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**CYCLE ANALYSIS OF INTERCOOLED AND REGENERATIVE
NAVAL GAS TURBINE**

**by
Yu Hou**

B.Sc., Beijing Institute of Aeronautics and Astronautics., 1985

**A Thesis Submitted to the Faculty of Graduate
Studies and Research in Partial Fulfilment
of the Requirements for the Degree of**

Master of Engineering

**Ottawa-Carleton Institute of Mechanical and
Aerospace Engineering**

**Department of Mechanical and Aerospace Engineering
Carleton University
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ABSTRACT

The problem of fuel consumption at part-loads and the desirability of developing a new generation of naval power system, which will result in a sensible reduction in overall machinery plus fuel weight, have brought the free world's navies back to the "old" concept — the intercooled regenerative (ICR) cycle gas turbines. This thesis reviews some early trials of the marine ICR gas turbines and describes a theoretical investigation of the basic thermodynamic behaviour of the ICR gas turbines. The studies focus on two issues, the design-point (DP) cycle performance and the off-design characteristic of an ICR gas turbine.

The design-point performance is studied by applying the basic principles of gas turbines. The calculations reveal how the regenerator and the intercooler are involved to promote the engine performance. WR-21 is a new ICR gas turbine being co-developed by Westinghouse and Roll-Royce. The DP analysis has verified the known performance of the engine.

The off-design analysis based on the WR-21 uses generalized compressor and turbine characteristics to account for the changes of component performance at part-loads. A simplified solution method using the compatibility of flow and work, and the matching of turbines in series, results in the unique characteristics of the controlling parameters of an ICR gas turbine. To avoid excessive graphic iteration, some component performance characteristics are adapted into a data base.

The effect of variable power turbine nozzle, which is featured in the WR-21, is tentatively estimated at the end of the thesis.

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Dedicated To

**My Parents, Mansu, Yunan
and DuoDuo**

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NOMENCLATURE

T	total temperature
P	total pressure
f	fuel/air ratio by weight
h	specific enthalpy
m	mass flow
η	isentropic efficiency
ϵ	thermal effectiveness
γ	ratio of specific heats
n	polytropic index
s	specific entropy
W	specific work
C_p	specific heat at constant pressure
ΔT	change of absolute temperature
N	shaft speed
Q^*	non-dimensional flow ($m\sqrt{T/P}$)
E^*	efficiency

[Suffix (*) indicates the parameters appear in graphics]

SUBSCRIPTS

a	air
g	combustion products
$1,2,3,\text{etc.}$	reference planes
∞	polytropic
cc	combustion chamber
ic	intercooler

m	mechanical
HE	regenerator
c	compressor
t	turbine
L	low pressure rotor
H	high pressure rotor
b	cooling bleed for turbine
r	cooling bleed for turbine rotor bleed
n	cooling bleed for turbine nozzle bleed

ABBREVIATIONS

LP	low pressure
HP	high pressure
LPC	low pressure compressor
HPC	high pressure compressor
LPT	low pressure turbine
HPT	high pressure turbine
PT	power turbine
TIT	turbine inlet temperature
ITT	temperature at the HPT outlet or the LPT inlet
PTIT	power turbine inlet temperature
OPR	overall pressure ratio
LPCR	pressure ratio of low pressure compressor
HPCR	pressure ratio of high pressure compressor
LPTR	pressure ratio of low pressure turbine
HPTR	pressure ratio of high pressure turbine

ISO	international standard organization
ICR	intercooled and regenerative
ECV	effective calorific value
DP	design point
SFC	specific fuel consumption

Chapter One

Introduction

1.0 BACKGROUND

Those connected with naval propulsion were long ago aware that the light-weight and simple-structure gas turbine engine could be an ideal power plant to power warships. The Royal Navy pioneered in this field by putting a gas turbine engine on board a Motor Gun Boat in 1947, only a few years after the advent of the first gas turbine powered airplanes. Since then this type of prime mover has gradually given it great appeal to the various navies and has been chosen for boost, main propulsion, and auxiliary service in full competition with possible alternative power plant , and numerous vessels and classes of vessels are so powered. However, despite the basic advantages of the gas turbine, e.g.

