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**A PARAMETRIC STUDY OF GAS TURBINE CYCLES
FOR ELECTRICAL POWER GENERATION**

by

Ian James Nickerson

B. Eng. (Mech.), Carleton University, Canada, 1991

**A thesis submitted to the Faculty of Graduate Studies and Research in partial fulfillment of
the requirements for the degree of**

Master of Aerospace Engineering

Ottawa-Carleton Institute of Mechanical and Aeronautical Engineering

Carleton University, Canada

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Carleton University

September 5, 1994

Abstract

Sales of industrial gas turbines have risen sharply recently, and it appears they may soon become the preferred method for base load power generation. This switch from more traditional powerplants is due to such factors as the gas turbine's high efficiency, small size and low emissions.

The purpose of this thesis is to analyze the performance of current state-of-the-art gas turbine power systems, using computer simulations. The major losses in real cycles are accounted for, and comparisons between program output and existing data are performed to check the validity of assumptions made.

Gas turbines examined include the well established aero-derivative and heavy frame single-shaft units, both in simple and combined cycle applications. Also analyzed is an older design currently re-entering the market, the reheat cycle gas turbine. Based on the performance data gathered, insights are made into the designs of current models, and the changes expected in future designs.

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Nomenclature

Symbols

ε	Heat Exchanger Effectiveness
η_{rc}	Compressor Polytropic Efficiency
η_{cc}	Combustion Efficiency
η_g	Generator Efficiency
η_m	Mechanical Efficiency
η_{rt}	Turbine Polytropic Efficiency
η_{th}	Thermal Efficiency
ΔT	Total Temperature Change
$^{\circ}\text{C}$	Degrees Celsius
C_p	Specific Heat at Constant Pressure
f	Fuel/Air Ratio
K	Degrees Kelvin
m	Mass Flow
M	Molecular Mass
P	Pressure
R	Gas Constant
s	Entropy
T	Temperature
W	Specific Work

Subscripts

1,2,3, etc.	Station Numbering
Prod	Exhaust Products

Abbreviations

CCGT	Combined Cycle Gas Turbine
CHU	Centigrade Heating Unit
EGT	Exhaust Gas Temperature
Exh	Exhaust
GT	Gas Turbine
HE	Heat Exchanger
HP	High Pressure
HPC	High Pressure Compressor
HPT	High Pressure Turbine
HRSG	Heat Recovery Steam Generator
IP	Intermediate Pressure
ISO	International Standards Organization
LCV	Lower Calorific Value
LP	Low Pressure
LPC	Low Pressure Compressor
LPT	Low Pressure Turbine
MW	Megawatt
OPR	Overall Pressure Ratio
PR	Pressure Ratio
RTF	Reheat Temperature Fraction

SFC	Specific Fuel Consumption
ST	Steam Turbine
Temp	Temperature
TIT	Turbine Inlet Temperature

Companies

ABB	Asea Brown Boveri
GE	General Electric
PW	Pratt and Whitney
RR	Rolls-Royce

CHAPTER 1

Introduction

1.1 Background

The use of gas turbines for electrical power generation dates back to the early 1940's. This use, however, has generally been on a very small scale, with most generating capacity being supplied by coal or oil fired plants, hydro-electric projects or nuclear power stations. The areas where industrial gas turbines have had the most success are, for the most part, small niche markets. These include emergency power supply, where a gas turbine's ability to achieve full power within a few minutes of startup is key. Peak load power supply is also a common role for gas turbines, augmenting electrical utilities' base load plants at times of high demand. Finally, the off-shore market has been almost totally captured by gas turbines due to their small size and large power density. In terms of installed power, however, that provided by gas turbines is only a very small percentage of the world total. Sales have been steady but not in large numbers.

Since the late 1980's, there has been a fairly dramatic change in the sales pattern of industrial gas turbines. This fact is clearly demonstrated by Figure 1A¹, which shows worldwide gas turbine sales, by capacity, for the period 1980-1998. The 1980's saw an average annual capacity addition of 6500 MW, but this figure is predicted to jump to 26,000 MW annually for the 1990's, a four-fold increase.

Figure 1A: Worldwide Gas Turbine Orders
Source: Reference 19

Table 1.1 shows the regional breakdown of worldwide turbine/generator sales forecasts made by General Electric Co., for the period 1994-2002. These values are for all types of powerplants, not just gas turbines.

¹ The convention used for figures in this thesis is as follows. Alphabetically designated figures, e.g. Figure 1A, appear within the text of the chapter, and are generally diagrams or reproductions. Numerically designated figures, e.g. Figure 1.1, appear at the end of their respective chapters, and show performance data generated for this thesis by computer programs.

Table 1.1: Forecast Worldwide Power Orders for 1994-2002
Source: Reference 20

Region	Orders (MW)	Installed (MW)	Annual Growth (%)
North America	180,000	120,000	2.0
Latin America	84,000	49,000	3.9
Japan	65,000	46,000	3.4
Pacific Rim	330,000	180,000	6.2
India	53,000	31,000	5.4
Europe and Africa	321,000	217,000	2.2
Worldwide Totals	1,033,000	643,000	3.0

The most surprising prediction is the way in which this new power will be provided. Of the 1.033 million megawatts of total power orders, GE expects over 45% to be supplied by gas turbine based systems, a huge increase over past figures. It is interesting to note that the highest annual growth rates are seen in the more underdeveloped areas of the world, where rapid industrialization is resulting in large new power requirements.

For North America, some figures are even more impressive. Utility Data Institute of Washington, DC, estimates that 58% of the new capacity additions American electric utilities expect to install in the 1990's will be gas turbine based (Reference 14). Many observers feel that this is conservative, with 65% a more realistic figure.

What has brought about this dramatic increase in demand for gas turbine based generating systems? There are several factors which play a role.

1. The efficiency and reliability of modern gas turbines have increased to the point where they are now competitive with large steam turbine powerplants for base load power supply.

2. World demand for power continues to soar, especially in developing regions. The ability of a gas turbine powerplant to be on-line within a year of contract signing is an important advantage over other types of plants.
3. There has been strong public opposition to some alternate forms of energy, especially nuclear energy. New hydro-electric projects are also coming under fire due to environmental concerns about land flooding.
4. In some countries, such as the UK, privatization of the national power supply has led to a switch from coal or nuclear plants to gas turbine plants. In the past, national energy policy had forced companies to use resources, such as coal, which were not necessarily the most efficient choice.
5. The natural gas fuel normally burned by gas turbines produces far lower harmful emissions than most alternate methods such as coal fired plants. In many cases the cost of meeting emission control standards for these older plants is prohibitively expensive.
6. Although primarily run on natural gas, the gas turbine has shown an ability to burn a variety of fuels, thus reducing its reliance on a single type of fuel. This has eased concerns over the diminishment of the world's natural gas stocks.

These factors have greatly increased the importance of gas turbine based powerplants, both now and in the years to come.

1.2 Thesis Objectives

In the past, the technology present in gas turbines used for industrial purposes has always lagged that of gas turbines for aero use. This was particularly evident in the large heavy frame machines which represent the bulk of today's installed power. The primary reason for this was that the much smaller production run for an industrial engine did not justify the huge amounts of development funding allocated the aero-engine industry. As a

result, makers of industrial engines tended to wait for new technologies to become established in aero-engines first before applying them to their products.

As of late, however, this technology gap has been greatly reduced. The increasingly competitive industrial market means that manufacturers must use the latest technologies or risk being outsold by competitors. This, and the increased sales now being realized, means that a greater portion of revenues is now funneled into research and development. This R&D funding is being used to advance current cycles, develop new ones and re-examine older ones. The objectives of this thesis are as follows:

1. To examine the performance levels attainable with current industrial gas turbine cycles.
2. To rationalize the design compromises made by current models.
3. To predict how expected advances in technology will influence future versions of existing cycles.
4. To examine the performance levels which can be expected for a new cycle expected to go into production in the near future.

The quantitative analysis of cycle performance involved a series of parametric studies, performed by computer programs written for this thesis. These programs are a tool used for performance evaluation, and are not themselves the goal of the thesis. Despite this, it was felt necessary to perform a brief validation of the program code, to verify that the cycle assumptions chosen provide reasonable results. This comparison of program output with actual performance data for some of today's latest models is done in the appropriate chapters.

1.3 Scope of Thesis Work

The investigative portion of the thesis will cover three broad categories of gas turbines. These three categories are:

1. HEAVY FRAME, SINGLE-SHAFT GAS TURBINES
2. AERO-DERIVATIVE, MULTI-SPOOL GAS TURBINES
3. HEAVY FRAME, REHEAT CYCLE GAS TURBINES

Table 1.2 shows the performance levels attained by latest models in each of the three categories listed above. Performance is given both for the gas turbine alone and for a combined cycle installation, in which the exhaust gases from the gas turbine are used to raise steam which drives a steam turbine.

Table 1.2: Current Gas Turbine Technology Levels
Source: References 15, 19

Model	Simple Cycle		Combined Cycle	
	Power (MW)	Efficiency (%)	Power (MW)	Efficiency (%)
1. GE Frame 7F	159	35.9	241	54.5
2. GE LM6000	41	39.7	54	52.0
3. ABB GT24	165	37.5	251	58.0

The heavy frame single-shaft design represents by far the greatest portion of currently installed power, and is typified by the General Electric Frame 7F, which first entered production in 1991. The latest in aero-derivative technology is shown by the GE LM6000, based on the GE CF6-80C2 aircraft turbofan. The third category, the heavy frame reheat cycle, is used by the ABB GT24, due to enter service in 1995.

Within these broad categories performance will be evaluated for both the traditional simple cycle as well as for more advanced cycles such as the combined cycle and regenerative cycle.

1.4 Analysis Technique

The method of analysis involves doing performance calculations over a wide range of the two major independent variables available to the designer. For most cycles these are the overall compressor pressure ratio (OPR) and the turbine inlet temperature (TIT). In the case of some of the more advanced cycles there are other independent variables which can be altered. Where this is done it will be clearly stated in the text.

Of course, there are many other variables present in a real cycle, chiefly relating to component efficiencies and pressure losses. Unless otherwise stated, these values will be fixed at a value appropriate to the current state-of-the-art when performing calculations. The choice of these values will be fully explained. For all cycles tested, the two main values used to examine performance are thermal efficiency and specific work output.

1.5 Literature Review

The subject of gas turbine power generation is a popular one, which is soon likely to receive even more attention, given its bright future. Much has been written, covering a broad range of topics. A number of authors have concerned themselves with a particular gas turbine model. The development of the GE Frame 7F, a 160 MW single-shaft unit, is examined by Brandt (1988). This paper is primarily concerned with the technology incorporated in the engine, with only a brief mention of the design study. A paper along similar lines by Scalzo et al. (1989) is centred on the Westinghouse 501F, another large single-shaft gas turbine. Neuhoff and Thorén (1994) look at the new GT24/26 reheat gas

turbines from ABB. The advantages and operation of the main innovation of these new models, sequential combustion (reheat), are examined in detail.

The current popularity enjoyed by combined cycle gas turbines (CCGT) has prompted a number of papers, covering many design aspects. Lugand and Parietti (1991) look at how a modern single-shaft unit, the GE Frame 9F, has been incorporated into a combined cycle layout. Most of the coverage concerns heat recovery steam generator and steam turbine technology. On a slightly different vein, Bolland (1991), basing his study on the Siemens V94.3, looks at the combined cycle performance which can be achieved using a variety of steam cycle options. These include multi-pressure levels, reheat steam turbines and supplementary firing. Rice (1987), has developed a graphical heat-balance method to predict the performance of a gas turbine in a CCGT environment, given certain parameters.

A parametric analysis along the lines of this thesis is done by Cerri (1987). However, the range of cycles is limited to the single-shaft CCGT, and the assumptions used are slightly dated. Gas turbine textbooks, such as that by Cohen et al. (1987), often use parametric studies in their explanations of the various cycles, but these studies are necessarily brief, given the wide range of subject matter covered.

The coverage given aero-derivative gas turbines has not been nearly as extensive as that of the large single-shaft units. There are some quite detailed brochures and articles, such as those by Oganowski (1990, GE LM6000) and Farmer (1994, RR Trent). These both give an in-depth look at the changes required to convert an aero-engine to shaft power. Most technical papers, however, are concerned with component performance, an example being Monhardt et al. (1990), detailing the design and testing of the PW FT8 low pressure compressor.

Looking ahead, there is much interest in the form future advanced gas turbine cycles will take. Little et al. (1992), of Westinghouse Electric, examine the feasibility of achieving a thermal efficiency of 60% from gas turbine based power systems in the next 10

years. They look at a number of techniques, utilizing intercooling, regenerators and reheat, and conclude that technological advances will indeed make the 60% goal attainable

Although the work published to date has portions which are of interest, very little deals directly with the goal of this thesis. This goal is a parametric study of the real performance of the different gas turbine types and gas turbine cycles used for power generation, both today and in the future. This will include a detailed analysis of the design compromises which must be made when choosing to optimize for either a stand-alone gas turbine or a combined cycle gas turbine.

CHAPTER 2

Historical Perspective

2.1 The Early Years

Although gas turbines have only come to prominence in the last fifty years, initial work dates back to the early part of the century. The first working gas turbine was built in 1905 by two French engineers, Armengaud and Lemale. The overall thermal efficiency of this unit was 3%, and very little useful work was produced, but it was a start.

The first sizable gas turbine expressly for power generation was built by Brown Boveri in 1939. This simple cycle unit produced 4 MW, and was designed to supply standby power. After being demonstrated at the Swiss National Exhibition at Zurich in 1939, it was formally commissioned at the Neuchatel underground power station, Switzerland, in 1940. Some pertinent data for this station is given in Table 2.1. The plant's very modest thermal efficiency of 18% (compared to a steam plant's 25%) was not of great concern, as annual running hours were very low, typically no more than few hundred hours.

Table 2.1: Test Results for Neuchatel Power-Plant, 1940
Source: Reference 2

Item	Value
Overall Pressure Ratio	4.38
Entry Mass Flow (kg/s)	62.8
Turbine Inlet Temperature (K)	848
Power (MW)	4.04
Thermal Efficiency (%)	18.0

In an effort to combat the low efficiency of the simple cycle, some manufacturers went to more advanced cycles on their gas turbines. In 1946, Escher Wyss of Zurich developed what may well be the most complex gas turbine power-plant of all time. This was the St. Denis 1 station in Paris, producing 12 MW in a closed cycle configuration. The station included intercoolers, air heaters, a precooler, a recuperator, and low , intermediate and high pressure compressors. The cycle diagram for this station is shown in Figure 2A. Due to its incredible complexity the plant was not fully complete and delivering its rated power until 1960, although it did achieve a thermal efficiency of 34%.

Brown Boveri installed the Beznau 1 and Beznau 2 plants in Switzerland in 1948. Beznau 1 gives 13 MW at 30.5% efficiency, with Beznau 2 improving to 27 MW at 34% efficiency. Aside from intercoolers and a recuperator, these units were the first to incorporate sequential combustion. A diagram of the Beznau 2 configuration is shown in Figure 2B. In a testimonial to the longevity of gas turbine power-plants, these two units are still in active service today, over 45 years after construction.

The first gas turbine for use in a central power station in the United States was installed in 1949, at the Huey Station in Oklahoma City. This was an early combined cycle plant, with the exhaust gases from the 3.5 MW gas turbine being used to preheat feed water for existing steam turbines. The thermal efficiency was 16% when the gas turbine operated alone, and 21% when the additional steam turbine output was accounted for.

Cycle diagram of the St. Denis 1 gas turbine power plant in Paris

- LC Low-pressure compressor
- MC Intermediate-pressure compressor
- HC High-pressure compressor
- HT High-pressure turbine
- LT Low-pressure turbine
- PC Precooler
- IC Intercooler
- H Recuperator
- AH Air heater
- G Generator
- M Starting motor
- CC Charging compressor
- ET Exhaust-gas turbine
- AP Combustion air preheater
- RB Recirculation blower
- MG Motor/generator

Figure 2A: St. Denis 1 Gas Turbine Power Plant
Reproduced from Reference 13

Layout of the 27-MW gas turboset Beznau no. 2 with intercoolers, recuperator and sequential combustion system

- 1 Free-wheeling LP shaft with LP turbine, LP and IP compressors
- 2 HP shaft with HP turbine, HP compressor and alternator
- 3 Intercoolers
- 4 Combustors
- 5 Recuperator

Figure 2B: Beznau 2 Gas Turbine Power Plant
Reproduced from Reference 13

2.2 Arrival of the Aero-Derivative

In the early 1960's it was recognized that the disconnection of a small portion of a large power grid could have very serious implications for the rest of the grid, due to a slight drop in grid frequency. The auxiliaries of large power-plants are very frequency conscious, and a decrease in their desired frequency could result in the disconnection of the plant from the grid. A snowball effect could rapidly ensue, possibly crippling the entire grid. This type of cascade failure did in fact occur in the 1960's, causing large-scale blackouts in both North America and Europe. The UK government's solution to this problem was to install an independent emergency power supply at all major power-plants, capable of running the plant auxiliaries (typically 15-30 MW). This emergency power was provided by a new type of gas turbine, the aero-derivative. In effect, an aero-derivative uses an aircraft jet engine as a gas generator, feeding a power turbine which in turn drives an electrical generator. The first test unit was installed in 1962 at the Hams Hall A station, using a Rolls-Royce Olympus aero-derivative producing 15 MW. Since this time, over 1000 MW of emergency power have been installed in the UK alone, based primarily on the Rolls-Royce Avon (14.5 MW) and Olympus (17.5 MW) gas turbines. The Pratt and Whitney FT-4 provides similar service at many North American sites. The key advantage of using an aero-derivative for emergency power is its quick start time. Because the starter only has to turn the relatively light gas generator, and not the much larger power turbine, it is possible for an aero-derivative to provide its full rated power in less than two minutes from startup.

The late 1960's also saw aero-derivatives being used as peak load providers, supplementing the main power grid in times of high electrical demand. A typical Rolls-Royce peak load station is shown in Figure 2C, its 4 Olympus gas turbines capable of producing 70 MW. The compact size and modular construction of the aero-derivative facilitate quick plant construction and easy maintenance.

Figure 2C: Rolls-Royce Olympus 70 MW Peak Load Station
Reproduced from Reference 6

2.3 Base Load Service

By the 1970's, the gas turbine was well established in the power generation business, but its fraction of the world's total installed power was tiny. The big players in the power business were plants which used giant steam turbines, or large hydro-electric projects. In addition, when gas turbines were used, it was generally in applications with very low annual running hours. For these roles the relatively low efficiency of the simple cycle gas turbine was not of prime concern. If higher efficiency from a gas turbine was needed, a more complex cycle had to be employed, increasing plant and maintenance costs.

For gas turbines to effectively compete with the large steam plants for base load supply, significant advances had to be made in both output and efficiency. These advances were to come in part from the aero-engine industry, where huge research and development budgets were making large gains in many areas. Aerodynamic improvements meant that compressor and turbine efficiencies were steadily improved, raising overall thermal efficiency. Improvements in high temperature materials and blade cooling techniques continually raised the maximum allowable turbine inlet temperature, which increased maximum output for the gas turbine. This rapid growth in turbine inlet temperature through the years is charted in Figure 2D.

Figure 2D: Turbine Inlet Temperature Growth Trend
Source: Reference 16

The other major boost for the gas turbine was the increasing use of the combined cycle, where the hot exhaust gas from a gas turbine is used to raise steam in a heat recovery steam generator, which is then passed through a steam turbine to increase plant output and efficiency. The exhaust gas temperature of large industrial gas turbines had steadily increased over the years, as had HRSG technology. These factors led to better steam cycle conditions, and hence greater output and efficiency.

