per HP per hour. For the second generator running with no bleed air, the corresponding value is 0.8. Therefore, if the second generator provides 1,260 KW with no bleed air, it would burn an average of 1,524 lbs-fuel per hour. The operating profile of the generators in this scenario is summarized in Table 5.5. In the same table the estimates of operating and life-cycle cost savings are shown.

Configuration: 1-ABC+ 2-Allison 501-K34			
% Vessel's Operation	Allisons in Use #	Allisons' Rating (KW)	ABC's Rating (KW)
27.2	2	1,260	¡1,260
37.4	1	1,260	1,260
35.4	0	-	2,525
Operating and Life-Cycle Cost Savings			
% of Vessel's Operation	ABC's Function	Annual Savings ($)	Life-Cycle Savings ($ Million)
27.2	Propulsion	47,500	0.9
37.4	Generator (1,260 KW)	158,620	2.72
35.4	Generator (2,525 KW)	321,000	5.6
Total Life-Cycle Savings ($ Million)			9.22

Table 5.5: The Operating Profile and the Cost Saving of the Second Operating Scenario for the ABC Aboard a DDG-51

An electric drive configuration would allow the ABC to contribute to the improvement of the LM2500's sfc, during the lower stage of the ship's operating profile. As shown in Table 5.5 the total cost savings from this operating scenario sum up to $9.22 million F95-USD of savings throughout the vessel's 40 years of operation for every LM2500+ABC engine. This is more than \$0.5 million savings a year which means that the ABC will pay-off the remaining \$1.23 million from its acquisition when it replaced the one Allison in less than three years.

LM2500+ABC	LM2500	ICR
Fuel Consumpt.($\frac{Lb}{hr}$): 2,978.6	4,223	2,802
sfc: 0.503	0.783	0.477
	Operating Scenario	
Cost Parameter	ABC as Main Propulsor	ABC as Electric Generator
Fuel Consumpt.($\frac{Lb}{hr}$)	2,979	2,979
Annual Savings (\$ Mill.)	0.35	0.6
Life-Cycle Savings (\$ Mill.)	6.0	10.62
Acquisition Cost (\$ Mill.)	*3.53*	

Table 5.6: The Operating and Life-Cycle Cost of the Air-Bottoming Cycle when installed aboard a DDG-51. The Acquisition Cost used is that developed from General Electric's estimates.

The acquisition cost used in this study for the air-turbine of the ABC is that provided by General Electric. The reason behind this selection is both that General Electric is the primary manufacturer interested in the development of the ABC engine and that General Electric's cost estimate is somewhere in the middle of all the estimates calculated. In the same manner an even more optimistic calculation of the LM2500+ABC engine's sfc was omitted. Under that estimate, the fuel burning rate of the combined engine was lowered to 2,375 lbs/hr. Concluding, in the first scenario, the figures are comparable to these of the ICR engine, which might be the primary competition for the ABC. In the second scenario, which is possible only due to the fact that the ABC has a separate power turbine, that gives the flexibility to be attached in both the propeller shaft and/or a separate generator shaft, the

ABC outperforms the ICR's operating cost by 30%. The fuel cost-savings resulting just by operating one ABC for supplying additional power to the propeller shaft, as in the first scenario, account for 22.5% of today's overall cost of fuel accounting all engines aboard a vessel, while the corresponding figure resulting from the second scenario is as high as 38.5%. Both scenarios give more than encouraging results for the cost effectiveness of the ABC engine. Table 5.6 summarizes the operating and life-cycle costs of the combined Air-Bottoming Cycle when it is installed aboard a U.S. Navy destroyer DDG-51 as well as the savings from replacing the LM2500 or not installing the ICR engine.

Chapter 6

Conclusions

A new engine, in order to be produced usually, needs to generate 15% of performance and cost improvements when it is targeted for military applications, and 20% when it is marketed in the commercial sector. The combination of the LM2500 and the ABC improves the performance of the former by 34%. Also the fuel consumption of the LM2500 can be reduced by more than 30%. The ABC not only meets the commercial limits, but also it gives such flexibility for various applications, that makes the new engine very attractive and a possible solution for todays limited budgets for military applications.

It should be taken into consideration, that this is an attempt to model an original conceptual design of a new engine. This thesis is not intended to provide a complete, final and detailed design of the new engine, nor is it intended to suggest that this new engine is more appropriate than another for certain applications. It is focused on the analysis, modeling and investigation of the feasibility of such combined gas-turbine cycle, that would lead to a more cost effective utilization of current gas-turbine technologies which are widely used on mid-sized surface combatants.

Further investigation may bring to light additional improvements, that will make the engine even more reliable and dependable. Also, modification of the cycle may change its operation profile. Such a modification could be the addition of a combustor unit that could be either shut or running according to the power requirement of the vessel. Such a scheme has been examined by the inventor of the cycle Mr. William Farrell of General Electric. Farrell has examined the performance of the ABC with axial turbine components, a design which results to similar improvements, but which is primarily developed for industrial applications. In this thesis' application, since the engine is supposed to operate aboard a surface combatant, both size and cost considerations forced towards a radial design, which may be slightly less efficient than the axial-flow equivalent.

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