- (1) Inherent simplicity;
- (2) Rapid starting;
- (3) Ease of automation;
- (4) Small volume and weight for the power developed;
- (5) High frequency noise can be relatively easily controlled;

the gas turbine's inherent disadvantage of poor fuel efficiency, especially at low power rating, made its user suffer big penalties on range or useful payload, if no other approach were used to either improve its fuel consumption or avoid part-load operation. The pressure is always on for engine manufacturers to improve the fuel efficiency of their products.

The intercooled and regenerative (ICR) gas turbine was among a few early approaches to improve fuel efficiency that were investigated and tested in the late 1940's and 1950's. Two examples are the Rolls-Royce RM 60, which was the first ICR cycle gas turbine tested by the Royal Navy in 1951, and at the about same time the British Thomson Houston (BTH) ICR gas turbine installed on board the Shell tanker "Auris" on a trial basis. Further details on both ships

will be mentioned in the next chapter. These early installations are older engines and the recuperator technology is out-of-date, and the improvement in fuel efficiency was not good enough to offset the additional weight and complexity in both operation and structure. Furthermore, the initial cost appeared too high for the commercial user, so the marine ICR gas turbine dropped from the scene for over thirty years. In the early 1980's, the U.S. Navy, prompted by availability of the advanced gas turbine and heat exchanger technology, restarted studies on ICR gas turbines. A group of companies, including Westinghouse, Allison, and AiResearch from US and Rolls-Royce from UK, joined the study. In 1992, the US Navy selected the Westinghouse-RR team to develop affordable, commercially based ICR gas turbine systems for future US Navy surface combatants and combat logistics ships. The new ICR gas turbine engine is named WR-21, which is an adaptation of the Roll-Royce RB211 aero engine and U.S. recuperator technology. It shows that the idea of using the ICR cycle gas turbine for main propulsion is regaining appeal to navies. Given the relative newness of the combination of twin-spool aero derivative engine, intercooler, regenerator and free power turbine with variable nozzle, so far very little detailed information and research papers on the ICR cycle have been published. To understand why the navies are making such a move to the ICR again, what benefit they can achieve with it, and how the engine cycle works, this thesis study on ICR gas turbines was initiated.

1.1 THESIS OBJECTIVES

The expressed purpose of this study is to provide an independent appraisal and study of intercooled and regenerative (ICR) marine gas turbines for naval application. The Westinghouse /Rolls-Royce WR-21 ICR gas turbine has been chosen as the model for the ICR cycle design point and off-design analysis. Specific objectives within this general study context were:

- (1) To gain an understanding of naval needs and problems related to the use of marine gas turbines as prime mover for warships, such as power, weight and size requirements, consumption of fuel, reliability, noise level, etc.
- (2) To review the history of marine use of ICR gas turbines, and illustrate the reasons why this type of power plant has regained appeal to the U.S. navy and the navies

of other nations.

- (3) To carry out a theoretical cycle analysis of the ICR engine to show how the intercooler and regenerator are incorporated to improve the engine performance in terms of fuel consumption and power.
- (4) To acquire reliable data for the Westinghouse/RR WR-21 engine, and do an accurate design point performance estimation.
- (5) To use computer tools and generalized compressor/turbine characteristic graphs to estimate the off-design characteristics of the WR-21 engine.
- (6) To examine the effect of using the variable power turbine guide vanes to improve the specific consumption at low-power.
- (7) To examine control parameters for this new cycle.
- (8) To extract from these analyses a complete performance picture of the WR-21 ICR Marine Gas Turbine.

1.2 LITERATURE REVIEW

The origination of the concept of intercooled and regenerative (ICR) marine gas turbines goes back to the 1940s. Two trials, the Rolls-Royce RM60 and a BTH gas turbine --- the only ICRs ever produced --- were carried out in the 1950s. Although some general information on the RM60 and the BTH ICR engine were introduced by Trewby(1955) and Lamb & Duggan(1962), respectively, more detailed information on engine cycle analysis and design is not available. Later in the 1980s, some studies were carried out on developing new industrial based or marine ICR power plants, these new proposals included the TF-10 of Thomassen Holland/Fern Engineering, the DC990 of Dresser-Rand and the Marine Spey ICR of Rolls-Royce. Although none of these proposals ended up with a real product, these studies did pave the road for the future development of ICR gas turbines.