An example of one of first modern combined cycle plants is the Korneuburg 2 station in Austria, built in the late 1970's by Brown Boveri. This plant provides 128 MW at a thermal efficiency of over 46%. For the first time, gas turbine based plants were able to deliver large amounts of power at thermal efficiencies higher than that of traditional steam turbine plants. This made the CCGT plant a realistic alternative for base load power supply. During the 1980's, few operators chose to go this route. For the 1990's and

beyond, however, the base load market for large gas turbine based systems looks set to boom, for the reasons put forth in Chapter 1.

CHAPTER 3

Program Assumptions and Theory

3.1 Cycle Constants

The performance of a real gas turbine cycle is influenced by a large number of variables, including component efficiencies and cycle losses. In order to have a similar reference to compare different cycles, some of these variables will be kept constant throughout the analysis. These variables, and their selected values, are described in detail below.

3.1.1 Ambient Conditions

All cycles are compared at standard ISO conditions of 15°C and 101.3 kPa. This thesis is designed to examine the design point performance of gas turbine cycles, so performance at other than ISO conditions will not be looked at.

3.1.2 Component Polytropic Efficiencies

In the past, aero-derivative gas turbines have always utilized more advanced technology than heavy frame units, primarily due to the vast amount of development work which has been poured into the aero-derivatives' aircraft cousins. To achieve similar component efficiencies for a given pressure ratio, heavy frame designs must use a larger number of less heavily loaded stages. Recently, however, much of the latest aero technology has been directly applied to new heavy frame designs. For companies such as General Electric, which manufacture both aero and industrial gas turbines, this is an in-house technology transfer. Other companies have obtained this technology in different ways. An example is the license agreement between Westinghouse and Rolls-Royce, which gives Westinghouse access to much of Rolls-Royce's latest aero-engine technology.

Because of the shrinking technology gap between aero-derivatives and heavy frame units, the polytropic efficiencies used for cycle work were chosen to be fixed at a certain value, irrespective of whether the cycle in question be more suitable for an aero-derivative or a heavy frame application.

3.1.2.1 Compressor Polytropic Efficiency, η_{ac}

The value of compressor polytropic efficiency used is 0.90. This is consistent with the current state-of-the-art for compressors.

3.1.2.2 Turbine Polytropic Efficiency, η_{at}

For turbines, it is necessary to assign two values of polytropic efficiency, depending on whether the turbine utilizes blade cooling or is uncooled. The process of mixing cooling air within a turbine introduces some losses, lowering its efficiency. For an uncooled turbine the efficiency is taken as 0.89, while for a cooled turbine 0.88 is used.

3.1.3 Other Efficiencies

Bearing friction causes some minor losses in all cycles. It slightly increases the work the turbine must do to drive the compressor (lowering available output) and also reduces the output available from the drive turbine. These are very minor effects, so a mechanical efficiency, η_m , of 0.995 is used.

Modern combustion techniques have resulted in near perfect combustion efficiency. Thus a combustion efficiency, η_{cc} , of 0.999 is used.

The output of an industrial gas turbine used for power generation is often quoted as a value 'at generator terminals'. This value takes the efficiency of the electrical generator into effect. A value for η_g of 0.98 is typical for a modern generator.

3.1.4 Pressure Losses

3.1.4.1 Inlet/Exhaust Losses

The intake and exhaust ducting present on an industrial gas turbine are both sources of pressure losses. Figures 3A and 3B show the effect on performance for the Rolls-Royce Olympus gas turbine. These types of pressure losses are usually quoted in inches of water. The loss values used here are common for an industrial gas turbine: 4" H₂O for the intake and 6" H₂O for the exhaust. In the case of a combined cycle, the exhaust loss is increased because of the addition of a heat recovery steam generator (HRSG). An exhaust loss of 10" H₂O is used for combined cycles.

The inlet loss is subtracted from ambient pressure to obtain the pressure at entry to the compressor, while the exhaust loss is added to ambient pressure to obtain the pressure at exit from the turbine.

Figure 3A: Intake Loss Effect on Performance (From Reference 9)

Figure 3B: Exhaust Loss Effect on Performance (From Reference 9)

3.1.4.2 Combustion Losses

The pressure drop associated with the combustion process is a very significant one. It is primarily caused by the resistance of flame-stabilizing and mixing devices necessary to maintain combustion. The mixing caused by the reaction itself also plays a role. A value of 3% of compressor delivery pressure is assumed for the combustion pressure loss in large heavy frame machines. The very compact combustors used in aero-derivatives, however, see a greater pressure loss, so a loss value of 6% of compressor delivery pressure is used.

3.1.5 Cooling Bleed

Due to the high turbine inlet temperature used on today's gas turbines, large amounts of air are required to cool portions of the turbine. There are many techniques available for this process, and they differ depending on engine configuration and manufacturer. For comparison's sake, a general method of cooling was devised. First, all cooling air is assumed to be supplied from the exit of the HP compressor. This is a typical method, although air is sometimes bled from an intermediate compressor stage. Secondly, all cooling air is assumed delivered at the first stage of the turbine, with 2/3 going to the stator and 1/3 to the rotor. For aero-derivatives, which often only have a single-stage high pressure turbine, this is a good approximation. For heavy frame units the turbine often has several stages with varying degrees of cooling, so the assumption may introduce some error. The final decision to be made is the amount of cooling flow. This value is normally expressed as a fraction of the compressor delivery flow. To examine the effect of varying this fraction, an analysis was performed on a typical single-spool heavy frame machine (PR=15, TIT=1500K). The results are shown in Figure 3.1. Specific work output is much more severely affected by increasing the cooling bleed than is thermal efficiency. Specific work is decreased by a combination of lower flow through the first turbine stage and the reduced temperature at entry to the first stage rotor, because of the addition of the cooling bleed in the stator. Thermal efficiency is less affected because the loss in work output is

partially offset by the reduced fuel flow required by the lower mass flow through the combustion chamber. The amount of cooling air is assumed to be 10% of the compressor delivery mass flow. This is believed to be a typical value for a cycle operating at a turbine inlet temperature of $\sim 1500\text{K}$, although a manufacturer's cooling technology is normally a closely guarded secret.

3.2 Subroutine FILEREAD

As the list of variable inputs required by each program is extensive, this input data is supplied as a text format input file, as opposed to being keyboard read. The user is prompted for the name of the input file, then the subroutine FILEREAD reads in the values. A sample input file for a single-shaft machine follows, with descriptions of the input variables provided to the right.

Design Run	Title of trial
2	Efficiency type variable, 1=Isentropic, 2=Polytropic
0.9	Compressor efficiency
0.88	Turbine efficiency
1	Ambient temperature type, 1=Kelvin, 2=Celsius, 3=Fahrenheit
288.15	Ambient temperature
14.6960	Ambient pressure (psia)
1330.	Entry mass flow (lbm/s)
11476.0	Fuel LCV (CHU/lbm)
0.7516	Carbon weight percentage in fuel
0.2483	Hydrogen weight percentage in fuel
0.1	Bleed fraction
0.01	Inlet pressure loss fraction
0.0	Diffuser pressure loss fraction
0.03	Combustion pressure loss fraction
0.015	Exhaust pressure loss fraction

The values of overall pressure ratio and turbine inlet temperature are provided by the user during program execution, so that a range of values can be tested

3.3 Calculation of the Properties of Air and Exhaust Products

3.3.1 Gas Constant for Air, R_{air}

The gas constant for air is given by:

$$R_{air} = R_0 / M_{air} = 287.1 \text{ J/kg K} \quad (3.1)$$

where R_0 is the molar universal gas constant and M_{air} is the molecular mass of air. The molecular composition of air is given in Appendix A.

3.3.2 Gas Constant for Exhaust Products, R_{prod}

The value of the gas constant R for the exhaust products is a function of the products' molecular masses and the fuel/air ratio, f . With the molecular mass of the products of combustion given by M_{prod} , and the carbon and hydrogen weight percentages of the fuel given by $\bar{C}$ and $\bar{H}$ respectively, it has been shown by Fielding and Topps (1959):

$$M' = \frac{M_{air}}{M_{prod}} = 1 + \frac{f}{1+f} \cdot (\bar{C} \cdot K_C + \bar{H} \cdot K_H) \quad (3.2)$$

where:

$$K_C = 3.6645 \cdot \frac{M_{air}}{M_{CO_2}} - 2.6645 \cdot \frac{M_{air}}{M_{O_2}} - 1 = -1.0 \quad (3.3)$$

$$K_H = 8.9365 \cdot \frac{M_{air}}{M_{H_2O}} - 7.9365 \cdot \frac{M_{air}}{M_{O_2}} - 1 = 6.1848 \quad (3.4)$$

Then for the exhaust products

$$R_{prod} = R_{air} \cdot M' \quad (3.5)$$

3.3.3 Specific Heat of Air at Constant Pressure, $C_{p_{air}}$

The specific heat of air, C_p , is relatively constant over a wide range of pressures. However, it has been shown to be a strong function of temperature. For accurate cycle calculations, it is necessary to employ a method of accounting for this variation. In this thesis, the fifth order polynomials in temperature developed by Chappell and Cockshutt (1974) are used. The specific heat of air, in CHU/lbm K, is given by:

$$C_{p_{air}} = C_0 + C_1 \cdot T^1 + C_2 \cdot T^2 + C_3 \cdot T^3 + C_4 \cdot T^4 + C_5 \cdot T^5 \quad (3.6)$$

Calculations for the specific heat of air are carried out in the CPA subroutine. The polynomial coefficients of Equation 3.6 are given in Appendix A. Figure 3.2 gives a graphical representation of Equation 3.6, showing the significant rise in C_p as temperature is increased.

3.3.4 Specific Heat of Exhaust Products at Constant Pressure, $C_{p_{prod}}$

It was initially planned to use the correction factors of Chappell and Cockshutt (1974) to determine the value of specific heat, C_p , for the products of combustion. However, these correction factors were based on the use of a 'standard' fuel which consists of 86% carbon and 14% hydrogen by weight. This is appropriate for gas turbines using kerosene or similar aviation fuels. However, natural gas, the most common fuel for industrial gas turbines, differs significantly from these fuels. As a result, an alternate method was required.

Fielding and Topps (1959) use a method in which the combustion products are broken down into their main constituents, air, carbon and hydrogen. The value of C_p is then found by adding a term including correction 'Theta' functions to the specific heat of air at the desired temperature. This term also includes the fuel/air ratio f and the carbon and hydrogen weight percentages in the fuel ($\bar{C}$ and $\bar{H}$ respectively). Thus:

$$\dot{C}p_{prod} = \dot{C}p_{air} + \frac{f}{1+f} \cdot (\bar{C} \cdot \theta_{Cp}^C + \bar{H} \cdot \theta_{Cp}^H) \quad (3.7)$$

The values of the Theta functions have been tabulated as functions of temperature. Apparao (1968) has fitted fourth order polynomials to this data as follows:

$$\theta_{Cp}^C = CPC_0 + CPC_1 \cdot T^1 + CPC_2 \cdot T^2 + CPC_3 \cdot T^3 + CPC_4 \cdot T^4 \quad (3.8)$$

$$\theta_{Cp}^H = CPH_0 + CPH_1 \cdot T^1 + CPH_2 \cdot T^2 + CPH_3 \cdot T^3 + CPH_4 \cdot T^4 \quad (3.9)$$

These Theta functions are evaluated in the program functions THCPC and THCPH. The values of the polynomial coefficients used are given Appendix A. The fuel used for the calculations in this thesis is natural gas, consisting of 94% methane and 6% ethane. The weight percentages of carbon and hydrogen for natural gas, methane and ethane are given in Table 3.1.

Table 3.1: Chemical Composition of Natural Gas

	Methane	Ethane	Natural Gas
Carbon Weight %	0.7486	0.7985	0.7516
Hydrogen Weight %	0.2513	0.2015	0.2483

Figure 3.2 shows a graphical representation of Equation 3.7. A typical fuel/air ratio of 0.02 is used.

3.4 Subroutine MASSFLOW

The mass flows at each of the chosen cycles stations are calculated in the MASSFLOW subroutine. The mass flows through the engine change due to the removal

of cooling air after the compressor, the addition of fuel at the combustion chamber, and the introduction of cooling bleed in the turbine.

3.5 Subroutine COMPRESS

When compressor calculations need to be performed, the COMPRESS subroutine is called. The value of compressor efficiency used is the polytropic efficiency. For compression from state 1 to state 2:

$$T_2 = T_1 \cdot \left(\frac{P_2}{P_1} \right)^{\frac{R}{C_p \eta_{pc}}} \quad (3.10)$$

As the working fluid is air, the value of the gas constant R is simply R_{air} . The calculation of C_p is somewhat more involved, as it is a function of temperature. The value of C_p used represents the average value during compression. It is necessary to use an iterative method to solve Equation 3.1, as follows:

1. Calculate C_p based on T_1
2. Using Eqn. 3.10 calculate T_2
3. Update C_p based on the average of T_1 and T_2
4. Goto 2

This process is iterated until the change in successive C_p values is within desired tolerances. For the cycle calculations in this thesis a tolerance of 0.05% is used.

3.6 Subroutine FUELFLOW

In order to calculate the fuel flow required for the engine the fuel/air ratio must be found. The values at entry (T_2) and exit (T_3) from the combustion chamber are known, as is the lower calorific value (LCV) of the fuel being used, quoted at a reference temperature of 298K. It is desired to find the fuel/air ratio f which will transform 1 kg of air at T_2 and f kg of fuel at fuel temperature T_f to $(1+f)$ kg of products at T_3 . It will be assumed that the fuel is at the reference temperature of 298K. As no work is done on the system the net change in enthalpy is 0:

$$(1+f) \cdot C_{p_{prod}} \cdot (T_3 - 298) + f \cdot n_{cc} \cdot LCV + C_{p_{air}} \cdot (298 - T_2) = 0 \quad (3.11)$$

Solving this equation for fuel/air ratio f yields:

$$f = \frac{C_{p_{air}} \cdot (T_2 - 298) - C_{p_{prod}} \cdot (T_3 - 298)}{C_{p_{prod}} \cdot (T_3 - 298) + n_{cc} \cdot LCV} \quad (3.12)$$

The value of $C_{p_{air}}$ is evaluated at the average of T_2 and 298K, while $C_{p_{prod}}$ is evaluated at the average of T_3 and 298K. Because $C_{p_{prod}}$ is itself a function of fuel/air ratio f , an iterative process must be used:

1. Assume an initial value of f
2. Calculate $C_{p_{air}}$ at $(T_2+298)/2$
3. Calculate $C_{p_{prod}}$ at $(T_3+298)/2$
4. Using Eqn. 3.12 calculate f
5. Update $C_{p_{prod}}$ based on new f
6. Goto 4

When the difference between successive values of f is less than 0.05% the iteration process is ended. For this thesis, the fuel used is natural gas with the composition given in Table 3.1. This fuel has an LCV value of 11,476 CHU/lbm.

3.7 Subroutine BALANCE

As mentioned previously, the cooling air available to a stage is assumed divided between the stator and rotor in the ratio 2:1. The cooling air introduced in the stator causes a decrease in the temperature of the main flow seen at rotor entry. In order to find this new temperature it is necessary to perform an enthalpy balance. In the following equation station 1 refers to the bleed flow, station 2 to the main flow and station 3 to the resulting mix.

$$m_2 \cdot Cp_{prod} \cdot T_2 + bleed \cdot m_1 \cdot Cp_{air} \cdot T_1 = m_3 \cdot Cp_{mix} \cdot T_3 \quad (3.13)$$

Solving this for the resulting temperature T_3 :

$$T_3 = \frac{m_2 \cdot Cp_{prod} \cdot T_2 + bleed \cdot m_1 \cdot Cp_{air} \cdot T_1}{m_3 \cdot Cp_{mix}} \quad (3.14)$$

3.8 Subroutine TURBINE

The subroutine TURBINE performs expansion calculations through the turbine. As in the compression process, polytropic efficiencies are utilized. For expansion from state 3 to state 4:

$$T_4 = T_3 \cdot \left(\frac{P_3}{P_4} \right)^{\frac{K \cdot \eta_{ad}}{C_p}} \quad (3.15)$$

Because bleed air has been introduced into the flow before the turbine is reached, the values of R and C_p cannot be calculated purely on the basis of exhaust products. Instead:

$$R = AP \cdot R_{air} + GP \cdot R_{prod} \quad (3.16)$$

$$C_p = AP \cdot C_{p_{air}} + GP \cdot C_{p_{prod}} \quad (3.17)$$

where AP and GP are the percentages of bleed air and products of combustion respectively. As C_p is temperature dependent the expansion process must be iterated as follows:

1. Using Eqn. 3.17 calculate C_p based on T_3
2. Using Eqn. 3.15 calculate T_4
3. Update C_p based on the average of T_3 and T_4
4. Goto 2

This process is iterated until successive C_p values are within 0.05%.

For the case of multi-spool gas turbines, the temperature drop across the high pressure turbine will be known, and it is the pressure at exit from the turbine which is desired. In this case, Equation 3.18 is used.

$$P_4 = P_3 \cdot \left(\frac{T_3}{T_4} \right)^{\frac{C_p}{R \cdot \eta_{st}}} \quad (3.18)$$

3.9 Subroutine PERFORM

Performance calculations for the cycles examined are carried out in the PERFORM subroutine. These calculations include specific power output, thermal efficiency, heat rate and specific fuel consumption, for both simple and combined cycles. Because the method

for calculating these variables varies depending on the cycle, the pertinent equations for the different cycles will be given in the appropriate section.

Figure 3.1: Cooling Bleed Effect on Performance

Figure 3.2: Specific Heat Temperature Variation

CHAPTER 4

Single-Shaft Gas Turbines

4.1 Background

The bulk of planned and installed gas turbine power is provided by large, single-shaft heavy frame units. With an ever growing need for power, these units are more attractive to electric utilities than smaller aero-derivatives. This is due in part to economies of size, as well as the better combined cycle performance of today's single-shaft units. Typically, a modern powerplant will employ anywhere from 2-6 of these units. Figure 4A shows an example of the ABB GT13E2 gas turbine. Large gas turbines such as this must be transported by rail. The layout of a combined cycle plant based on the GE Frame 9F is given in Figure 4B. The size of the gas turbine itself is seen to be relatively small when compared to the overall plant size.

The main players in the very competitive single-shaft gas turbine market are listed in Table 4.1, with performance data (simple cycle) taken from Gas Turbine World's 1993 Performance Specs.

Figure 4A: ABB GT13E2 Single-Shaft Gas Turbine
Reproduced from Reference 22

Figure 4B: GE Frame 9F Combined Cycle Installation
Reproduced from Reference 17

Table 4.1: Current Single-Shaft Gas Turbine Designs
Source: Reference 15

Model	Year	Shaft-Speed (rpm)	PR	Power (MW)
Westinghouse 501F	1989	3600	14.0	161
GE Frame 7F	1991	3600	15.0	159
Siemens V84.3	1992	3600	16.1	153
ABB GT11N2	1993	3600	14.6	109
GE Frame 9F	1991	3000	15.0	226
Siemens V94.3	1993	3000	16.1	222
ABB GT13E2	1993	3000	15.0	164
Siemens V64.3	1990	5400	16.1	62
ABB GT8C	1994	6200	15.7	53

As is clearly evident, current designs are very similar. It can also be seen that the units designed for the 60 Hz North American market (3600 rpm) are somewhat smaller than their 50 Hz European cousins (3000 rpm). The reason for this can be traced to Mach number considerations. For maximum efficiency, gas turbines are designed to run with the highest possible axial flow velocity, and hence blade tip speed. If the relative Mach number at rotor entry is allowed to rise too high, however, large aerodynamic losses can be incurred. This effectively fixes an upper limit for the blade tip speed, $U = \pi DN$, where D is the diameter at the blade tip and N is rotational speed. For 60 Hz gas turbines, the value of N is 1.2 times that of 50 Hz machines, 3600 versus 3000 rpm. To keep the same value of U , the diameter must be decreased by a factor of 1.2. For a given hub/tip ratio, the effective flow area of a gas turbine is proportional to D^2 , so a 60 Hz machine will have 1/1.44, or 70% of the flow area of a 50 Hz machine. With axial velocity fixed and air density known, entry mass flow is proportional to flow area. For 60 Hz single-shaft gas

turbine, therefore, both mass flow and output power are 30% less than a similarly configured 50 Hz gas turbine.