How can the intercooling and regeneration arrangement improve the cycle performance? The statement appearing in the DC990 ICR proposal (Staudt,et al, 1989) can perfectly answer the question --- "For most simple-cycle gas turbines, the enthalpy of the turbine exhaust exceeds that of the combustor inlet. If a heat exchanger is used to recover some of the energy in the turbine

exhaust and add this energy to the air stream between the compressor discharge and the combustor inlet, the amount fuel needed in the combustor to raise the air temperature to full turbine inlet temperature can be reduced....If intercooling is used on a gas turbine engine to reduce the work of compression, the net work output of the engine can be greatly increased.... There is a good synergism that occurs when intercooling and regeneration are combined."

A complete study on the ICR cycle gas turbine was carried out by Hendriks and Levine(1980) in connection with the design of the proposed TF-10 ICR cycle industrial gas turbine. The new 10,000 shp power system featured a two stage intercooled centrifugal compressor, a regenerator and an annular combustor with a single spool gas generator and a free power turbine arrangement. The cycle optimization concluded that, for the ICR cycle, the selection of overall pressure ratio(OPR), turbine inlet temperature(TIT) and regenerator inlet temperature were closely inter-related. The studies showed a higher TIT would always produce a better cycle efficiency but it was often fixed by the creep stress limit of the turbine blade; the selection of OPR based on engine efficiency should be kept in the range of 8-12, a further higher OPR would result in a drop in cycle efficiency because a higher OPR would produce a higher compressor delivery temperature and a lower regenerator inlet temperature, thus reducing the benefit of using the regenerator. On the other hand, the selection of a lower OPR would mean a higher regenerator inlet temperature, which in turn requires a more expensive stainless steel heat exchanger. Hendriks and Levine have also examined the off-design performance of the ICR cycle, showing that, for the single spool gas generator plus free turbine ICR layout, the part-load operation reflected a big drop in TIT, for example, a 50% power rating of the TF-10 was obtained at a TIT of 1600 °F, about 400 degrees lower than the rated TIT of 2042 °F. The proposed TF-10 was predicted to be capable of achieving a 20% efficiency improvement over the simple cycle gas turbine. It was widely agreed that the use of centrifugal compressors could offer an easy arrangement of flow path. The 8400 bhp DC990 ICR also featured a two stage centrifugal compressor (Staudt, et al, 1989). However, for large simple cycle gas turbines with a power rating of 10000 bhp and above, axial compressors show advantages over centrifugal compressors because of higher component efficiency (Saravanamuttoo,1987). Both the proposed ICR Spey(Thomas,1985) and the current program WR-21 (Westinghouse/RR) are adapted from the large modern turbofan engines, which featured the use of axial compressors.

Woodward (1972) introduced a general overview of marine gas turbines, which has clarified the duty and requirements for naval propulsion in the aspects of power output, fuel burn, weight and volume, reliability, maintenance and engine control. Spyrou (1988) addressed the importance of the fuel efficiency of the marine power plant. The major disadvantage of marine gas turbines is the poor fuel consumption at low-power cruising, thus the complex machinery arrangements must be used (Saravanamuttoo, 1978). The recuperator technology used in the early installations, the RM60 and the BTH ICR, was unsatisfactory, the compact sized heat exchanger used in RM60 could only offer an effectiveness of 48 % (Trewby, 1955). Modern regenerator technology advancements, however, can produce a compact size plate-fin type heat exchanger with 90 % effectiveness and 4 % pressure drop (Parker, 1977). The modelling of cycle off-design performance requires the compressor and turbine performance maps, however, these product performance maps are not often released by the manufacturers. Saravanamuttoo and MacIsaac (1983) suggested the concept of using generalized non-dimensional compressor and turbine performance diagrams, which were obtained from typical component tests, to predict the off-design performance of other engines. The generalized compressor and turbine characteristics used by Saravanamuttoo and MacIsaac were found to be very reliable in estimating the off-design performance of gas turbines with compressor pressure ratio less than 6. The thermodynamic theory used in cycle modelling is based on the method of matching turbines in series and matching the mass and work relations of the gas turbine rotors (Cohen, Rogers and Saravanamuttoo, 1987).

Barry (1976) introduced a method of accounting for the aerodynamic penalties of air-cooled turbine blades. Based on design and tests results of a number of Roll-Royce gas turbines, including the Avon, Spey and RB211, Barry used convective heat transfer theory to determine cooling flow requirement, and used enthalpy analysis to calculate the stage efficiency of a cooled turbine stage.