If, for a given level of technology, it is desired to produce a machine of lower power, the output shaft of the turbine must be geared to drive the load at the correct speed. This is exemplified by the ABB GT8C, which has a rotational speed of 6200 rpm. An advantage of this model, and the Siemens V64.3, is that the only adaptation necessary to modify it for 50 Hz or 60 Hz operation is a change of gear ratio on the output shaft.

For the generic example of this chapter, a four stage turbine is assumed, with the first of these stages being cooled by compressor delivery air. The complete list of assumptions used is presented in Table 4.2.

Table 4.2: Single-Shaft Cycle Assumptions

Efficiencies:	Compressor (Polytropic)	0.90
	Cooled Turbine (Polytropic)	0.88
	Mechanical	0.995
	Combustion	0.999
	Generator	0.98
Pressure Losses:	Intake	4 inches H ₂ O
	Combustion	3% of delivery pressure
	Exhaust (Simple Cycle)	6 inches H ₂ O
	Exhaust (Combined Cycle)	10 inches H ₂ O
Cooling Bleed:	Bleed Amount	10% of compressor delivery
	Bleed Delivered To	1st Stator 67%
		1st Rotor 33%

4.2 Simple Cycle

The most common cycle used by gas turbines for electrical power generation is the simple cycle. The output shaft of the turbine is used to drive the load, in this case an electrical generator. In most cases, the output shaft of large (>100 MW) single-shaft units is designed to run at synchronous speed, 3600 rpm (60 Hz) in North America and 3000 rpm (50 Hz) in the rest of the world. This eliminates the need for a gearbox or power turbine, required on smaller models.

A schematic layout of the single-shaft, simple cycle is given in Figure 4C, showing the station numbering convention used when performing calculations.

Figure 4C: Single-Shaft, Simple Cycle Configuration

The reason for the inclusion of additional stations 2-3 and 4-5 is to accommodate cooling bleed. The bleed air is removed after the compressor, reducing the mass flow at combustion entry, and returned after combustion exit, increasing mass flow and decreasing temperature before rotor entry. The work output from the simple cycle is given by:

$$W = \left[(m_5 \cdot C p_{s6} \cdot \Delta T_{s6}) - (m_1 \cdot C p_{12} \cdot \Delta T_{12} / \eta_m) \right] \cdot \eta_m \cdot \eta_g \quad (4.1)$$

The actual calculation is slightly more complicated because allowances for the cooling bleed must be made. Only a portion of the cooling air introduced in the first stage of the

turbine does work in the rotor. With work output known, thermal efficiency and specific fuel consumption are found:

$$\eta_{th} = \frac{W}{f \cdot m_3 \cdot LCV} \quad (4.2)$$

$$sfc = \frac{3600 \cdot f \cdot m_3}{W} \quad (4.3)$$

4.2.1 Simple Cycle Performance

The performance of the simple cycle will be looked at over a wide range of the designer's two major independent variables, pressure ratio (PR) and turbine inlet temperature (TIT).

The specific work output of the simple cycle is presented in Figure 4.1. For any given TIT, specific work rises with pressure ratio to a maximum, then decreases very slowly. This maximum is reached at a successively greater pressure ratio as TIT is increased, but for the most part the curves are flat. To maximize specific work, it is evident that one must use the highest possible value of turbine inlet temperature, especially as the marginal benefits to specific work increase as TIT increases. With the mass flow through the engine basically fixed by aerodynamic concerns, increasing TIT is the major way of increasing the power output of an industrial gas turbine. Most of today's engines are operating at a typical pressure ratio of 15 and TIT of 1500K. From Figure 4.1 this is seen to be close to the maximum on the specific work curve. The question is, was this a conscious decision or the by-product of other decisions? Due to the flat nature of the curve the designer has the option of going to a much higher PR while suffering only a minimal specific work penalty.

One of the principle goals of the designer of a gas turbine is to achieve the highest possible thermal efficiency. In theory, it can be shown that for an ideal system, efficiency is solely a function of the pressure ratio selected. In practice, however, the many losses present in a real cycle greatly impact this statement. Figure 4.2 shows how efficiency

varies as a function of pressure ratio for a range of TIT's. Clearly, although pressure ratio is the most important factor, TIT also affects the cycle efficiency. This effect is minimal at pressure ratios below 10, but becomes increasingly more important at higher PR's. As TIT is increased above 1500K, successive increases in TIT have less effect, with the curves beginning to collapse together. Most modern engines operate at TIT's of 1450-1550K, so pressure ratio is the dominant concern when looking for maximum efficiency. The efficiency curves begin to flatten out as pressure ratio is increased over 14. Indeed, at 1550K the difference in efficiency between pressure ratios of 14 and 24 is only 3 percentage points. While this is significant, it should be noted (see Table 4.1) that all of today's engines operate with pressure ratios in the very small range of 14-16. Obviously, there are other factors which must be examined.

Exhaust gas temperature might not initially seem like an important indicator of performance for an industrial gas turbine. Indeed, for the simple cycle examined in this section it is not. However, the next section will show that for the combined cycle, it is one of the most important variables when determining overall performance. Because of this, an initial look will be taken at the behaviour of EGT. Figure 4.3 shows how EGT varies for the range of conditions examined. As pressure ratio is increased, the greater expansion causes a larger temperature drop across the turbine, thereby lowering the exhaust gas temperature. As one would expect, EGT is also strongly dependent on the value of turbine inlet temperature used. Every 100K increase in TIT produces approximately a 55K increase in EGT, an effect that is relatively constant over a wide range of pressure ratios. Most modern heavy frame industrial gas turbines operate with an EGT of from 520-590°C. The line at 620°C represents the maximum temperature which can be used at entry to a heat recovery steam generator's unfired boiler. The significance of this will be discussed at length in the section on combined cycles.

In looking at the performance of the simple cycle, one main question remains. Why do the designers not utilize higher pressure ratios, which would give an increase in efficiency for virtually no penalty in specific work? It will be seen in the following section

that the gas turbine's performance in a combined cycle application is the major reason. Other factors, however, do play some role. One is the increased mechanical complexity required for very high pressure ratios on a single spool. The aerodynamics of the latter stages of such a compressor become critical, and if not carefully controlled could lead to surge or stall when operating at off-design conditions. The method of control involves the extensive use of variable geometry stators in the compressor, increasing the cost and complexity of the engine. Closely related to this is engine reliability. Obviously, the more complex the engine the more likely things are to go wrong. In this highly competitive age, with manufacturers providing performance guarantees for their products, and engines operating thousands of hours a year, reliability is extremely important. Finally, many of the major producers of industrial heavy frame units do not have access to the latest in aero-engine technology. Thus a jump to a much higher pressure ratio might involve a large increase in research and development costs, not to mention increased risk.

4.3 Combined Cycle

In the case of the simple cycle, the gases leaving the turbine are exhausted directly to the atmosphere. This represents a loss of energy, as these high temperature exhaust gases contain a large amount of surplus enthalpy. If, however, these gases are utilized to raise steam which is then passed through a steam turbine, significant gains can be made in both net output and net thermal efficiency. Of course, a plant designed for use in a combined cycle would have a larger capital cost than a simple cycle plant. Also, plant size would increase substantially due to the large size required for the heat recovery steam generator (HRSG). Despite this, most large modern plants designed for base load electrical power generation are designed as combined cycle units. Their annual hours of operation are high enough that the savings in fuel costs provided by the higher cycle efficiency far outweighs the initial cost disadvantage over the simple cycle.

For the most part, combined cycle plants first operate with only the simple cycle gas turbines operational, as their construction time is relatively short, usually less than a year. The steam turbine and HRSG, which require somewhat more construction time, usually come on line about two years later. However, one of the more recent implementations of the combined cycle involves the 'repowering' of older plants. This scheme takes an existing coal or oil fired plant, and replaces the old boiler system with gas turbines and a HRSG. Planning time is greatly reduced as an existing site is used. Because the original plant's steam turbine is utilized, capital costs are kept down and construction time is fairly quick. The net result of repowering is a modern, efficient plant with much higher capacity, for which no new ground had to be broken. A detailed study of the repowering of Virginia's Chesterfield plant is given by Catina et al. (1988).

When examining the performance of gas turbine cycles for power generation, therefore, the performance in a combined cycle role must not be ignored. For this thesis, it was desired to confine the actual cycle analysis to that of the gas turbine portion only. It was not desired to undertake an extensive thermodynamic examination of the variety of steam cycles possible. What was wanted was a way to quickly and accurately estimate the effect of adding a steam cycle to a gas turbine cycle, using known existing characteristics of the gas turbine.

Table 4.3: Steam Turbine Performance in Combined Cycle Applications
Source: References 11, 15, 19

Type	GT Exh Temp (°C)	GT Exh Flow (kg/s)	ST Power (MW)	ST Work (kJ/kg)
ABB GT24	610	376.0	86.0	228.7
GE Frame 7F	590	417.7	88.7	212.4
WE 501F	584	430.4	85.5	198.7
Siemens V84.3	550	433.1	78.0	180.1
Siemens V64.3	529	189.0	30.0	158.8
ABB GT11N2	525	375.1	56.6	150.9
ABB GT8C	517	179.1	26.5	147.9
GE LM1600	487	45.35	4.83	106.5
RR RB211	487	91.61	10.9	119.0
GE LM6000	477	124.7	13.0	104.2
PW FT8	443	85.26	7.58	88.9
RR Trent	436	159.4	14.0	87.8
GE LM5000	434	121.1	10.6	87.9

The method devised has the advantage of being both quick and simple. Table 4.3 lists parameters for several of today's gas turbines in combined cycle applications. Both heavy-frame and aero-derivatives are included. Where possible, the values of gas turbine exhaust flow and temperature, and steam turbine output, were obtained from the Gas Turbine World's 1993 Performance Specs. The fourth column contains a new variable created to aid in the analysis. It is a form of steam turbine specific work, but is referenced to the gas turbine exhaust mass flow, not the steam flow through the steam turbine. Figure 4D shows the result of plotting this new variable against gas turbine exhaust temperature, in degrees Celsius. A clear relation was seen to exist. Using the method of least squares a second-order polynomial was fitted to the data. This had the form of:

Figure 4D: Steam Turbine Performance in Combined Cycle Applications

$$ST_SpecificWork = (1.3E - 03) \cdot T^2 - (5.22E - 01) \cdot T + 64.418 \quad (4.4)$$

Thus for a known value of gas turbine exhaust flow and exhaust temperature, the value of steam turbine work can be found directly. It is not necessary to know any particulars of the steam cycle itself, such as the number of pressure levels and whether or not reheat is employed. The curve created is based on the assumption that best possible steam cycle is used, given the available exhaust gas temperature. The curve represents current state-of-the-art steam turbine technology. Advances in this technology, such as improved turbine efficiency, would see the curve of Figure 4D shift upward, although its shape would likely remain similar.

If a slightly less advanced steam cycle were to be utilized for a given EGT, the combined cycle performance curves generated will be slightly optimistic. The error introduced, however, is normally very small. When choosing between an advanced reheat steam cycle versus a more conventional non-reheat cycle, Warner and Nielsen (1993), as well as Bolland (1991), have shown that the difference in overall thermal efficiency is only about 0.3 percentage points.

With the steam turbine output now known, it is a simple matter of calculating net plant output.

$$Net_SpecificWork = GT_SpecificWork + \frac{m_6}{m_1} \cdot ST_SpecificWork \quad (4.5)$$

The mass flow ratio on the final term is necessary to relate it to the gas turbine entry mass flow. Previously, the steam turbine work of Eqn. 4.4 had been found using the gas turbine exhaust mass flow.

In summary, the method developed here for dealing with combined cycles uses known gas turbine exhaust conditions, and then 'grafts' a steam cycle on using the relationship of Equation 4.4.

An assumption introduced for the combined cycle was that the HRSG uses a conventional unfired boiler; that is, there is no additional burning of fuel in the boiler. In doing the steam cycle performance calculations this introduces a major limitation. Figure 4D shows how the steam turbine output increases with exhaust gas temperature. However, in practice, this trend does not continue indefinitely. To date, the highest exhaust gas temperature planned for a combined cycle plant is 610°C , as will be used on the ABB GT24/26. Current HRSG design does not permit temperatures higher than this to be utilized unless more expensive, advanced materials are used. There are no immediate plans to use such advanced designs for CCGT's. Thus, when doing calculations an artificial limit of 620°C was placed on the effective temperature at entry to the steam cycle. If the EGT of the gas turbine was higher than this value, it was assumed the exhaust gases were cooled to 620°C before entering the HRSG. In reality this would not be done as it represents wasted energy, but it was felt necessary to modify the calculations in this way so as not to create misleading performance data.

4.3.1 Combined Cycle Performance

Specific work output for the combined cycle is shown in Figure 4.4. Turbine inlet temperature is the major factor affecting the output, both in its direct effect on gas turbine output and its secondary effect on steam turbine output. At higher TIT's (above 1300K) there is seen to be a maximum in the specific work output, the pressure ratio of which increases with TIT. This maximum is primarily due to the program imposed restriction on the temperature at steam cycle entry, discussed in the previous section. After this peak the specific work output decreases steadily with increasing pressure ratio. This is in contrast to the simple cycle, where the output curves were relatively flat over a wide range.

Figure 4.5 shows how combined cycle efficiency varies with pressure ratio for the range of turbine inlet temperatures examined. The results are sharply different from those of the simple cycle. TIT, which had very little effect on simple cycle efficiency, is seen to

be the dominant factor in combined cycle efficiency. This is due to the increased value of EGT which results from a high TIT. In turn, this provides much better steam conditions and hence greater steam turbine output. Pressure ratio also is a significant factor with overall efficiency, after reaching a maximum, falling as the pressure ratio is increased. This is the reverse of what occurred with the simple cycle, and shows how significant the steam portion of a combined cycle is. The increase in simple cycle efficiency that comes with a higher pressure ratio is turned into a significant decrease in combined cycle efficiency due to the lower EGT, which lowers steam turbine output. As many single-shaft gas turbines are now built as CCGT's, this answers the question posed earlier, asking why single-shaft units did not use higher pressure ratios to improve their simple cycle performance. When examining the performance of a typical heavy frame unit with a PR of 15 and TIT of 1500K, the combined cycle efficiency is 53%, which is 17 percentage points higher than that of the simple cycle at 36%.

4.4 Regenerative Cycle

The regenerative, or heat exchange, cycle is an alternate way of increasing the efficiency of a simple cycle gas turbine. It was used fairly successfully on a number of older designs, but is no longer common today.

From Equation 3.1, it can be seen that the fuel/air ratio is directly related to the temperature rise in the combustion chamber. If the temperature at entry to the combustion chamber can be raised, the fuel flow will be lowered with no change in work output, thus raising efficiency. This is the approach employed by the regenerative cycle. Instead of being immediately exhausted to atmosphere, the gas turbine's hot exhaust gases are first used to preheat the compressor delivery air, before entry to the combustion chamber. This energy transfer takes place in a heat exchanger. A schematic of the cycle is given in Figure 4E.

Figure 4E: Single-Shaft, Regenerative Cycle Configuration

Conservation of energy in the heat exchanger gives:

$$m_7 \cdot C_{p_{78}} \cdot \Delta T_{78} = m_3 \cdot C_{p_{34}} \cdot \Delta T_{34} \quad (4.6)$$

This equation cannot be solved immediately as both T_4 and T_8 are unknown. A term commonly used to measure heat exchanger performance is its effectiveness, ε . This is a measure of the actual temperature rise of the compressor air to its maximum possible value. This occurs when the temperature of the air at entry to the combustion chamber is equal to that of the exhaust gases leaving the turbine.

$$\varepsilon = \frac{\Delta T_{34}}{\Delta T_{37}} \quad (4.7)$$

With this value specified for a given heat exchanger, the values of T_4 , at entry to the combustion chamber, and T_8 , at exit from the heat exchanger, are found:

$$T_4 = \varepsilon \cdot \Delta T_{37} + T_3 \quad (4.8)$$

$$T_8 = \frac{m_7 \cdot C_{p_{78}} \cdot T_7 - m_3 \cdot C_{p_{34}} \cdot \Delta T_{34}}{m_7 \cdot C_{p_{78}}} \quad (4.9)$$

Calculations of this type are carried out in the HEATEX subroutine. The process for T_8 involves iteration as the average C_p values over the given temperature range must be used.

There are limitations and new losses introduced with the employment of heat exchangers. Obviously, if the temperature of the gas turbine exhaust gases is lower than that of the compressor delivery air, there is no point in employing a heat exchanger. It would only serve to lower the air temperature at entry to the combustion chamber, increasing the fuel/air ratio required and decreasing efficiency. The friction caused by the airflow passing through the heat exchanger is a significant source of pressure losses, both on the air-side and the gas-side. Typical values of these factors were used in the analysis and are given in Table 4.4, as is the value of heat exchanger effectiveness, chosen as per the state-of-the-art.

Table 4.4: Heat Exchanger Assumptions

Parameter	Value
Effectiveness	0.9
Air-side Pressure Loss $\Delta P_{34}/P_3$	0.03
Gas-side Pressure Loss $\Delta P_{78}/P_8$	0.05

4.4.1 Regenerative Cycle Performance

A major performance limitation of the heat exchange cycle is the temperature at entry to the heat exchanger itself. This temperature is limited due to metallurgical and stress considerations. Figure 4.6 shows how this temperature varies over the range of pressure ratios and turbine inlet temperatures tested. With present heat exchanger technology this temperature is limited to approximately 900K. From Figure 4.6 this can be seen to seriously restrict the designer's options at low pressure ratios, although further analysis will show this is not as severe as might first be thought. An increase in allowable

temperature to 1000K would, however be desirable as it would allow almost all of the practical working range to be utilized.

The specific work output of the cycle is given in Figure 4.7. The curves are very similar to those of the simple cycle, although shifted down slightly. The heat exchanger does not affect compressor work required, and does not affect the temperature at entry to the turbine. The effect it does have is to lower the available expansion ratio across the turbine, reducing the turbine work, and hence overall output. This is because of the two pressure losses associated with the operation of a heat exchanger, one which reduces the pressure at entry to the turbine and the other which raises the back pressure at exhaust. Figure 4.7 shows how the specific work curves rise steadily at first, then becomes virtually flat. The effect of the 900K temperature limit imposed by the heat exchanger is shown by the dark line, with the area to the left of this line being unusable. In terms of work output, though, the flat nature of the curves means the restricted operating range is not particularly significant. High turbine inlet temperatures do, however, require the use of higher pressure ratios to avoid this restriction, increasing compressor complexity.

Figure 4.8 shows the thermal efficiency performance of the regenerative cycle. At lower pressure ratios the regenerative cycle greatly outperforms the simple cycle, which is included on the figure for comparison (1500K case). As pressure ratio is increased, the various regenerative efficiency lines eventually intersect that of the simple cycle. This intersection represents the temperature of the turbine exhaust gas entering the heat exchanger being identical to that of the compressor delivery air entering the heat exchanger. Above this PR the heat exchanger lowers the temperature of the compressor delivery air, thereby lowering efficiency to a value below that of the simple cycle.

At low pressure ratios (less than 4) the efficiency curves for the various TIT's are relatively collapsed, but at higher pressure ratios TIT is major factor in cycle efficiency, much more so than with the simple cycle. At a pressure ratio of 10, for example, each 100K increase in TIT results in an efficiency gain of two percentage points. Each efficiency curve quickly reaches a maximum at a particular pressure ratio, then gradually

declines. This maximum value increases with increasing TIT, but the range is fairly small, being between 6 and 9. In practice, it is not possible to operate at this maximum efficiency value, due to the heat exchanger temperature limit. The area to the left of the dark line in Figure 4.8 is unusable. This restriction is more serious than the specific work case, but only at the higher TIT values, where it causes a loss of about two percentage points in efficiency.

From the performance curves two possible heat exchange cycles were selected as valid possibilities and their performance compared to that of a typical simple cycle machine (e.g. GE Frame 7F). This is shown in Table 4.5. HE 1 represents a very conservative choice of engine parameters, while HE 2 is a more advanced design using the highest possible turbine inlet temperature.

Table 4.5: Proposed Regenerative Cycle Configurations

Cycle	PR	TIT (K)	EGT (°C)	Specific Work (kJ/kg)	Efficiency (%)
Simple	15	1550	571	378	36.5
HE 1	8	1400	354	266	40.6
HE 2	12	1550	417	350	42.5

In terms of a first cost analysis the two regenerative cycles would be more expensive than the simple cycle due to the additional cost of the heat exchanger. However this difference would be lessened by the simpler compressor used on the regenerative designs, especially the HE 1 design. In terms of performance, the 4 and 6 percentage point gain in efficiency by the HE 1 and HE 2 designs over the simple cycle are significant. This leads to an interesting question: Why has no one built a regenerative cycle machine in recent years? The principle reason is the rapid growth of the combined cycle, which can boost the efficiency of single-shaft gas turbines to 54%. Although a combined cycle plant

is larger and more expensive than a regenerative cycle plant, the large difference in performance more than justifies the investment. The low exhaust temperatures of the regenerative cycle means they are not well suited for a combined cycle, and even if this were done, the resulting plant would be extremely complex and expensive. A niche area where regenerative cycles could work is in dry climates, where the feed water required by a combined cycle is unavailable. However, this small market alone could not support the required development costs. Also, the latest aero-derivatives are producing efficiencies within two points of the HE 2 design proposed above. In effect, the regenerative cycle has been squeezed out of future markets by advanced aero-derivatives from below and the latest combined cycles from above.

4.5 Program Validation

In order to check that program assumptions were reasonable, a brief comparison of program output with published data was performed. The model chosen for comparison was the GE Frame 7F, which is a very recent (1991) design producing 159 MW. Performance data for this model was obtained from Reference 15. Results of the comparison are given in Table 4.6.

Table 4.6: Validation Case: GE Frame 7F, TIT=1560K, PR=15.0

		Simple Cycle		Combined Cycle	
	EGT (°C)	Efficiency (%)	Spec Work (kJ/kg)	Efficiency (%)	Spec Work (kJ/kg)
GE Data	589	35.9	388	54.5	588
Program	577	36.5	384	55.2	581

Considering the program model is of a generalized single-shaft unit, the two sets of data are in surprisingly close agreement. With other models, the correlation may well be less exact, but the basic program assumptions would seem to have been validated.

• 4.6 Future Single-Shaft Designs

In this section, current and future performance capabilities will be examined with the aid of 'mesh' performance graphs. On these graphs the operating locations of some of today's latest gas turbines are shown, based on their published pressure ratios and turbine inlet temperatures. The pressure ratios of modern gas turbines are readily obtained from many sources. However, getting accurate values for turbine inlet temperature is somewhat more difficult. Part of the reason is that manufacturers tend to be fairly secretive about their designs, especially their latest technology. Another reason is the very definition of turbine inlet temperature, which can vary significantly. In this thesis, TIT is taken as the maximum cycle temperature, also known as the firing temperature, which occurs at exit from the combustion chamber. In some circles, though, turbine inlet temperature is given as the temperature at entry to the first rotor of the turbine. This version yields a value for TIT anywhere from 50-100K lower than the previous definition, as it takes into account the cooling bleed delivered at the first stator. The distinction between definitions is often not stated when a TIT value is quoted, so some judgement is involved when assessing published values. A final reason for variance in turbine inlet temperature numbers is that depending on the application or user, a range of turbine inlet temperatures can be used for the same type of gas turbine. A gas turbine used for base load service, for example, would use a lower value of TIT than one for peak load service, to extend engine life. The values used on the figures are believed to be the correct base load firing temperatures.

From the previous analysis, it is apparent that large single-shaft gas turbines gas turbines face a design compromise between simple cycle and combined cycle operation. As most of the single-shaft units now entering service are in CCGT installations, it is the

performance in this area which is the more important in the optimization process. Figure 4.9 shows the thermal efficiency versus specific work mesh for the combined cycle. The TIT lines are darkened to aid in the clarity of the graph. The foldover effect on the right half of the figure represents heat being wasted due to an overly high exhaust gas temperature, which cannot be fully utilized in the unfired boiler of the HRSG. If a fired boiler were used, or one with advanced high temperature materials, the graph would look somewhat different. Note that such devices would represent a substantial increase in plant cost, and no CCGT plants currently utilize EGT's over 600°C. Included on the graph are the operating locations of three modern single-shaft units; the ABB GT11N2, Siemens V84.3 and GE Frame 7F. The GT11N2 is conservative in design, using a relatively low turbine inlet temperature, while the other two models are very much state-of-the-art, with combined cycle efficiencies of about 55%. All the designs shown operate at pressure ratios just below the crest in their respective turbine inlet temperature lines. Although the V84.3 and Frame 7F have similar turbine inlet temperatures, the GE model has the performance edge due to its lower pressure ratio.

In designing a new, advanced technology single-shaft gas turbine, the turbine inlet temperature to be used is selected first, then the optimum pressure ratio is found. A future advanced design in 5 years or so might well use a TIT value of 1650K. From Figure 4.9, the optimum pressure ratio to maximize thermal efficiency is seen to be about 18 or 19, giving a thermal efficiency of over 57%. This is 2 to 3 percentage points higher than the best current designs, with specific output increasing by about 20%.

The simple cycle mesh is given in Figure 4.10. The advanced design proposed above yields a gain of 2 thermal efficiency percentage points over existing models, so it is apparent that most of the improvement in the combined cycle performance actually comes from simple cycle gains. The physical changes required to uprate an existing model of PR=15, TIT=1550K, to the proposed advanced cycle of PR=19, TIT=1650K, are not particularly extensive. They could be accomplished by adding additional compressor stages to manufacturers' current designs, and employing better blade cooling. A 60 Hz

unit such as this, perhaps a GE Frame 7G or a Siemens V84.4, could produce almost 190 MW in simple cycle configuration. Westinghouse have recently launched their latest 50 Hz model, the 501G, with a reported turbine inlet temperature of almost 1700K.

It must be emphasized that the prediction for an advanced future design is based on maintaining the cycle assumptions of current designs, which may introduce some error. However, this error would be relatively small, for a variety of reasons. While the higher TIT would presumably require more cooling flow, advances in cooling technology would minimize this increase. As a new model will most likely be an advancement of an existing design, aerodynamic blade improvements would not tend to be extensive. Finally, pressure and mechanical losses have been fairly constant for some time, and are unlikely to change much in the near future.

Figure 4.1: Simple Cycle Specific Work Output

Figure 4.2: Simple Cycle Thermal Efficiency

Figure 4.3: Simple Cycle Exhaust Gas Temperature

Figure 4.4: Combined Cycle Specific Work Output

Figure 4.5: Combined Cycle Thermal Efficiency

Figure 4.6: Heat Exchanger Entry Temperature

Figure 4.7: Regenerative Cycle Specific Work Output

Figure 4.8: Regenerative Cycle Thermal Efficiency

Figure 4.9: Single-Shaft Combined Cycle Mesh

Figure 4.10: Single-Shaft Simple Cycle Mesh

CHAPTER 5

Aero-Derivative Gas Turbines

5.1 Background

As its name implies, an aero-derivative is a modification of an existing aircraft engine to adapt it for shaft power applications. Aside from physical changes to the engine components, the turbine inlet temperature used on an aero-derivative is usually somewhat lower than the aircraft version, for reasons of reliability and engine life. Aero-derivative gas turbines are available in a wide variety of sizes and configurations, depending on the required application. Traditionally, they have been employed as an emergency power supply where their unmatched ability to reach full power from a black start in less than two minutes is crucial. Other areas where aero-derivatives have found wide acceptance include peak load supply, augmenting the large base load plants of electrical utilities. For off-shore drilling platforms, the small size and excellent power density make the aero-derivative gas turbine a popular choice. Figure 5A shows a typical aero-derivative power station, in this case for the Rolls-Royce Trent. The very compact nature of the station is evident, with a footprint of only 88x44 ft.

Figure 5A: Rolls-Royce Trent Plant Layout
Reproduced from Reference 11

Table 5.1 shows several of the recent designs available on the aero-derivative market, with most of the simple cycle data taken from Gas Turbine World's 1993 Performance Specs. In the layout column, dd refers to a direct-drive machine, and pt a power turbine model. The range of engine configurations is seen to be much more extensive than was the case for single-shaft heavy frame units. This is in part due to the diversity of uses for aero-derivatives, and also to the large number of aero-engine designs suitable for conversion to industrial use. It should be noted, though, that there is a clear upper limit to the power which can be achieved by an aero-derivative. This limit is set by parameters of the aircraft version of the engine, in particular, the amount of flow through the high pressure core.

Table 5.1: Current Aero-Derivative Gas Turbine Designs
Source: References 11, 15

Model	Year	Layout	PR	Power (MW)
RR Trent	1995	3-spool, dd	35.0	51.2
GE LM6000	1992	2-spool, dd	29.8	41.0
GE LM5000	1985	2-spool, pt	24.8	34.4
RR RB211	1974	2-spool, pt	21.0	27.2
PW FT8	1990	2-spool, pt	20.3	25.4
GE LM2500	1973	1-spool, pt	18.8	22.8
GE LM1600	1987	2-spool, pt	22.3	13.4

In terms of mechanical configuration, there are several possible approaches. Figure 5B shows those which have been used to date by aero-derivatives. The earliest designs consisted of a single-spool gas generator feeding a power turbine (i), which was sized to rotate at the correct synchronous speed. The most advanced technology using this layout is the GE LM2500, which first appeared in 1973, based on the GE TF39 turbofan. The

most common aero-derivative design in use today is a twin-spool gas generator feeding a power turbine (ii), as used by such types as the RB211 and LM5000. The latest offering from GE, the LM6000, uses the twin-spool concept, but drives the load generator directly from the LP shaft (iii). Rolls-Royce also uses the direct-drive method for the planned industrial version of the Trent, in this case in a triple-spool layout (iv). The implications of doing away with a separate power turbine and driving the load directly are presented in Section 5.2.

Figure 5B: Aero-Derivative Configurations

Rolls-Royce is the only manufacturer to have gone with a triple-spool design for an aero-derivative. There are certain advantages to this approach, as compared to the more conventional twin-spool design. The better aerodynamic stage matching afforded by the extra spool gives slightly more efficient performance, as the HP and IP spools can rotate at their own optimum speeds. This allows the use of fewer stages, and reduces the requirement for regulating devices such as variable stators, common on twin-spool models. The use of fewer stages translates into a shorter, more compact engine which allows tighter tip clearances and is said to have reduced performance deterioration.

5.2 Method of Driving the Load (Generator)

In the past, almost all aero-derivative gas turbines for industrial use were equipped with a power turbine to drive the load. This was necessary due to the high rotational speed of the aircraft version's low pressure shaft. Examples of this technology are the General Electric LM5000 and the Rolls-Royce RB211. The LM5000 is derived from the CF6-50C2 aircraft engine, and uses a new power turbine and LP turbine. The industrial version of the triple-spool RB211 was created by removing the LP shaft completely, then feeding a new power turbine with a gas generator comprising the existing IP and HP spools. The very latest high thrust turbofans, however, have a low pressure shaft speed close to the synchronous speed required for 60 Hz power generation, 3600 rpm. This has enabled aero-derivative designs to do away with the power turbine and direct-drive the load from the low pressure shaft. The two newest designs given in Table 5.1, the GE LM6000 and the RR Trent, both utilize this technique, which has many implications for the prospective operator.

5.2.1 Advantages of Direct-Drive

- 1 The elimination of the power turbine provides a large savings in capital cost. For the most part, previous aero-derivatives had the LPC, HPC and HPT common with the

aircraft version. However, for the industrial version the LPT had to be completely redesigned into a new LPT and a separate power turbine. Due to the much smaller production numbers of an industrial aero-derivative (at most 10-20% of its aircraft cousin) these unique parts represented 40-50% of its cost. When using the direct-drive method, on the other hand, it is possible to utilize the aircraft LPT intact, as on the LM6000, or only slightly modified, as is the case for the Trent. This utilization of existing aircraft engine modules realizes large economy of scale benefits.

2. Performance for a direct-drive machine is slightly better. Typically, the LPT and power turbines used on industrial aero-derivatives exhibit lower performance levels than their aircraft derived components, because their smaller production numbers do not justify as much development cost.
3. The exhaust ducting for the direct-drive machine, and thus the total installation size, is somewhat smaller than for a power turbine version. This is because the power turbine's larger diameter requires more ducting. For applications where space is at a premium, such as off-shore platforms, this is an important consideration.
4. The use of a power turbine for an aero-derivative gas turbine usually requires the installation of a costly lubrication system, whereas the existing system can be used if the load is driven directly from the output shaft of the aero-engine.
5. Because the entire LP shaft is running at synchronous speed, it is possible to connect the load to either end of a direct-drive machine. When the load is driven from the compressor (cold) end, recovery of exhaust heat is simplified when a combined cycle is used.

5.2.2 Disadvantages of Direct-Drive

1. Even at reduced loads, it is necessary for the LP shaft of a direct-drive machine to maintain a constant 3600 rpm. To do this requires the use of components not typically required for a power turbine version. First, variable inlet guide vanes must be used

before the LP compressor. These modulate the airflow into the compressor to maintain synchronous speed at different loads, typically down to about 70% of design values. If the gas turbine must operate at very low powers, it is necessary to incorporate some sort of blowoff valve. On a two-spool engine this would normally be located between the LP and HP compressors. The effect of blowoff is to greatly decrease turbine work, at the expense of efficiency.

2. Because the machine is designed from the outset to operate at 3600 rpm, it is not very flexible in adapting to different roles. With a power turbine version, a redesigned power turbine could be used for operation at different speeds.

5.3 Aero-Derivative Cycle Assumptions

The cycle calculations done for aero-derivative configurations will be done based on the assumption that the load is driven directly off the LP shaft, using a twin-spool design. This is the method shown in Figure 5B(iii). If a power turbine were used the performance values would be slightly lower.

One other decision which had to be made was in what ratio to split the compression between the low and high pressure spools. The FT8 aero-derivative has an overall pressure ratio of $OPR = 4.8 \times 4.2 = 20.3$. This corresponds to 55% of the total compression temperature rise occurring in the LP compressor and 45% in the HP compressor. The LM6000, though, has $OPR = 4 \times 7.5 = 30$, which corresponds to 41% of the compression temperature rise in the LPC and 59% in the HPC. For the analysis of this chapter a split of 45% of total temperature rise for the LP compressor and 55% of total temperature rise for the HP compressor is used. A brief analysis showed that the performance differences incurred when changing the LP compressor's percentage of total temperature rise from 30% to 60% is very small. The complete list of assumptions used is given in Table 5.2

Table 5.2: Aero-Derivative Cycle Assumptions

Efficiencies:	Compressors (Polytropic)	0.90
	Cooled Turbine (Polytropic)	0.88
	Uncooled Turbine (Polytropic)	0.89
	Mechanical	0.995
	Combustion	0.999
	Generator	0.98
Pressure Losses:	Intake	4 inches H ₂ O
	Combustion	6% of delivery pressure
	Exhaust (Simple Cycle)	6 inches H ₂ O
	Exhaust (Combined Cycle)	10 inches H ₂ O
Cooling Bleed:	Bleed Amount	10% of HPC delivery
	Bleed Delivered To	HPT 1st Stator 67%
		HPT 1st Rotor 33%
Compression Split	LP Compressor	45% of temperature rise
	HP Compressor	55% of temperature rise

5.4 Simple Cycle

When used for power generation, most aero-derivative gas turbines have traditionally employed the simple cycle. This is mainly due to the types of applications for which they are used. For peak load or emergency supply, the annual running hours are low, so the increased capital cost associated with a combined cycle is not justified. In the off-shore industry space is at a premium, so the large increase in surface area required by the addition of a combined cycle is not feasible.

The basic layout of the twin-spool aero-derivative is shown in Figure 5C, along with the station numbering system used. As with the case of the single-shaft design, the

additional stations present (3-4 and 5-6) are required to allow for cooling bleed take-off and addition.

Figure 5C: Twin-Spool, Simple Cycle Configuration

In order to find the temperature drop across the HP turbine, an enthalpy balance must be performed with the HP compressor:

$$m_2 \cdot C_{p_{23}} \cdot \Delta T_{23} = m_6 \cdot C_{p_{67}} \cdot \Delta T_{67} \quad (5.1)$$

which gives:

$$T_7 = T_6 - \frac{m_2 \cdot C_{p_{23}} \cdot \Delta T_{23}}{m_6 \cdot C_{p_{67}}} \quad (5.2)$$

Because $C_{p_{67}}$ is itself a function of T_7 , an iterative process is used to solve for T_7 :

- 1 Assume an initial value of T_7
2. Calculate $C_{p_{67}}$ based on the average of T_6 and T_7
3. Using Eqn. 5.2 calculate T_7
4. Goto 2

This iteration is repeated until successive values of T_7 are within suitable tolerances. For the twin-spool engine the work output is given by:

$$W = \left[(m_7 \cdot C_{p78} \cdot \Delta T_{78}) - (m_1 \cdot C_{p12} \cdot \Delta T_{12} / \eta_m) \right] \cdot \eta_m \cdot \eta_s \quad (5.3)$$

The actual calculation is slightly modified to allow for the inclusion of the cooling bleed, returned in the high pressure turbine. With work output known, thermal efficiency and specific fuel consumption are found:

$$\eta_{th} = \frac{W}{f \cdot m_4 \cdot LCV} \quad (5.4)$$

$$sfc = \frac{3600 \cdot f \cdot m_4}{W} \quad (5.5)$$

5.4.1 Simple Cycle Performance

The performance of the simple cycle will be looked at for turbine inlet temperatures from 1300-1700K and for pressure ratios from 16-40. This range covers all of the major designs of today and those expected to come along in the next several years. Although the performance calculations are performed over a wide range of engine parameters, one thing must be kept in mind. An aero-derivative industrial gas turbine is exactly that; a derivative of an existing aircraft engine. Thus a designer looking to meet a certain requirement is not free to run a wide variety of simulations, determine the best choice of pressure ratio and turbine inlet temperature, and send the results to the shop. They must work with existing designs, with some flexibility in configuration but not much. The power of an aero-derivative is basically fixed by the airflow through the gas generator, which is almost the same as the aircraft engine.

Aero-derivative performance in the area of specific work is examined in Figure 5.1. Specific work is seen to be most influenced by the value of turbine inlet temperature (TIT) used. At a pressure ratio of 16, increasing TIT from 1300K to 1700K increases specific work output by 190%. A similar increase in TIT when operating at a pressure ratio of 40

gives a 250% increase in specific work. As pressure ratio is increased, the specific work output drops for turbine inlet temperature below 1700K. The rate of decrease is more pronounced at lower TIT values. The 1700K curve is basically flat, with output at a pressure ratio of 40 less than 5% below that of the maximum.