Woodward (1972) introduced the use of variable power turbine nozzle on General Electric MS-5002B industrial gas turbine. A control system, called "Speedtronic", was used to hold the engine speed at 80% of rated value over the 0~50% load range by nozzle changes.

Chapter Two

Marine Gas Turbines

and ICR Cycle

2.0 REQUIREMENTS FOR NAVAL POWER PLANTS

It is instructive to discuss the special requirements facing a naval propulsion engine before doing any detailed studies on the "marinized" gas turbines, or the more complex ICR systems, since an understanding of these requirements is essential in deciding on the suitability of a power plant, and in making a decision of developing a new type of engine, such as ICR gas turbines.

It is generally agreed what constitutes the basic requirements for marine propulsion engines, but the same agreement does not always appear in listing these in order of relative importance. For naval propulsion, a logical order for the list seems to be:

- (a) Minimum fuel consumption
- (b) Maximum reliability
- (c) Minimum initial and operating/maintenance cost
- (d) Minimum weight and volume
- (e) Maximum maneuverability

A more detailed analysis of these requirements can be done in three categories: those demanded by (1) the load, (2) the fuel consumption, and (3) the service.

(1) Load Requirements

In marine propulsion, the thrust, with few exceptions, is always produced by a screw propeller, which is driven by a propulsion engine through a gearing system. The gearbox carries out the function of matching the prime mover's output with the torque and RPM requirement of the propulsor. Most marine prime movers, including the diesel engine, steam turbine and two-

spool gas turbines, have very good torque characteristics during cruising. Also, it is required to match the load during accelerating and decelerating, starting and stoping. A related point is the time taken to bring the engine from a cold condition to the condition at which it can produce useful power. It is worth mentioning that although several hours of warm-up time is sometime acceptable for a marine engine, often advantage can be gained from the ability to start in a short interval of time. In maneuvering situations, the propulsion load changes rapidly. The gas turbine is outstanding in meeting these requirements, since the inertia of the moving parts in gas turbines is small, it takes much less time to bring the whole engine to full speed and to warm up the installation.

(2) Fuel Consumption Requirement

The endurance of a warship is usually designated by the distance the vessel can travel on one loading of fuel at a specified fraction of rated propulsive power. It is most directly related to the weight or volume that can be devoted to the carriage of fuel. Figure 2.1 illustrates the improvement of thermal efficiency of medium speed diesel and gas turbines, these two types of engine are the major competitor for naval propulsion. The diesel usually enjoys about 10-15 % better fuel efficiency than the best simple cycle gas turbine at full power, and so regains at least part of the latter's weight advantage when fuel weight is considered.

Fig.2.1 Improvement of Thermal Efficiency of Marine Diesel and Gas Turbine Over Years[From Ref.17]

Also, most naval vessels operate at the cruise condition for most of the time, then the power requirement might be only 15 % of the rated power. This implies that good low-power fuel characteristics is very important to obtain a long endurance. Figure 2.2 illustrates a typical operating profile of warships. Simple gas turbines have a major disadvantage because of their poor part-load fuel consumption. One comparison indicates the typical part-power characteristic of marine diesel, simple and ICR gas turbine, as shown in Figure 2.3. The overall power plant plus fuel weight is one of the major criteria in selecting an engine for naval propulsion.

Fig.2.2 Typical Operating Profile of Warships [From Ref.15]

Fig.2.3 Typical Part-load SFC Characteristics of Various Power Plants
[From Ref. 12 and 21]

(3) Service Requirements

The service requirements for naval propulsion include:

a) Weight/Volume Requirement

This requirement has been the strongest point in favour of the gas turbine, especially for aero-derivative engines. For example, a typical naval power plant rated at 20,000 hp might weigh 100 tons if based on aircraft gas turbine, about 200 tons if the intercooling and regeneration (ICR) is added, and about 700 tons if using medium speed diesel. The volume comparison runs in the same order, although the gas turbine advantages are less striking because of the large volumes occupied by inlet and outlet ducting, and by heat regenerator when that device is present.[based on Chapter 1.3 of Marine Gas Turbines,J.B.Woodward (1972)]

b) Noise

In the naval application, lower under-water noise is always required to avoid the sonar detection from enemy ships. The significant feature of diesel noise is its inherent low frequency noise which is not a steady level with the pulsing nature. The problem with the

low frequency sound is one of sympathetic vibration which is reproduced in other items that are subjected to the vibration. The feature of gas turbine noise is the basically high frequency steady sound. It is found in practice that a pulsing type of noise is far more obtrusive than a steady one.