Figure 5.2 looks at aero-derivative thermal efficiency. Whereas efficiency was almost exclusively a function of pressure ratio for the single-shaft unit, the aero-derivative efficiency is also strongly affected by the choice of turbine inlet temperature, due to the higher pressure ratios used. This effect becomes more important the higher the pressure ratio. At a pressure ratio of 16 an increase in TIT from 1300K to 1700K brings about an increase of 2 percentage points in thermal efficiency. For the case of a PR of 40, the same increase in TIT results in a gain of over 8 percentage points. The benefit of going to higher TIT's decreases the higher one goes, but remains significant. Pressure ratio also has an effect on efficiency. At the lowest TIT tested the efficiency curve reaches a maximum at a PR of 28. At 1400K the efficiency curve has just reached its maximum value at a PR of 40. However, for the higher TIT values the efficiency continues to increase with increasing pressure ratio. Thus if a high enough turbine inlet temperature is used (1600-1700K) the benefits from increased efficiency might outweigh the increased mechanical complexity required at a pressure ratio of, say, 40. At 1500K, though, the increase in efficiency when going from a pressure ratio of 28 to 40 is only one percentage point, not worth the added capital cost and complexity, or the minor loss in specific work output.

As noted in Chapter 4, the exhaust gas temperature from the outlet turbine is a key factor in the performance of combined cycle applications. Exhaust gas temperatures for the aero-derivatives examined in this chapter are shown in Figure 5.3, again with the 620°C upper limit for the HRSG shown by the horizontal line. Although the curves are slightly flatter, the trends shown are very similar to that of single-shaft engines, with each 100K increase in turbine inlet temperature giving a 55K increase in exhaust gas temperature, at a given pressure ratio. The values of exhaust gas temperature, however, are significantly lower than those of single-shaft units. This is due to the higher pressure

ratios used by aero-derivatives, in the range of 20-35. The resultant increased expansion ratio across the turbines produces a greater temperature drop, thereby lowering the exhaust temperature. A typical aero-derivative adapted from a large aircraft turbofan might have a pressure ratio of 30 with a turbine inlet temperature of 1530K. This gives an exhaust gas temperature of 440°C, over 100K below that of the latest single-shaft units. The effect of this decrease will be examined in the next section.

5.5 Combined Cycle

As mentioned earlier, aero-derivatives usually employ the simple cycle for most applications. However, units utilizing the combined cycle have become increasingly popular, for various reasons.

1. The reliability of aero-derivatives have increased to the point where they are quite capable of supplying base load power, running many more hours annually than had been the case in the past. When these running hours become high enough, it makes sense to use the most efficient cycle possible, hence the use of the combined cycle.
2. There has been a large increase in the number of small independent cogeneration plants supplying excess generating capacity to electric utilities. By providing this excess at times when the actual plant requirements are low, more efficient utilization of the plant's gas turbines is achieved. The increased running hours for the gas turbines mean that a high efficiency combined cycle may be the best choice, with long-term fuel savings more than offsetting the increased capital costs.

The method for handling the addition of a combined cycle to an aero-derivative is the same as that used for the single-shaft gas turbine of Chapter 4. Utilizing the gas turbine exhaust conditions, Figure 4D will be used to obtain the steam turbine output. This should

remain an accurate method, as 6 of the data points used to construct the curve of Figure 4D were supplied by aero-derivative designs.

5.5.1 Combined Cycle Performance

In examining the performance of aero-derivatives in a combined cycle environment, one key question had to be answered. How seriously would the lower exhaust gas temperatures of the aero-derivatives affect their performance values? As for the single-shaft gas turbine, the maximum usable temperature at entry to the heat recovery steam generator is 620°C, assuming an unfired boiler. In actual fact, this restriction only came into effect for one data point, at a pressure ratio of 16 and a TIT of 1700K (see Figure 5.3).

The specific work performance of the aero-derivative combined cycle is given in Figure 5.4. Specific work declines steadily as pressure ratio is increased at a given turbine inlet temperature. The benefits of a high turbine inlet temperature are again evident, with this benefit slightly increased with each successive increase in TIT.

Figure 5.5 shows the combined cycle efficiency for aero-derivatives. The shapes of the curves are very similar to those of specific work output. As pressure ratio is increased, the efficiency decreases. Thus the increase in simple cycle efficiency afforded by higher pressure ratios is more than offset by the reduced steam cycle performance, which is a direct result of the lower exhaust gas temperature. The rate of decrease is more severe at lower turbine inlet temperatures, at the higher TIT's the drop in efficiency when going from a pressure ratio of 16 to 40 is less than 5 percentage points. Figure 5.5 clearly shows the benefit of using the highest possible TIT. Each 100K increase is worth about 4 percentage points in thermal efficiency. This is achieved through increases in both gas turbine and steam turbine performance.

5.6 Program Validation

To verify that the aero-derivative assumptions used were reasonable, program output was compared with published data for the Pratt and Whitney FT8. The FT8 is a 1990 design based on the extremely successful JT8D family of aircraft turbofans. Comparison results are given in Table 5.3.

Table 5.3: Validation Case: PW FT8, TIT=1433K, PR=20.3

	EGT (°C)	Simple Cycle		Combined Cycle	
		Efficiency (%)	Spec Work (kJ/kg)	Efficiency (%)	Spec Work (kJ/kg)
PW Data	443	38.1	304	48.7	386
Program	454	38.3	311	49.8	404

The numbers are in fairly good agreement, the simple cycle case more so than the combined cycle. The combined cycle output given by the program is approximately 4% higher than the published value. This is mainly due to the higher exhaust temperature given by the program trial, which gives increased output from the steam turbine. Of course, it must again be stated that the program output is only a rough estimate of the real engine's performance, and it is not intended to exactly match the PW data. Among other things, the cooling bleed amounts used on the actual engine are not known, nor is the turbine inlet temperature used, 1433K, certain.

5.7 Future Aero-Derivative Designs

Unlike the large single-shaft units, combined cycle performance is not a driving factor in aero-derivative designs. Being derived from aircraft engines, they are already optimized for simple cycle operation. Figure 5.6 shows the thermal efficiency versus

specific work mesh for the simple cycle aero-derivative. Three of the most recent current designs are plotted at their correct PR-TIT locations. These designs span the range from the conservative FT8 to the leading edge technology of the Trent, due to enter service in 1995. The Trent's efficiency is 4 percentage points higher than the FT8, and its specific work output is 20% greater.

When considering how aero-derivative designs may change in the coming years, one must first look at the new aircraft engines coming on-line. In the large engine market, the big three manufacturers are General Electric, Pratt and Whitney and Rolls-Royce. These companies have based their future plans around the GE90, PW4000, and RR Trent turbofans respectively. With the base design set, future versions of these aero-engines will likely not see much of an increase over their existing pressure ratios of 30-40. The more likely way to meet higher thrust requirements is to increase mass flow and/or turbine inlet temperature. With this in mind, a possible future aero-derivative of the next 5 years is shown on Figure 5.6. The pressure ratio is 36, with turbine inlet temperature increased to 1650K, about 100K higher than current models. The performance of such a design would be 43% thermal efficiency, which is 1-2 points greater than the Trent or LM6000. Specific work would increase 10-20%, to 420 kJ/kg.

Figure 5.7 shows the combined cycle mesh for aero-derivatives, with the same three engine models. As with the single-shaft example, the TIT lines are darkened to help clarify the graph. It is interesting to note that the difference in efficiency from the FT8 to the Trent is approximately the same as it was for the simple cycle case. Performance in the steam cycle area is approximately the same for all three engines, despite the Trent's more advanced technology. The reason for this is that the different models all have very similar exhaust gas temperatures. The Trent's higher TIT is offset by its higher pressure ratio, resulting in near identical steam conditions for the three models. The advanced design suggested earlier, with a pressure ratio of 36 and a TIT of 1650K, is also shown on Figure 5.7. It delivers 55% thermal efficiency and 540 kJ/kg of specific work output. This efficiency is 3-4 points higher than the Trent/LM6000, whereas it was 1-2 points higher

for the simple cycle. Thus the relative increase in combined cycle efficiency and output is a combination of improvements in both the gas turbine and steam cycles.

With the advanced design of above, it would be possible to deliver power levels of 60-65 MW, if based on the large turbofans about to go into service. Currently, the mid-size of aero-derivatives are represented by such types as the RB211 and FT8, both relatively old designs beginning to show their age. It would appear there is a potential market for a new, modern aero-derivative based on an aircraft engine a step below the large turbofans. Suitable candidate engines for such a design include the RB211-535, PW2000 and CFM56. These could provide the latest in technology for power requirements ranging from 20-35 MW in simple cycle operation.

Figure 5.1: Simple Cycle Specific Work Output

Figure 5.2: Simple Cycle Thermal Efficiency

Figure 5.3: Simple Cycle Exhaust Gas Temperature

2 of/de 2

PM-1 3½"x4" PHOTOGRAPHIC MICROCOPY TARGET
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PRECISIONSM RESOLUTION TARGETS

Figure 5.4: Combined Cycle Specific Work Output

Figure 5.5: Combined Cycle Thermal Efficiency

Figure 5.6: Aero-Derivative Simple Cycle Mesh

Figure 5.7: Aero-Derivative Combined Cycle Mesh

CHAPTER 6

Reheat Cycle Gas Turbines

6.1 Background

The fuel/air ratio of most modern gas turbines is very low, with a value of 0.02 being common. Most of the air passing through the gas turbine is not involved in the combustion process. The reason the fuel/air ratio must be kept low is to keep the turbine inlet temperature at an acceptable value, as burning too much fuel in the airflow results in a turbine inlet temperature which will exceed the material limits of the turbine. If, however, the gases exiting the combustion chamber are partially expanded through the turbine, their temperature will be lowered. It is then quite possible to burn more fuel in the flow, before completing the expansion process through the turbine. This process is known as the reheat cycle, and although it first appeared in a gas turbine design as early as 1948, no new reheat gas turbine has been built for many years. The design concept has been revived, however, by ABB Power Generation.

The two new ABB designs utilizing the reheat cycle are shown in Table 6.1, with performance data taken from an ABB paper by Neuhoﬀ and Thorén (1994). The ABB

GT24 is designed for the North American market, while the GT26 is a 1.2x scale-up of the GT24, for use in countries with 50 Hz grids. Although neither machine has yet reached the prototype stage, orders have been placed, and the first units should be on-line within two years.

Table 6.1: Current Reheat Cycle Designs
Source: Reference 19

Model	Year	Speed (rpm)	PR	Power (MW)
ABB GT24	1995	3600	30	165
ABB GT26	1996	3000	30	240

The reheat cycle has many potential advantages and disadvantages over the simple cycle.

6.1.1 Reheat Cycle Advantages

1. Because it is possible to burn more fuel for the same entry airflow, power output and thus power density are increased for a reheat gas turbine.
2. At the design stage, the selection of where the turbine split takes place can be used to control exhaust gas temperature, an important factor for combined cycle applications.
3. With a conventional simple cycle, the chief way to increase power is to raise the turbine inlet temperature. However, it has been shown that harmful NO_x emissions increase greatly with temperature. With the reheat cycle, power can be increased without increasing TIT over that of existing conventional designs.

6.1.2 Reheat Cycle Disadvantages

1. The reheat cycle is more complex than the simple cycle, with its requirement for an extra combustion chamber. This added complexity means increased cost and the possibility of lower reliability.
2. For a given pressure ratio and turbine inlet temperature, the reheat cycle will give a slightly lower value of thermal efficiency than the simple cycle. This is because the reheat cycle involves adding a less efficient cycle to the simple cycle, less efficient as it occurs over a smaller temperature range. Note that this lower efficiency is for a stand alone gas turbine. The thermal efficiency in combined cycle applications will be looked at in detail later in this chapter.

6.2 Reheat Theory

A schematic of the reheat cycle for a single-shaft unit is given in Figure 6A, along with the station numbering used. As in previous chapters, the additional stations from 2-3, 4-5 and 7-8 are required to take into account cooling bleed takeoff and addition.

Figure 6A: Single-Shaft Reheat Cycle Configuration

Note that although the cycle being examined is a single-shaft design, the terms high and low pressure turbine are used, more normally associated with a twin-spool design.

These terms will be used throughout to distinguish between the portion of the turbine before the reheat combustion chamber (high pressure turbine) and the portion after (low pressure turbine). The total output power delivered by the reheat cycle shown in Figure 6A is:

$$W = [(m_5 \cdot C_{p_{56}} \cdot \Delta T_{56} + m_8 \cdot C_{p_{89}} \cdot \Delta T_{89}) - (m_1 \cdot C_{p_{12}} \cdot \Delta T_{12} / \eta_m)] \cdot \eta_m \cdot \eta_g \quad (6.1)$$

When determining thermal efficiency and specific fuel consumption, the fuel flows for both the main combustion chamber and the reheat combustion chamber must be accounted for. The method for calculating the fuel/air ratio, f , for the main combustion chamber is identical to that given in Chapter 3. For the reheat chamber this method must be slightly modified, because the mixture entering the reheat chamber is not pure air, as was the case for the main chamber. Instead, it is a combination of the products from the initial combustion and the cooling air returned in the high pressure turbine. The equation for the fuel/air ratio of the reheat chamber, f_2 , is:

$$f_2 = \frac{C_{p_{prod1}} \cdot (T_6 - 298) - C_{p_{prod2}} \cdot (T_7 - 298)}{C_{p_{prod2}} \cdot (T_7 - 298) + n_{cv} \cdot LCV} \quad (6.2)$$

The value of $C_{p_{prod1}}$ is evaluated at the average of T_6 and 298K, while $C_{p_{prod2}}$ is evaluated at the average of T_7 and 298K. Because $C_{p_{prod2}}$ is itself a function of fuel/air ratio f_2 , an iterative process must be used:

1. Assume an initial value of f_2
2. Calculate $C_{p_{prod1}}$ at $(T_6 + 298)/2$
3. Calculate $C_{p_{prod2}}$ at $(T_7 + 298)/2$
4. Using Eqn. 6.2 calculate f_2
5. Goto 3

When the difference between successive values of f_2 is less than 0.05% the iteration process is ended. This calculation is performed in the program subroutine FUELFLO2. With f_2 now found the thermal efficiency and specific fuel consumption are calculated.

$$\eta_{th} = \frac{W}{(f \cdot m_3 + f_2 \cdot m_6) \cdot LCV} \quad (6.3)$$

$$sfc = \frac{3600 \cdot (f \cdot m_3 + f_2 \cdot m_6)}{W} \quad (6.4)$$

The method of handling the bleed flow remains similar to that given in Chapter 3, but the amount of bleed air has been increased, as there are now two combustion chambers. The HP turbine following the main combustion chamber still receives 10% of the compressor delivery flow, while the LP turbine following the reheat chamber receives an additional 10% of the compressor delivery flow. A full 20% of the compressor delivery flow is therefore utilized for cooling purposes.

For the main combustion chamber, the value of combustion pressure loss remains at 3% of delivery pressure. However, this value needs some modification for the reheat combustion chamber, for two reasons. First, the temperature rise in the reheat chamber tends to be quite a bit less than that of the initial chamber, which reduces the losses associated with exothermic mixing. Second, because the temperature at entry to the reheat chamber is higher than that of the initial chamber, the amount of flame-stabilizing devices required is reduced, lowering air resistance. In fact, ABB does away with a flame-holder completely in the reheat chamber of its new GT24 and GT26 models. Because of these factors, the pressure loss in the reheat combustion chamber is assumed to be 1.5% of delivery pressure, half of that used for the main chamber. The complete list of assumptions used for the calculations of this chapter is presented in Table 6.2.

Table 6.2: Reheat Cycle Assumptions

Efficiencies:	Compressor (Polytropic)	0.90
	Cooled Turbine (Polytropic)	0.88
	Mechanical	0.995
	Combustion	0.999
	Generator	0.98
Pressure Losses:	Intake	4 inches H ₂ O
	Combustion: Main	3% of delivery pressure
	Combustion: Reheat	1.5% of delivery pressure
	Exhaust (Simple Cycle)	6 inches H ₂ O
	Exhaust (Combined Cycle)	10 inches H ₂ O
Cooling Bleed:	HP Bleed Amount	10% of compressor delivery
	HP Bleed Delivered To	HPT 1st Stator 67%
		HPT 1st Rotor 33%
	LP Bleed Amount	10% of compressor delivery
	LP Bleed Delivered To	LPT 1st Stator 67%
		LPT 1st Rotor 33%

6.3 New Independent Variables

With a reheat cycle, the designer has two new independent variables to work with. These are the turbine inlet temperature following reheat combustion, and the location of the reheat process in the expansion process (where the split in the turbine occurs)

In previous chapters the advantages of using as high a TIT as possible has been clearly seen, both to obtain maximum power and, to a lesser degree, maximum thermal efficiency. For this reason it will be assumed for the cycle calculations of this chapter that the two TIT's, one following the main combustion chamber and the second following the reheat chamber, are identical in value.

The choice for the second new variable, the location of the reheat process, is not immediately obvious, nor are its effects on performance. Along with the usual variables of turbine inlet temperature and pressure ratio, the performance of the cycle using a variety of split locations for the turbine will be examined. The method for describing the location of the reheat process is done with the aid of Figure 6B, which shows the relevant portion of the temperature-entropy diagram. This diagram is shown in simplified form, in that it does not include losses or bleed effects.

Figure 6B: Reheat Cycle Temperature-Entropy Diagram

The temperature drop from 3-4 occurs during the expansion in the high pressure turbine. The gases are then reheated at constant pressure from 4-5, before being expanded again in the low pressure turbine, giving the temperature drop from 5-6. The expansion from 3-4' represents the simple cycle case, with no reheating. The variable used to define the reheat location is the reheat temperature fraction (RTF), and is calculated as follows:

$$\text{Reheat_Temperature_Fraction} = \frac{T_3 - T_4}{(T_3 - T_4) + (T_5 - T_6)} \quad (6.5)$$

As the reheat temperature fraction approaches a value of 0, the reheat cycle approaches the simple cycle in performance, with almost no reheating involved. At a value of 0.5, the temperature drops in the high and low pressure turbines are equal. The range of reheat locations examined includes reheat temperature fractions from 0.1 to 0.7.

6.4 Reheat Cycle Performance

The introduction of a third independent variable, the location of reheat in the expansion process, greatly increases the amount of potential data produced. In order to examine the performance trends while keeping graphs to a minimum, two methods are used. First, the performance of the reheat cycle when using a fixed turbine inlet temperature of 1500K is examined. Specific work and thermal efficiency are plotted against the reheat temperature fraction. This is done for a range of pressure ratios. The second method of analysis is similar to the first, but involves the use of a fixed pressure ratio of 28 instead of a fixed TIT, and examining results over a range of TIT's

6.4.1 Fixed TIT=1500K

For a turbine inlet temperature of 1500K, Figure 6.1 shows how specific work varies with the location of reheat. For each value of pressure ratio examined there is a very clear maximum reached. This peak occurs at slightly higher temperature fractions for higher pressure ratios, but the range is very small, all being approximately 0.4. For a system with no losses, it can be shown that maximum work occurs when the expansion (and hence temperature drop) is evenly split between the high and low pressure turbines, a reheat temperature fraction of 0.5 (see Reference 10, p.37). The losses present in a real system have slightly modified the location of this maximum value. Over most of the

temperature fraction range a higher pressure ratio gives increased work output, but the curves begin to collapse rapidly when the pressure ratio gets above 24. At very low temperature fractions, less than 0.2, the lower pressure ratios give higher work output.

Thermal efficiency for the 1500K case is examined in Figure 6.2. It is seen to steadily decrease as the reheat temperature fraction is raised. Efficiency is improved by going to a higher pressure ratio, but this effect is less significant at higher temperature fractions.

Based on the above analysis, it is evident no designer would design a reheat gas turbine in which the reheat temperature fraction is greater than that for maximum specific work, as shown in Figure 6.1. This is because not only will specific work output be decreased, but the thermal efficiency will also be lower (Figure 6.2). It must be noted, however, that the analysis so far has dealt solely with the performance of the reheat cycle alone. The above statements may be modified when the reheat cycle is examined in a combined cycle environment.

Figure 6.3 shows how the exhaust gas temperature varies with reheat temperature fraction. The EGT increases steadily as the temperature fraction increases. This is due to the decrease in expansion ratio across the turbine after reheat, which results in a lower temperature drop and a correspondingly higher EGT. This fact will play a very important role in combined cycle applications, especially when one considers that the maximum temperature desirable at entry to the heat recovery steam generator is in the order of 620°C, for an unfired boiler.

6.4.2 Fixed $PR=28$

Figure 6.4 shows how specific work varies with reheat temperature fraction at a fixed pressure ratio of 28. There are large benefits to be gained by increasing the turbine inlet temperature, with each successive increase in TIT bringing a slightly larger relative increase in specific work. In fact the output at 1700K is twice that at 1300K. The specific

work curves are fairly flat, but do reach a maximum. In all cases this maximum occurs within a small temperature fraction range of 0.4-0.45, slightly less than the 0.5 predicted by theory for a system with no losses.

Thermal efficiency, Figure 6.5, declines steadily as the temperature fraction is increased. Increasing turbine inlet temperature raises the thermal efficiency for a given temperature fraction, but this effect is very small when higher values of TIT are reached.

Values of EGT produced are given in Figure 6.6. For most of the temperature fractions over 0.3 the EGT is too high to be fully utilized by the HRSG, whose upper limit is shown by the dark horizontal line. This will affect combined cycle performance.

6.5 Reheat Combined Cycle Performance

6.5.1 Fixed $TIT=1500K$

The specific work output of a reheat combined cycle is given in Figure 6.7. Although a maximum is reached for each pressure ratio examined, this peak is much less defined than the non-combined cycle case. For low reheat temperature fractions (less than 0.2) lower pressure ratios give the best work output, but this situation is reversed as the temperature fraction increases. In terms of an absolute maximum, the highest pressure ratios give the highest work outputs, but the difference is small.

Thermal efficiency for the combined cycle is examined in Figure 6.8. The trends are very similar to those shown for specific work. Efficiency rises to a maximum then decreases. Again, at low temperature fractions the lower pressure ratios perform better, but are soon overtaken. The higher pressure ratios produce the highest values of thermal efficiency, rising from 55% at a PR of 16 to 57% at a PR of 32.

The reason for the decreasing efficiency at higher temperature ratios can be traced to the high exhaust gas temperatures they produce, which was shown in Figure 6.3. In most cases, a temperature fraction over 0.3 will result in an exhaust gas temperature well over the desirable limit for the heat recovery steam generator.

It is very interesting to note that for the reheat combined cycle with $TIT=1500K$, any given pressure ratio reaches its maximum value of both specific work and thermal efficiency at approximately the same value of temperature fraction. This is quite unlike the case of the plain reheat cycle.

6.5.2 Fixed $PR=28$

Figure 6.9 shows the variation in specific work for a fixed pressure ratio of 28. The gains made by increasing turbine inlet temperature are quite significant. For each TIT, a peak is reached in the specific work curve at a particular value of reheat temperature fraction. This value ranges somewhat depending on the TIT used, but is generally around 0.4. For TIT's 1500K and higher the specific work curve is almost flat, so specific work would not be the determining factor in deciding where to accommodate reheat into the cycle.

Thermal efficiency performance, Figure 6.10, is quite different from that of specific work. For each turbine inlet temperature, a maximum in thermal efficiency is reached, but the location of this peak varies substantially. For the 1700K case, the maximum has already been reached slightly before a temperature fraction of 0.1. Each such successive decrease of 100K in TIT moves the location of the thermal efficiency maximum another 0.1 to the right. It should also be noted that the value of this maximum steadily decreases for each decrease in TIT, going from a high of over 58% for the 1700K case to 52% for the 1300K case.

As was the case for the fixed TIT analysis, the reason for the thermal efficiency reaching a maximum and thereafter steadily decreasing is due to the high exhaust gas temperatures produced, Figure 6.6. If the temperature fraction selected yields an EGT of over $620^{\circ}C$, the heat content over this amount is assumed wasted, reducing efficiency.

6.6 Comparison with ABB GT24

In the case of the reheat cycle, validation of the results given by the program are somewhat more difficult than was the case for the single-shaft and aero-derivative gas turbines. There are several reasons for this:

1. At the present time, there are no reheat gas turbines in production, nor have there been for many years.
2. The current planned ABB GT24/26 models will not enter service until 1995/96, so any performance data provided by ABB is based on prediction.
3. When doing a comparison, there are several cycle values which must be known. Manufacturers keep some of these values confidential, especially with new designs, so some educated guesses must be made.

Despite these obstacles, a reasonable effort was made to compare program output with the ABB GT24, a reheat gas turbine producing 165 MW in its simple cycle configuration, and 251 MW in a combined cycle. A schematic of the ABB GT24 is shown in Figure 6C, with Figure 6D showing the layout of the combustors, flow going from right to left.

The three major inputs necessary for the program are the gas turbine pressure ratio and turbine inlet temperature, and the location of the reheat chamber in the expansion process. The overall pressure ratio of the GT24 is quoted as 30, but an official value for turbine inlet temperature could not be found. ABB are traditionally fairly conservative in this area, so a value of 1525K is assumed, slightly below the state-of-the-art. The location of reheat is not known, but the design uses a 1-stage HP turbine before the reheat chamber and a 4-stage LP turbine after. The following analysis will look at a range of possible reheat locations, and determine the most likely based on the mechanical layout given above.

Figure 6C: ABB GT24 Reheat Gas Turbine
Reproduced from Reference 19

Figure 6D: ABB GT24 Combustor Layout
Reproduced from Reference 19

Figure 6.11 shows how thermal efficiency, specific work and exhaust gas temperature vary with reheat temperature fraction. Performance figures from ABB on the GT24 give a thermal efficiency of 37.5%, specific work of 447 kJ/kg and an exhaust gas temperature of 610°C. Examining Figure 6.11, this corresponds with 25-35% of the total temperature drop occurring before reheat, in the HP turbine. Further investigation revealed that a value of 30% gave the best correlation with ABB data.

Next, the combined cycle performance of the GT24 was looked at. Figure 6.12 shows how specific work and thermal efficiency vary. There are clear maximums reached by both curves, with the efficiency curve reaching its peak at a temperature fraction of 0.3. Efficiency then falls off, due to the inability of the following steam cycle to fully utilize the high exhaust gas temperature. The specific work curve is relatively flat over a range of temperature fractions from 0.3 to 0.5. These observations would tend to agree with the earlier choice of a reheat temperature fraction of 0.3, when the reheat cycle was examined without a following steam cycle.

A comparison of program performance data with that quoted by ABB is given in Table 6.3, for both simple reheat and combined cycle cases.

Table 6.3: Comparison Case: ABB GT24, TIT=1525K, PR=30.0

		Simple Cycle		Combined Cycle	
	EGT (°C)	Efficiency (%)	Spec Work (kJ/kg)	Efficiency (%)	Spec Work (kJ/kg)
ABB Data	610	37.5	447	58.0	680
Program	613	37.2	440	57.0	673

The correlation between the two sets of numbers is very good, but again, the question is whether the value of 30% of the total temperature drop occurring before reheat is correct. The 30% corresponds to a pressure ratio of 2.54 before reheat and 11.00

after. These values make sense in light of the 1-stage high pressure turbine and 4-stage low pressure turbine used by the GT24.

In designing for optimum performance in the combined cycle area, Figure 6.12 shows the design compromises which had to be made are not that difficult, as both work output and thermal efficiency peak at approximately the same reheat location. For the simple cycle of Figure 6.11 this is not the case, with increasing work output being linked with decreasing thermal efficiency.

6.7 Future Reheat Cycle Designs

With the first new reheat gas turbine in many years still over a year away from entering service, it is perhaps a bit early to predict on future designs, or if in fact there will be new designs of this type. Other manufacturers will certainly be watching the progress of the GT24/26 with great interest. Perhaps the one striking advantage the reheat cycle offers over conventional single-shaft units is its ability to control exhaust gas temperatures, to a large degree independently from the chosen pressure ratio and turbine inlet temperature. This control is provided, at the design stage, by varying the location of reheat in the turbine. In addition, at the operational level, EGT control can be accomplished by changing the fuel flow, and hence the turbine inlet temperature, of the reheat combustion chamber.

Figure 6.1: Reheat Cycle Specific Work Output, TIT=1500K

Figure 6.2: Reheat Cycle Thermal Efficiency, TIT=1500K

Figure 6.3: Reheat Cycle Exhaust Gas Temperature, TIT=1500K

Figure 6.4: Reheat Cycle Specific Work Output, PR=28

Figure 6.5: Reheat Cycle Thermal Efficiency, PR=28

Figure 6.6: Reheat Cycle Exhaust Gas Temperature, PR=28

Figure 6.7: Reheat Combined Cycle Specific Work Output, TIT=1500K

Figure 6.8: Reheat Combined Cycle Thermal Efficiency, TIT=1500K

Figure 6.9: Reheat Combined Cycle Specific Work Output, PR=28

Figure 6.10: Reheat Combined Cycle Thermal Efficiency, PR=28

Figure 6.11: ABB GT24 Simple Cycle Predicted Performance

Figure 6.12: ABB GT24 Combined Cycle Predicted Performance

CHAPTER 7

Conclusions

7.1 Conclusions

1. Based on comparisons made with existing gas turbine data, the programs used in the analysis portion of the thesis very accurately model the performance of real cycles, with losses being accounted for.
2. The mechanical configuration of today's single-shaft gas turbines has been optimized for combined cycle applications. This has kept pressure ratios at values of 14-16, in order to provide a high exhaust gas temperature for the following steam cycle. The steam cycle boosts overall efficiency and work output by as much as 50% compared to a simple cycle gas turbine. Future advances in turbine inlet temperature will see the use of slightly higher pressure ratios, to keep the exhaust gas temperature at an optimum value. A proposed advanced design could reach almost 39% simple cycle and 58% combined cycle thermal efficiency, using a pressure ratio of 19 and a turbine inlet temperature of 1650K.

3. The regenerative cycle, while capable of providing a thermal efficiency about 5 percentage points above that of the simple cycle single-shaft units, is unlikely to be used in designs of the near future. The inherently low exhaust gas temperature of the regenerative cycle makes it impractical to add a following steam cycle. As a result, it suffers badly in comparison to the latest combined cycle units. At the same time, the latest aero-derivatives are able to provide nearly the same efficiency as the regenerative cycle, but do so while operating on the simple cycle.
4. Aero-derivatives, being closely related to aircraft engines, have their designs optimized for the simple cycle. The high pressure ratios used means that the exhaust gas temperature for modern aero-derivatives is from 100-150K lower than single-shaft gas turbines, which results in less efficient combined cycle performance. Still, there are clear benefits to employing a combined cycle, if running hours are great enough. Thermal efficiency and specific work are increased by about 20% over the simple cycle. A proposed advanced design could produce 43% simple cycle and 55% combined cycle efficiency. Such a design could be based on one of the large turbofans about to enter service on aircraft such as the Boeing 777.
5. The implementation of the reheat cycle, on ABB's new GT24/26 design, removes the pressure ratio restriction previously faced by single-shaft units. The reheat cycle allows the use of high pressure ratios (30 on the GT24/26) while at the same time keeping the exhaust gas temperature high, providing excellent combined cycle performance. This performance increase can be obtained without the requirement of higher turbine inlet temperatures, helping keep NO_x levels down.

7.2 Recommendations for Future Work

This thesis examines the gas turbine cycles in use today, or soon to enter production, for power generation purposes. Several other advanced cycles have been

proposed for future designs, but not yet committed to production. Future work could examine the performance of these cycles, which include the humid air turbine (HAT) cycle, and versions of the intercooled-regenerative (ICR) cycle, which is currently being proposed as an advanced naval propulsion system.

The use of alternate fuels for gas turbines, reducing reliance on natural gas, is another area of interest. Experimental gas turbine plants utilizing coal gasification methods have already been built, and hold promise for the future.

The method of dealing with the combined cycle in this thesis was to base performance on that achieved by existing designs. By doing the actual thermodynamic steam cycle calculations, it would be possible to see how the various options such as the number of steam pressure levels affect performance.

Finally, it would be beneficial to look at off-design performance of the cycles examined, not just the design point performance at standard ISO conditions. In particular, the off-design performance of the reheat cycle could prove quite interesting, as varying the second turbine inlet temperature provides a new form of control.

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APPENDIX A

Table A1: Chemical Composition of Air

Element	Molecular Mass (kg/kmol)	Vol. Analysis	Grav. Analysis
Nitrogen, N ₂	28.013	0.7809	0.7553
Oxygen, O ₂	31.999	0.2095	0.2314
Argon, Ar	39.948	0.0093	0.0128
Carbon Dioxide, CO ₂	44.010	0.0003	0.0005

Table A2: Polynomial Coefficients for the Specific Heat of Air
Source: Reference 8

Symbol	Temperature Range	
	200K-800K	800K-2200K
C ₀	+.24336328E+00	+.19075549E+00
C ₁	-.32921648E-04	+.12752498E-03
C ₂	+.47395140E-07	-.54651988E-07
C ₃	+.10126885E-09	+.89378182E-11
C ₄	-.89883655E-13	+.00000000E+00
C ₅	+.00000000E+00	+.00000000E+00

Table A3: Polynomial Coefficients for Specific Heat of Exhaust Products
Source: Reference 1

Symbol	θ_{Cp}^C	Symbol	θ_{Cp}^H
CPC ₀	-.39277299E+00	CPH ₀	+.21027252E+01
CPC ₁	+.14186817E-02	CPH ₁	-.93246352E-03
CPC ₂	-.15339020E-05	CPH ₂	+.21762758E-05
CPC ₃	+.76790508E-09	CPH ₃	-.90091221E-09
CPC ₄	-.14392196E-12	CPH ₄	+.11172113E-12

APPENDIX B

The following sections of the appendix, B1-B4, list the main portions of the program code used for the analysis part of this thesis. Because many of the subroutines used are common to several or all of these programs, the subroutines are listed in a separate section, B5. All programs were written using Microsoft Fortran 5.1.

B1 Single-Shaft Gas Turbine Program Code

Program SINGLE-SHAFT GAS TURBINE

```
implicit none
real ncc,ng,nm,ncl,ntl,lcr,ltr,f
real T(0:12),P(0:12),W(0:12),Q(0:12),CP(8),R(8)
real fuel,sfc,LCV,ComPOWER(4),GP,AP,Cper,Hper,HR,ntherm
real Power,SIPower,SIHR,Rair,bleed,IntL,DiffL,CombL,ExhL,SR
real SIsfc,ccSIsfc,ccntherm,GTW,TW
character*16 S(0:20)
character*50 Outfile,Infile,RunTitle
integer i,More,EffType,TempType,OutType
data nm,ncc,ng,Rair/0.995,0.999,0.98,0.06856/
```

```
S(0)='Ambient      '
S(1)='Comp Entry   '
S(2)='Comp Exi     '
S(3)='Comb Entry   '
S(4)='Turb Entry    '
S(5)='Stator1 Exit  '
S(6)='Rotor1 Exit   '
S(7)='Stator2 Exit  '
S(8)='Rotor2 Exit   '
S(9)='Stator3 Exit  '
S(10)='Rotor3 Exit  '
S(11)='Stator4 Exit  '

```

```

S(12)='Rotor4 Exit      '

2   Print *,' '
    Print *,'Enter name of output file: (eg. c:\trent\out.dat)'
    Read (*,795) Outfile
    Print *,' '

    Open (unit=7,file=Outfile)

    print *,'Do you wish to have: 1. Multi-concise output'
    print *,'                               2. Single-full output'
    read (*,796) OutType

    call FILEREAD(infile,RunTitle,EffType,ncl,ntl,TempType,T,P,W,
$LCV,Cper,Hper,bleed,IntL,DiffL,CombL,ExhL)

4   Print *,'Enter compressor ratio:'
    Read *,lcr
    Print *,'Enter TIT (K):'
    Read *,T(4)

C***** This step converts the supplied ambient temperature
C***** to Kelvin.

    If (TempType.EQ.1) then
      T(0)=T(0)
    Else if (TempType.EQ.2) then
      T(0)=T(0)+273.15
    Else if (TempType.EQ.3) then
      T(0)=(T(0)+459.67)*0.555556
    Endif

    T(1)=T(0)
    P(1)=(1-IntL)*P(0)

C***** It is necessary to assume an initial value for
C***** fuel/air ratio f here, as the MASSFLOW subroutine
C***** requires it as input. When the actual f has been determined
C***** the massflows will be updated.

    f=0.02
    call MASSFLOW(W,f,bleed)
    P(2)=P(1)*lcr
    call COMPRESS(1,1,ncl,P,T,CP,R,Rair,EffType)

    T(3)=T(2)
    P(3)=(1-DiffL)*P(2)
    P(4)=(1-CombL)*P(3)
    c. 11 FUELFLOW(ncc,T,f,LCV,Cper,Hper)

```

C***** Now that the correct fuel/air ratio for the given conditions
 C***** has been determined, the MASSFLOW subroutine is called
 C***** again, to update those massflows after combustion occurs.

call MASSFLOW(W,f,bleed)

C***** As bleed air is introduced in several points in the
 C***** turbines, it is necessary to know both the pure air and
 C***** combusted air percentages (AP,GP). These are used in CP and
 C***** R calculations.

P(12)=(1+ExhL)*P(0)
 ltr=P(4)/P(12)
 SR=ltr**0.25

GP=0.5*(W(4)/W(4)+W(4)/W(5))
 AP=1.0-GP
 call BALANCE(2,4,bleed,f,0.,W,T,GP,AP,Cper,Hper)
 P(5)=P(4)
 call LPTURB(5,2,ntl,f,W,CP,R,Rair,P,T,SR,Cper,Hper,EffType)

GP=0.5*(W(4)/W(6)+W(4)/W(7))
 AP=1.0-GP
 call BALANCE(2,6,0.,f,0.,W,T,GP,AP,Cper,Hper)
 P(7)=P(6)
 call LPTURB(7,3,ntl,f,W,CP,R,Rair,P,T,SR,Cper,Hper,EffType)

GP=0.5*(W(4)/W(8)+W(4)/W(9))
 AP=1.0-GP
 call BALANCE(2,8,0.,f,0.,W,T,GP,AP,Cper,Hper)
 P(9)=P(8)
 call LPTURB(9,4,ntl,f,W,CP,R,Rair,P,T,SR,Cper,Hper,EffType)

GP=0.5*(W(4)/W(10)+W(4)/W(11))
 AP=1.0-GP
 call BALANCE(2,10,0.,f,0.,W,T,GP,AP,Cper,Hper)
 P(11)=P(10)
 call LPTURB(11,5,ntl,f,W,CP,R,Rair,P,T,SR,Cper,Hper,EffType)

call PERFORM(W,T,P,Q,ComPOWER,CP,Power,nm,ng,SIPower,Fuel,f,
 \$sfc,LCV,HR,SIHR,ntherm,GTW,TW,SIsfc,ccSIsfc,ccntherm)

If (OutType.EQ.2) Goto 6

write (7,770) lcr
 write (7,771) T(4)
 write (7,772) GTW,TW
 write (7,773) SIsfc,ccSIsfc
 write (7,774) ntherm,ccntherm
 write (7,775) T(12)-273.15
 write (7,711)


```

print *, 'Do you wish to: 1. Continue trial'
print *, '                2. Finish trial'
read (*,796) More
print *, ' '
If (More.EQ.1) Goto 4
If (More.NE.1) Goto 8

6   write (7,711)
    write (7,701)
    write (7,702) Outfile
    write (7,703) RunTitle
    write (7,711)
    write (7,711)
    write (7,711)
    write (7,720)
    write (7,721)
    write (7,711)
    Do 12 i=0,12
12  write (7,710) S(i),W(i),P(i),T(i),Q(i)
    Continue
    write (7,711)
    write (7,711)
    write (7,711)
    write (7,711)
    write (7,725) Power,SIPOWER
    write (7,726) Fuel
    write (7,727) sfc
    write (7,728) lcr
    write (7,760) HR,SIHR
    write (7,761) ntherm
    write (7,711)
    write (7,711)
    write (7,711)
    write (7,711)
    write (7,730)
    write (7,762)
    write (7,711)
    write (7,731) lcr,ncl,CP(1),ComPOWER(1)
    write (7,735) ltr,ntl,CP(2),ComPOWER(2)

700 format(10X,F12.4)
701 format(10X,'SINGLE-SHAFT GAS TURBINE')
702 format(10X,'File Name: ',(A50))
703 format(10X,'Run Title: ',(A50))
710 format(10X,(A16),4(F12.2))
711 format(10X)
720 format(10X,' Station           W           P           T
$      WRT/P')
721 format(10X,'                (lbm/s)      (psia)      (K
$)')
725 format(10X,'Output Power=',F8.1,' Hp      ( ',F7.3,' MW )')
726 format(10X,'Fuel Flow=',F11.1,' lbm/hr')
```

```

727 format(10X,'sfc=',F17.4,' lbm/Hp hr')
728 format(10X,'OPR=',F17.3)
730 format(10X,'Component      PR      ETA      Cp      Power')
731 format(13X,'LPC',F14.3,F10.4,F9.4,F14.1)
735 format(13X,'HPT',F14.3,F10.4,F9.4,F14.1)
737 format(13X,'LPT',F14.3,F10.4,F9.4,F14.1)
760 format(10X,'Heat Rate=',F11.1,' BTU/Hp hr      ( ',F7.1,' BTU/KW hr
$)')
761 format(10X,'Efficiency=',F10.4)
762 format(42X,'(CHU/lbm K)      (Hp)')
770 format(10X,'PR=',F6.1)
771 format(10X,'TIT=',F6.0,'K')
772 format(10X,'SP=',F8.1,' kJ/kg',10X,F8.1,'kJ/kg')
773 format(10X,'SFC=',F6.3,' kg/kWh',10X,F6.3,'kg/kWh')
774 format(10X,'Eff=',F6.3,16X,F6.3)
775 format(10X,'EGT=',F5.0,'C')
776 format(10X,'bld=',F5.2)
795 format(A30)
796 format(I1)

```

```

8      Close(Unit=7)

```

```

print *, 'Do you wish to: 1. Run another trial'
print *, '                  2. Exit from program'
read (*,796) More
print *, ' '
If (More.EQ.1) Goto 2

```

```

End

```

B2 Regenerative Gas Turbine Program Code

Program SINGLE-SHAFT REGENERATIVE GAS TURBINE

```

implicit none
real ncc,ng,nm,ncl,ntl,lcr,ltr,f,fold
real T(0:12),P(0:12),W(0:12),Q(0:12),CP(4),R(4)
real fuel,sfc,LCV,ComPOWER(4),GP,AP,Cper,Hper,HR,ntherm
real Power,SIPower,SIHR,Rair,bleed,IntL,DiffL,CombL
real HEaL,HEgL,HEeff
character*16 S(0:20)
character*50 Outfile,Infile,RunTitle
integer i,More,EffType,TempType,OutType
data nm,ncc,ng,Rair/0.995,0.999,0.98,0.06856/

S(0)='Ambient'
S(1)='Comp Entry'
S(2)='Comp Exit'
S(3)='Heat Ex Entry'
S(4)='Comb Entry'
S(5)='Turb Entry'
S(6)='Rotor Entry'
S(7)='Turb Exit'
S(8)='Heat Ex Exit'

2  Print *,' '
   Print *,'Enter name of output file: (eg. c:\trent\out.dat)'
   Read (*,795) Outfile
   Print *,' '

   Open (unit=7,file=Outfile)

   print *,'Do you wish to have: 1. Multi-concise output'
   print *,'                        2. Single-full output'
   read (*,796) OutType

   call FILEREAD(infile,RunTitle,EffType,ncl,ntl,TempType,T,P,W,
    $LCV,Cper,Hper,bleed,HEeff,IntL,DiffL,HEaL,CombL,HEgL)

4  Print *,'Enter compressor ratio:'
   Read *,lcr
   Print *,'Enter TIT (K):'
   Read *,T(5)

C***** This step converts the supplied ambient temperature
C***** to Kelvin.

```

```

      If (TempType.EQ.1) then
        T(0)=T(0)
      Else if (TempType.EQ.2) then
        T(0)=T(0)+273.15
      Else if (TempType.EQ.3) then
        T(0)=(T(0)+459.67)*0.555556
      Endif

      T(1)=T(0)
      P(1)=(1-IntL)*P(0)

C***** It is necessary to assume an initial value for
C***** fuel/air ratio f here, as the MASSFLOW subroutine
C***** requires it as input. When the actual f has been determined
C***** the massflows will be updated.

      P(2)=P(1)*lcr
      call COMPRESS(1,1,ncl,P,T,CP,R,Rair,EffType)

      T(3)=T(2)
      P(3)=(1-DiffL)*P(2)
      P(4)=(1-HEaL)*P(3)
      P(5)=(1-CombL)*P(4)

      f=0.02
5      fold=f
      call MASSFLOW(W,f,bleed)

C***** As bleed air is introduced in several points in the
C***** turbines, it is necessary to know both the pure air and
C***** combusted air percentages (AP,GP). These are used in CP and
C***** R calculations.

      GP=0.5*(W(5)/W(5)+W(5)/W(6))
      AP=1.0-GP
      call BALANCE(2,5,bleed,f,0.,W,T,GP,AP,Cper,Hper)
      P(6)=P(5)

      P(7)=(1+HEgL)*P(0)
      P(8)=P(0)
      call LPTURB(ntl,f,W,CP,R,Rair,P,T,ltr,Cper,Hper,EffType)

      call HEATEX(W,T,HEeff,f,Cper,Hper)

      call FUELFLOW(ncc,T,f,LCV,Cper,Hper)

      If (abs((f-fold)/f).GT.0.0005) Goto 5

```

```

    call PERFORM(W,T,P,Q,ComPOWER,CP,Power,nm,ng,SIPower,Fuel,f,
    $sfc,LCV,HR,SIHR,ntherm,bleed)

    If (OutType.EQ.2) Goto 6

    write (7,770) lcr
    write (7,771) T(5)
    write (7,772) SIPower*1000.*2.205/W(1)
    write (7,773) Fuel/(SIPower*1000*2.205)
    write (7,774) ntherm
    write (7,776) T(7)
    write (7,775) T(8)-273.15
    write (7,711)

    print *, 'Do you wish to: 1. Continue trial'
    print *, '                2. Finish trial'
    read (*,796) More
    print *, ' '
    If (More.EQ.1) Goto 4
    If (More.NE.1) Goto 8

6    write (7,711)
    write (7,701)
    write (7,702) Outfile
    write (7,703) RunTitle
    write (7,711)
    write (7,711)
    write (7,711)
    write (7,711)
    write (7,720)
    write (7,721)
    write (7,711)
    Do 12 i=0,8
    write (7,710) S(i),W(i),P(i),T(i),Q(i)
12  Continue
    write (7,711)
    write (7,711)
    write (7,711)
    write (7,711)
    write (7,725) Power,SIPower
    write (7,726) Fuel
    write (7,727) sfc
    write (7,728) lcr
    write (7,760) HR,SIHR
    write (7,761) ntherm
    write (7,711)
    write (7,711)
    write (7,711)
    write (7,711)
    write (7,730)
    write (7,762)
    write (7,711)
    write (7,731) lcr,ncl,CP(1),ComPOWER(1)
    write (7,735) ltr,ntl,CP(2),ComPOWER(2)

```

```

700  format(10X,F12.4)
701  format(10X,'SINGLE-SHAFT REGENERATIVE GAS TURBINE')
702  format(10X,'File Name: ',(A50))
703  format(10X,'Run Title: ',(A50))
710  format(10X,(A16),4(F12.2))
711  format(10X)
720  format(10X,' Station           W           P           T
$      WRT/P')
721  format(10X,'           (lbm/s)       (psia)       (K
$)')
725  format(10X,'Output Power=',F8.1,' Hp      ( ',F7.3,' MW )')
726  format(10X,'Fuel Flow=',F11.1,' lbm/hr')
727  format(10X,'sfc=',F17.4,' lbm/Hp hr')
728  format(10X,'OPR=',F17.3)
730  format(10X,'Component      PR      ETA      Cp      Power')
731  format(13X,'LPC',F14.3,F10.4,F9.4,F14.1)
735  format(13X,'HPT',F14.3,F10.4,F9.4,F14.1)
737  format(13X,'LPT',F14.3,F10.4,F9.4,F14.1)
760  format(10X,'Heat Rate=',F11.1,' BTU/Hp hr      ( ',F7.1,' BTU/KW hr
$)')
761  format(10X,'Efficiency=',F10.4)
762  format(42X,'(CHU/lbm K)      (Hp)')
770  format(10X,'PR=',F6.1)
771  format(10X,'TIT=',F6.0,'K')
772  format(10X,'SP=',F8.1,'kJ/kg')
773  format(10X,'SFC=',F6.3,'kg/kWh')
774  format(10X,'Eff=',F6.3)
775  format(10X,'EGT=',F5.0,'C')
776  format(10X,'HET=',F5.0,'K')
795  format(A30)
796  format(I1)

8      Close(Unit=7)

      print *, 'Do you wish to: 1. Run another trial'
      print *, '                        2. Exit from program'
      read (*,796) More
      print *, ' '
      If (More.EQ.1) Goto 2

      End

```

B3 Twin-Spool Gas Turbine Program Code

Program TWIN-SPOOL GAS TURBINE

```
implicit none
real ncc,nm,ng,ncl,nch,ntl,nth,lcr,hcr,ltr,htr,f
real T(0:8),P(0:8),W(0:8),Q(0:8),CP(4),R(4),opr
real fuel,sfc,LCV,ComPOWER(4),GP,AP,Cper,Hper,HR,ntherm
real Power,SIPower,SIHR,Rair,bleed,IntL,DiffL,CombL,ExhL
real GTW,TW,SIsfc,ccSIsfc,ccntherm
character*16 S(0:20)
character*50 Outfile,Infile,RunTitle
integer i,More,EffType,TempType,OutType
data nm,ncc,ng,Rair/0.995,0.999,0.98,0.06856/
```

```
S(0)='Ambient      '
S(1)='LPC Entry    '
S(2)='HPC Entry    '
S(3)='HPC Exit     '
S(4)='Combustion Entry'
S(5)='HPT Rotor Entry '
S(6)='HPT Exit     '
S(7)='LPT Rotor Entry '
S(8)='LPT Exit     '
```

```
2  Print *, ' '
   Print *, 'Enter name of output file: (eg. c:\trent\out.dat) '
   Read (*,795) Outfile
   Print *, ' '

   Open (unit=7,file=Outfile)

   print *, 'Do you wish to have: 1. Multi-concise output'
   print *, '                        2. Single-full output'
   read (*,796) OutType

   call FILEREAD(infile,RunTitle,EffType,ncl,nch,nth,
    $ntl,TempType,T,P,W,LCV,Cper,Hper,bleed,IntL,DiffL,CombL,ExhL)

4  Print *, 'Enter overall compressor ratio:'
   Read *,opr
   Print *, 'Enter TIT (K):'
   Read *,T(5)
```

```
C* ***** This step converts the supplied ambient temperature
C* ***** to Kelvin.
```

```

      If (TempType.EQ.1) then
        T(0)=T(0)
      Else if (TempType.EQ.2) then
        T(0)=T(0)+273.15
      Else if (TempType.EQ.3) then
        T(0)=(T(0)+459.67)*0.555556
      Endif

      T(1)=T(0)
      P(1)=(1-IntL)*P(0)
      lcr=SQRT(opr/1.5)
      hcr=opr/lcr

C***** It is necessary to assume an initial value for
C***** fuel/air ratio f here, as the MASSFLOW subroutine
C***** requires it as input. When the actual f has been determined
C***** the massflows will be updated.

      f=0.02
      call MASSFLOW(W,f,bleed)
      P(2)=P(1)*lcr
      call COMPRESS(1,1,ncl,P,T,CP,R,Rair,EffType)

      P(3)=P(2)*hcr
      call COMPRESS(2,2,nch,P,T,CP,R,Rair,EffType)

      T(4)=T(3)
      P(4)=(1-DiffL)*P(3)
      P(5)=(1-CombL)*P(4)
      call FUELFLOW(ncc,T,f,LCV,Cper,Hper)

C***** Now that the correct fuel/air ratio for the given conditions
C***** has been determined, the MASSFLOW subroutine is called
C***** again, to update those massflows after combustion occurs.

      call MASSFLOW(W,f,bleed)

C***** As bleed air is introduced in several points in the
C***** turbines, it is necessary to know both the pure air and
C***** combusted air percentages (AP,GP). These are used in CP and
C***** R calculations.

      GP=0.5*(W(5)/W(5)+W(5)/W(6))
      AP=1.0-GP
      call BALANCE(3,5,bleed,f,W,T,GP,AP,Cper,Hper)
      P(6)=P(5)

      call HPTURB(nm,nth,f,W,CP,R,Rair,P,T,htr,Cper,Hper,EffType)

      P(8)=(1+ExhL)*P(0)

```



```

call LPTURB(nt1,f,W,CP,R,Rair,P,T,ltr,Cper,Hper,EffType)

call PERFORM(W,T,P,Q,ComPOWER,CP,Power,nm,ng,SIPower,Fuel,f,
$sfsc,LCV,HR,SIHR,ntherm,GTW,TW,SIsfc,ccSIsfc,ccntherm,bleed)

If (OutType.EQ.2) Goto 6

write (7,770) opr
write (7,771) T(5)
write (7,772) GTW,TW
write (7,773) SIsfc,ccSIsfc
write (7,774) ntherm,ccntherm
write (7,775) T(8)-273.15
write (7,711)

print *, 'Do you wish to: 1. Continue trial'
print *, '                2. Finish trial'
read (*,796) More
print *, ' '
If (More.EQ.1) Goto 4
If (More.NE.1) Goto 8

6  write (7,711)
   write (7,701)
   write (7,702) Outfile
   write (7,703) RunTitle
   write (7,711)
   write (7,711)
   write (7,711)
   write (7,720)
   write (7,721)
   write (7,711)
   Do 12 i=0,8
   write (7,710) S(i),W(i),P(i),T(i),Q(i)
12  Continue
   write (7,711)
   write (7,711)
   write (7,711)
   write (7,711)
   write (7,725) Power,SIPower
   write (7,726) Fuel
   write (7,727) sfsc
   write (7,728) opr
   write (7,760) HR,SIHR
   write (7,761) ntherm
   write (7,711)
   write (7,711)
   write (7,711)
   write (7,711)
   write (7,730)
   write (7,762)

```

```

write (7,711)
write (7,731) lcr,ncl,CP(1),ComPOWER(1)
write (7,733) hcr,nch,CP(2),ComPOWER(2)
write (7,735) htr,nth,CP(3),ComPOWER(3)
write (7,737) ltr,ntl,CP(4),ComPOWER(4)

700  format(10X,F12.4)
701  format(10X,'TWIN-SPOOL GAS TURBINE')
702  format(10X,'File Name: ',(A50))
703  format(10X,'Run Title: ',(A50))
710  format(10X,(A16),4(F12.2))
711  format(10X)
720  format(10X,' Station                W                P                T
$      WRT/P')
721  format(10X,'                (lbm/s)        (psia)        (K
$)')
725  format(10X,'Output Power=',F8.1,' Hp      ( ',F6.3,' MW )')
726  format(10X,'Fuel Flow=',F11.1,' lbm/hr')
727  format(10X,'sfc=',F17.4,' lbm/Hp hr')
728  format(10X,'OPR=',F17.3)
730  format(10X,'Component      PR      ETA      Cp      Power')
731  format(13X,'LPC',F14.3,F10.4,F9.4,F14.1)
733  format(13X,'HPC',F14.3,F10.4,F9.4,F14.1)
735  format(13X,'HPT',F14.3,F10.4,F9.4,F14.1)
737  format(13X,'LPT',F14.3,F10.4,F9.4,F14.1)
760  format(10X,'Heat Rate=',F11.1,' BTU/Hp hr      ( ',F6.1,' BTU/KW hr
$)')
761  format(10X,'Efficiency=',F10.4)
762  format(42X,' (CHU/lbm K)      (Hp)')
770  format(10X,'PR=',F6.1)
771  format(10X,'TIT=',F6.0,'K')
772  format(10X,'SP=',F8.1,' kJ/kg',10X,F8.1,'kJ/kg')
773  format(10X,'SFC=',F6.3,' kg/kWh',10X,F6.3,'kg/kWh')
774  format(10X,'Eff=',F6.3,16X,F6.3)
775  format(10X,'EGT=',F5.0,'C')
795  format(A30)
796  format(I1)

8  Close(Unit=7)
   print *, 'Do you wish to: 1. Run another trial'
   print *, '                2. Exit from program'
   read (*,796) More
   print *, ' '
   If (More.EQ.1) Goto 2

End

```

B4 Reheat Gas Turbine Program Code

Program REHEAT GAS TURBINE

```
implicit none
real ncc,ng,nm,ncl,ntl,lcr,htr,ltr,f,f2,f3
real T(0:20),P(0:20),W(0:20),Q(0:20),CP(12),R(12),split
real fuel,sfc,LCV,ComPOWER(8),GP,AP,Cper,Hper,HR,ntherm
real Power,SIPower,SIHR,Rair,bleed,IntL,Diffl,CombL,ExhL
real SISfc,ccSISfc,ccntherm,GTW,TW,SR
character*16 S(0:20)
character*50 Outfile,Infile,RunTitle
integer i,More,EffType,TempType,OutType
data nm,ncc,ng,Rair/0.995,0.999,0.98,0.06856/
```

```
S(0)='Ambient'
S(1)='Comp Entry'
S(2)='Comp Exit'
S(3)='Comb1 Entry'
S(4)='Turb Entry'
S(5)='Stator 1 Exit'
S(6)='Rotor 1 Exit'
S(7)='Stator 2 Exit'
S(8)='Rotor 2 Exit'
S(9)='Turb Entry'
S(10)='Stator 3 Exit'
S(11)='Rotor 3 Exit'
S(12)='Stator 4 Exit'
S(13)='Rotor 4 Exit'
S(14)='Stator 5 Exit'
S(15)='Rotor 5 Exit'
S(16)='Stator 6 Exit'
S(17)='Rotor 6 Exit'
```

```
2 Print *,' '
Print *,'Enter name of output file: (eg. c:\trent\out.dat) '
Read (*,795) Outfile
Print *,' '

Open (unit=7,file=Outfile)

print *,'Do you wish to have: 1. Multi-concise output'
print *,'                        2. Single-full output'
read (*,796) OutType

call FILEREAD(infile,RunTitle,EffType,ncl,ntl,TempType,T,P,W,
$LCV,Cper,Hper,bleed,IntL,Diffl,CombL,ExhL)
```

```

4      Print *, 'Enter compressor ratio:'
      Read *, lcr
      Print *, 'Enter TIT (K):'
      Read *, T(4)
      Print *, 'Enter turbine split:'
      Read *, split

```

```

C***** This step converts the supplied ambient temperature
C***** to Kelvin.

```

```

      If (TempType.EQ.1) then
        T(0)=T(0)
      Else if (TempType.EQ.2) then
        T(0)=T(0)+273.15
      Else if (TempType.EQ.3) then
        T(0)=(T(0)+459.67)*0.555556
      Endif

```

```

      T(1)=T(0)
      P(1)=(1-IntL)*P(0)
      T(9)=T(4)

```

```

C***** It is necessary to assume an initial value for
C***** fuel/air ratio f here, as the MASSFLOW subroutine
C***** requires it as input. When the actual f has been determined
C***** the massflows will be updated.

```

```

      f=0.02
      f2=0.02
      call MASSFLOW(W, f, f2, bleed)
      P(2)=P(1)*lcr
      call COMPRESS(1, 1, ncl, P, T, CP, R, Rair, EffType)

      T(3)=T(2)
      P(3)=(1-DiffL)*P(2)
      P(4)=(1-CombL)*P(3)
      call FUELFLOW(ncc, T, f, LCV, Cper, Hper)

```

```

C***** Now that the correct fuel/air ratio for the given conditions
C***** has been determined, the MASSFLOW subroutine is called
C***** again, to update those massflows after combustion occurs.

```

```

      call MASSFLOW(W, f, f2, bleed)

```

```

C***** As bleed air is introduced in several points in the
C***** turbines, it is necessary to know both the pure air and
C***** combusted air percentages (AP, GP). These are used in CP and
C***** R calculations.

```

```

      P(17)=(1+ExhL)*P(0)

```

```

P(5)=P(4)
htr=((1-0.5*CombL)*P(5)/P(17))**split
SR=htr**0.5

GP=0.5*(W(4)/W(4)+W(4)/W(5))
AP=1.0-GP
call BALANCE(2,4,bleed,f,W,T,GP,AP,Cper,Hper)
call LPTURB(5,2,ntl,f,W,CP,R,Rair,P,T,SR,Cper,Hper,EffType)

GP=0.5*(W(4)/W(6)+W(4)/W(7))
AP=1.0-GP
call BALANCE(2,6,0.,f,W,T,GP,AP,Cper,Hper)
P(7)=P(6)
call LPTURB(7,3,ntl,f,W,CP,R,Rair,P,T,SR,Cper,Hper,EffType)

call FUELFLO2(ncc,T,f,f2,f3,LCV,Cper,Hper,W)

call MASSFLOW(W,f,f2,bleed)

P(9)=(1-0.5*CombL-DiffL)*P(8)
P(10)=P(9)
ltr=P(10)/P(17)
SR=ltr**0.25

GP=0.5*(W(9)/W(9)+W(9)/W(10))
AP=1.0-GP
call BALANCE(2,9,bleed,f3,W,T,GP,AP,Cper,Hper)
P(10)=P(9)
call LPTURB(10,4,ntl,f3,W,CP,R,Rair,P,T,SR,Cper,Hper,EffType)

GP=0.5*(W(9)/W(11)+W(9)/W(12))
AP=1.0-GP
call BALANCE(2,11,0.,f3,W,T,GP,AP,Cper,Hper)
P(12)=P(11)
call LPTURB(12,5,ntl,f3,W,CP,R,Rair,P,T,SR,Cper,Hper,EffType)

GP=0.5*(W(9)/W(13)+W(9)/W(14))
AP=1.0-GP
call BALANCE(2,13,0.,f3,W,T,GP,AP,Cper,Hper)
P(14)=P(13)
call LPTURB(14,6,ntl,f3,W,CP,R,Rair,P,T,SR,Cper,Hper,EffType)

GP=0.5*(W(9)/W(15)+W(9)/W(16))
AP=1.0-GP
call BALANCE(2,15,0.,f3,W,T,GP,AP,Cper,Hper)
P(16)=P(15)
call LPTURB(16,7,ntl,f3,W,CP,R,Rair,P,T,SR,Cper,Hper,EffType)

call PERFORM(W,T,P,Q,ComPOWER,Cf,Power,nm,ng,SIPower,Fuel,f,f2,
$sfsc,LCV,HR,SIHR,ntherm,GTW,TW,SI$fc,ccSI$fc,ccntherm)

If (CutType.EQ.2) Goto 6

```

```

write (7,769) split
write (7,770) lcr
write (7,771) T(4)
write (7,772) GTW,TW
write (7,773) SIsfc,ccSIsfc
write (7,774) ntherm,ccntherm
write (7,775) T(17)-273.15
write (6,800) (T(5)-T(8))/(T(5)-T(8))+T(10)-T(17))
write (7,711)

print *, 'Do you wish to: 1. Continue trial'
print *, '                        2. Finish trial'
read (*,796) More
print *, ' '
If (More.EQ.1) Goto 4
If (More.NE.1) Goto 8

6  write (7,711)
   write (7,701)
   write (7,702) Outfile
   write (7,703) RunTitle
   write (7,711)
   write (7,711)
   write (7,711)
   write (7,711)
   write (7,720)
   write (7,721)
   write (7,711)
   Do 12 i=0,17
   write (7,710) S(i),W(i),P(i),T(i),Q(i)
12  Continue
   write (7,711)
   write (7,711)
   write (7,711)
   write (7,711)
   write (7,725) Power,SIPOWER
   write (7,726) Fuel
   write (7,727) sfc
   write (7,728) lcr
   write (7,760) HR,SIHR
   write (7,761) ntherm
   write (7,711)
   write (7,711)
   write (7,711)
   write (7,711)
   write (7,730)
   write (7,762)
   write (7,711)
   write (7,731) lcr,ncl,CP(1),ComPOWER(1)
   write (7,735) htr,ntl,CP(2),ComPOWER(2)
   write (7,737) ltr,ntl,CP(3),ComPOWER(3)
   write (7,711)

```

```

700  format(10X,F12.4)
701  format(10X,'REHEAT GAS TURBINE')
702  format(10X,'File Name: ',(A50))
703  format(10X,'Run Title: ',(A50))
710  format(10X,(A16),4(F12.2))
711  format(10X)
720  format(10X,' Station           W           P           T
$      WRT/P')
721  format(10X,'           (lbm/s)       (psia)       (K
$)')
725  format(10X,'Output Power=',F8.1,' Hp      ( ',F7.3,' MW )')
726  format(10X,'Fuel Flow=',F11.1,' lbm/hr')
727  format(10X,'sfc=',F17.4,' lbm/Hp hr')
728  format(10X,'OPR=',F17.3)
730  format(10X,'Component      PR           ETA           Cp           Power')
731  format(13X,'LPC',F14.3,F10.4,F9.4,F14.1)
732  format(13X,'HPT',F14.3,F10.4,F9.4,F14.1)
737  format(13X,'LPT',F14.3,F10.4,F9.4,F14.1)
760  format(10X,'Heat Rate=',F11.1,' BTU/Hp hr      ( ',F7.1,' BTU/KW hr
$)')
761  format(10X,'Efficiency=',F10.4)
762  format(42X,'(CHU/lbm K)           (Hp)')
769  format(10X,'SPL=',F5.2)
770  format(10X,'PR=',F6.1)
771  format(10X,'TIT=',F6.0,'K')
772  format(10X,'SP=',F8.1,' kJ/kg',10X,F8.1,'kJ/kg')
773  format(10X,'SFC=',F6.3,' kg/kWh',10X,F6.3,'kg/kWh')
774  format(10X,'Eff=',F6.3,17X,F6.3)
775  format(10X,'EGT=',F5.0,'C')
795  format(A30)
796  format(I1)
800  format(10X,'DelT/T=',F6.3)

8      Close(Unit=7)

      print *, 'Do you wish to: 1. Run another trial'
      print *, '                        2. Exit from program'
      read (*,796) More
      print *, ' '
      If (More.EQ.1) Goto 2

End

```

B5 Program Functions and Subroutines

The various functions and subroutines used by the four programs are listed below in alphabetical order. Where a common subroutine, such as FILEREAD, is slightly different for each program, the version given is that for the single-shaft gas turbine program of B2.

```

subroutine BALANCE(s1,s2,bleed,f,f2,W,T,GP,At,Cper,Hper)
real bleed,f,f2,W(0:12),T(0:12),Cpal,Cpgl,CpaOld,CpgOld,GP,AP
real Cper,Hper
integer s1,s2

Cpal=CPA((T(s1)+T(s2))/2)
Cpgl=GP*(CPA(T(s2))+(f+f2)/(1+f+f2)*(Cper*ThCpC(T(s2))+Hper*ThCpH
$(T(s2))))+AP*CPA(T(s2))
22  CpaOld=Cpal
    CpgOld=Cpgl
    T(s2+1)=(bleed*W(s1)*Cpal*T(s1)+W(s2)*Cpgl*T(s2))/
$(bleed*W(s1)*Cpal+W(s2)*Cpgl)
    Cpal=Cpa((T(s2)+T(s2+1))/2)
    Cpgl=GP*(Cpa((T(s2)+T(s2+1))/2)+(f+f2)/(1+f+f2)*(Cper*ThCpC(T(s2)
$+T(s2+1))/2)+Hper*ThCpH((T(s2)+T(s2+1))/2))+AP*Cpa((T(s2)+
$T(s2+1))/2)
    If(.NOT.((abs((Cpal-CpaOld)/Cpal).LT.0.00005).AND.
$(abs((Cpgl-CpgOld)/Cpgl).LT.0.00005))) Goto 22
    Return
End

subroutine COMPRESS(j,i,nc,P,T,CP,R,Rair,EffType)
real nc,P(0:12),T(0:12),CP(8),CPold,R(8),Rair
integer i,j,EffType

R(j)=Rair
CP(j)=CPA(T(i))
20  CPold=CP(j)
    If (EffType.EQ.1) then
    T(i+1)=T(i)*(1+1/nc*((P(i+1)/P(i))**(R(j)/CP(j))-1))
    Else
    T(i+1)=T(i)*(P(i+1)/P(i))**(R(j)/(CP(j)*nc))
    Endif
    CP(j)=CPA((T(i)+T(i+1))/2)
    If(.NOT.(abs((CP(j)-CPold)/CP(j)).LT.0.00005)) Goto 20
    Return
End

```



```

real function CPA(T)
real T,C(0:13)
integer i
data(C(i),i=0,13)/0.21336328,-0.32921648E-04,0.47395140E-07,
$0.10126885E-09,-0.89883655E-13,0.0,0 19075549,0.12752498E-03,
$-0.54651988E-07,0.89378182E-11,0.0,0.0,-0.405671,0.11317630E+02/
If (T.LT.800) then
CPA=C(0)+C(1)*T+C(2)*T**2+C(3)*T**3+C(4)*T**4+C(5)*T**5
Else
CPA=C(6)+C(7)*T+C(8)*T**2+C(9)*T**3+C(10)*T**4+C(11)*T**5
Endif
Return
End

subroutine FILEREAD(infile,RunTitle,EffType,ncl,ntl,TempType,
$T,P,W,LCV,Cper,Hper,bleed,IntL,DiffL,CombL,ExhL)
real ncl,ntl,T(0:12),P(0:12),W(0:12)
real LCV,Cper,Hper,bleed,IntL,DiffL,CombL,ExhL
integer EffType,TempType
character*50 infile,RunTitle

Print *, 'Enter name of input file: (eg. c:\trent\in.dat) '
Read (*,801) Infile
Print *, ' '
open (unit=8,file=infile)
read (8,801) RunTitle
read (8,802) EffType
read (8,800) ncl
read (8,800) ntl
read (8,802) TempType
read (8,800) T(0)
read (8,800) P(0)
read (8,800) W(0)
read (8,800) LCV
read (8,800) Cper
read (8,800) Hper
read (8,800) bleed
read (8,800) IntL
read (8,800) DiffL
read (8,800) CombL
read (8,800) ExhL
800 format (F20.4)
801 format (A50)
802 format (I1)
close (unit=8)
Return
End

```

```

subroutine FUELFLOW(ncc,T,f,LCV,Cper,Hper)
real ncc,T(0:12),f,LCV,Cper,Hper,Cp1,Cp2,fold

f=0.02
Cp1=CPA((T(3)+298.)/2)
21 fold=f
Cp2=CPA((T(4)+298.)/2)+f/(1+f)*(Cper*THCPC((T(4)+298.)/2)+
$Hper*THCPH((T(4)+298.)/2))
f=(Cp1*(T(3)-298.)-Cp2*(T(4)-298.))/(ncc*(Cp2*(T(4)-298.)-LCV))
If (abs((f-fold)/f).GT.0.0005) Goto 21
Return
End

subroutine FUELFLO2(ncc,T,f,f2,f3,LCV,Cper,Hper,W)
real ncc,T(0:20),f,f2,LCV,Cper,Hper,Cp1,Cp2,f2old,W(0:20)
real GP,AP

f2=0.02
GP=W(4)/W(8)
AP=1-GP
Cp1=GP*(CPA((T(8)+298.)/2)+f/(1+f)*(Cper*THCPC((T(8)+298.)/2)+
$Hper*THCPH((T(8)+298.)/2)))+AP*(CPA((T(8)+298.)/2))
121 f2old=f2
Cp2=CPA((T(9)+298.)/2)+(f+f2)/(1+f+f2)*(Cper*THCPC((T(9)+298.)/2)+
$Hper*THCPH((T(9)+298.)/2))
f2=(Cp1*(T(8)-298.)-Cp2*(T(9)-298.))/(ncc*(Cp2*(T(9)-298.)-
$-LCV))
If (abs((f2-f2old)/f2).GT.0.00005) Goto 121
f3=(f*W(3)+f2*W(8))/(W(8)-f*W(3))
Return
End

subroutine HEATEX(W,T,HEeff,f,Cper,Hper)
real W(0:12),T(0:12),f,Cper,Hper,Cp34,Cp78,Cp78old

T(4)=T(3)+HEeff*(T(7)-T(3))
Cp34=CPA((T(3)+T(4))/2)
GP=W(5)/W(7)
AP=1.0-GP
Cp78=GP*(CPA(T(7))+f/(1+f)*(Cper*THCPC(T(7))+Hper*
$THCPH(T(7))))+AP*CPA(T(7))
22 Cp78old=Cp78
T(8)=T(7)-W(3)*Cp34*(T(4)-T(3))/(W(7)*Cp78)
Cp78=GP*(CPA((T(7)+T(8))/2)+f/(1+f)*(Cper*THCPC
$((T(7)+T(8))/2)+Hper*THCPH((T(7)+T(8))/2)))+
$+AP*CPA((T(7)+T(8))/2)
If (abs((Cp78-Cp78old)/Cp78).GT.0.0005) Goto 22
Return
End

```

```

subroutine HPTURB(nm,nt,f,W,CP,R,Rair,P,T,tr,Cper,Hper,EffType)
real nm,nt,f,W(0:8),CP(4),R(4),P(0:8),T(0:8),tr,CPold,GP,AP
real Cper,Hper,Rair
integer EffType

GP=W(5)/W(6)
AP=1.0-GP
R(3)=GP*Rair*(1+f/(1+f)*(-Cper+Hper*6.1848))+AP*Rair
CP(3)=GP*(CPA(T(6))+f/(1+f)*(Cper*THCPC(T(6))+Hper*
24 $THCPH(T(6))))+AP*CPA(T(6))
CPold=CP(3)
T(7)=T(6)-W(2)/W(6)*CP(2)/CP(3)*(T(3)-T(2))/nm
CP(3)=GP*(CPA((T(6)+T(7))/2)+f/(1+f)*(Cper*THCPC((T(6)+T(7))
$/2)+Hper*THCPH((T(6)+T(7))/2)))+AP*CPA((T(6)+T(7))/2)
If(.NOT.(abs((CP(3)-CPold)/CP(3)).LT.0.00005)) Goto 24

If (EffType.EQ.1) then
tr=(1-(T(6)-T(7))/(T(6)*nt))*(-CP(3)/R(3))
Else
tr=(T(6)/T(7))*((CP(3)/(R(3)*nt))
Endif
P(7)=P(6)/tr
Return
End

subroutine LPTURB(i,j,nt,f,W,CP,R,Rair,P,T,SR,Cper,Hper,EffType)
real nt,f,W(0:12),CP(8),R(8),P(0:12),T(0:12),SR,CPold,GP,AP
real Cper,Hper,Rair
integer EffType,i,j

P(i+1)=P(i)/SR
GP=W(4)/W(i)
AP=1.0-GP
R(j)=GP*Rair*(1+f/(1+f)*(-Cper+Hper*6.1848))+AP*Rair
CP(j)=GP*(CPA(T(i))+f/(1+f)*(Cper*THCPC(T(i))+Hper*
28 $THCPH(T(i))))+AP*CPA(T(i))
CPold=CP(j)
If (EffType.EQ.1) then
T(i+1)=T(i)-nt*T(i+1)*(1-(1/(P(i)/P(i+1))))*(R(i)/CP(j)))
Else
T(i+1)=T(i)*(P(i+1)/P(i))*((R(j)*nt)/CP(j))
Endif
CP(j)=GP*(CPA((T(i)+T(i+1))/2)+f/(1+f)*(Cper*THCPC
$((T(i)+T(i+1))/2)+Hper*THCPH((T(i)+T(i+1))/2)))
$+AP*CPA((T(i)+T(i+1))/2)
If(.NOT.(abs((CP(j)-CPold)/CP(j)).LT.0.00005)) Goto 28
Return
End

```

```
subroutine MASSFLOW(W,f,bleed)
real W(0:12),f,bleed
```

```
W(1)=W(0)
W(2)=W(1)
W(3)=(1-bleed)*W(2)
W(4)=(1+f)*W(3)
W(5)=W(4)+bleed*2*W(2)/3
W(6)=W(5)+bleed*W(2)/3
W(7)=W(6)
W(8)=W(7)
W(9)=W(8)
W(10)=W(9)
W(11)=W(10)
W(12)=W(11)
Return
End
```

```
Subroutine PERFORM(W,T,P,Q,ComPOWER,CP,Power,nm,ng,SIPower,Fuel,
$f,sfc,LCV,HR,SIHR,ntherm,GTW,TW,SIsfc,ccSIsfc,ccntherm)
real W(0:12),T(0:12),P(0:12),Q(0:12),ComPOWER(4),CP(4),Power,nm
real SIPower,Fuel,f,sfc,LCV,HR,SIHR,ntherm,ng,GTW,STW,TW,SILCV
real ccSIsfc,ccntherm,SIsfc,TEyh
integer i
```

```
ComPOWER(1)=W(1)*CP(1)*(T(2)-T(1))*1.8*778./550.
ComPOWER(2)=W(5)*CP(2)*(T(5)-T(6))*1.8*778./550.+
$      W(7)*CP(3)*(T(7)-T(8))*1.8*778./550.+
$      W(9)*CP(4)*(T(9)-T(10))*1.8*778./550.+
$      W(11)*CP(5)*(T(11)-T(12))*1.8*778./550.
```

```
Power=(ComPOWER(2)-ComPOWER(1)/nm)*nm*ng
SIPower=Power*0.0007457
```

```
GTW=1000.*SIPower/(W(1)/2.205)
If (T(12).LT.893.15) then
TEyh=T(12)
Else
TEyh=893.15
Endif
STW=(0.0013*TEyh**2-0.522*TEyh+64.418)*nm*ng
TW=GTW+W(12)/W(1)*STW
```

```
Fuel=3600.*f*W(3)
SIFuel=Fuel/2.205
sfc=Fuel/Power
SIsfc=SIFuel/(SIPower*1000.)
ccSIsfc=SIFuel/(TW*W(1)/2.205)
SILCV=LCV*1.8*2.326
HR=sfc*LCV*1.8
SIHR=HR/0.7457
ntherm=3600./(HR*778./550.)
ccntherm=3600./(ccSIsfc*SILCV)
```

```

10      Do 10 i=0,12
         Q(i)=W(i)*sqrt(T(i))/P(i)
         Continue
         Return
         End

```

```

      real function THCPC(T)
      real T,CPC(0:4)
      integer i
      data(CPC(i),i=0,4)/-0.39277299,0.14186817E-02,
$-0.1533902E-05,0.76790508E-09,-0.14392196E-12/
      THCPC=CPC(0)+CPC(1)*T+CPC(2)*T**2+CPC(3)*T**3+CPC(4)*T**4
      Return
      End

```

```

      real function THCPH(T)
      real T,CPH(0:4)
      integer i
      data(CPH(i),i=0,4)/0.21027252E+01,-0.93246352E-03,
$0.21762758E-05,-0.90091221E-09,0.11172113E-12/
      THCPH=CPH(0)+CPH(1)*T+CPH(2)*T**2+CPH(3)*T**3+CPH(4)*T**4
      Return
      End

```

END

3 1 - 0 5 - 9 5

FIN

Figure 5.6: Aero-Derivative Simple Cycle Mesh