c) Reliability/Maintenance

Reliability of a power plant is the probability that it will accomplish its assigned mission. Maintenance is the process of examinations and replacements that is required to keep the propulsion machinery operating in a satisfactory manner. Marine gas turbine designers have devoted much effort to improve reliability of gas turbines, for example, by using rust-resistant coating and material and lowering the working temperature of the engine hot section. At present, the working life of marine gas turbines before major overhaul has exceeded 10,000 hours. The marine diesel offers much better reliability, but has the problem of trouble shooting. Another significant advantage of the aero-derivative gas turbines is the capability of removing the gas generators via the intake duct, permitting a policy of replacement rather than on-site repair.

2.1 CLASSIFICATION/ILLUSTRATION OF MARINE GAS TURBINES

The marine gas turbine, like most other engines used for naval propulsion, is usually an adaptation of a machine that was originally developed for some other purpose. The aircraft-derivative gas turbine is a prominent example. This is simply because a huge amount of research and development funding has already been put into the development of aircraft gas turbines, and it is found more economical to use a "marinised" aircraft gas turbine for marine application than to design and develop a totally new engine.

The application of marine gas turbines falls into three categories — boost propulsion, main propulsion and auxiliary power plant. As mentioned before, the marine gas turbines have a major disadvantage of poor fuel efficiency, especially at low-power operating condition, whilst a typical naval vessel spends most of its operating time cruising at low power rating. This fact makes it uneconomical and unacceptable to use a single gas turbine to meet the whole power requirements of the vessel. To get over this problem, splitting power, i.e., combined power plant consisting of

gas turbines in conjunction with steam turbines, diesels and other gas turbines, was considered and used widely by navies around the world. These installations go by names such as COSAG, CODOG, COGAG, etc. CO stands for 'combined'; S,D,G for 'steam', 'diesel' and 'gas turbine'; the final A or O for 'and', 'or'. Many modern mainstream warships are currently powered by CODOG, COGOG or COGAG installations.

Some typical gas turbine arrangements, which have been adapted in marine application, are sketched schematically in Figure 2.4 — One may speak of a simple gas turbine (Figure 2.4a), of a regenerative gas turbine (Figure 2.4b), and of an intercooled and regenerative cycle (ICR) gas turbine (Figure 2.4c).

For simple cycle gas turbines, a good example is the General Electric LM2500 marine gas turbine engine, which has been used or co-used with diesel on up to 11 different naval warships, including DD-963 Destroyer, CG-47 Cruiser and FFG-7 Frigate. LM2500 is an adaptation of the TF39 military and CF-6 commercial turbofan aircraft engines, and has a nominal rating of 25,000 horsepower. The cutaway picture of the engine is shown in Figure 2.5.

In recent years, driven by the naval requirement of minimised overall engine plus fuel weight, there is a tendency that the ICR cycle gas turbine and the arrangement of COGAG are regaining its popularity. The U.S. Navy has selected the Westinghouse-RR WR-21 ICR as a potential power plant for future naval warships, or for retrofitting existing ships.

Fig.2.4 Typical Marine Gas Turbine Cycle Arrangements
[From Marine Engineering, 1971]

Fig.2.5 Cutaway View of LM2500 Marine Gas Turbine
[From Ref. (14)]

2.2 HISTORY OF MARINE ICR GAS TURBINES

It was soon after World War II when some people in the shipping industry felt that the heavy, complicated diesel engine would sooner or later have to make way for a power installation of greater simplicity and of less weight for a given power developed. Also, it appeared at that time, and nothing had happened since to materially change the view, that whilst its weight per unit power developed might be reduced, the diesel engine was thought, as regards thermal efficiency, to have reached its limit. On the other hand, in the case of steamships, part of the propelling plant was used for handling other power requirements on board, but not in an economical manner. The whole of the steam plant might have to remain idle for a great proportion of the ship's life. So apart from the huge size and heavy weight, the steamship had another problem of power splitting. It appeared then that the simply structured gas turbine could be an ideal alternative to replace the diesel and steam plant as marine power plant.

Some early research and trials started in 1940's. The Royal Navy (R.N.) pioneered in this field by using a Metrovick simple gas turbine in the MGB2009 gun boat for boost propulsion in 1947. The first gas turbine, named "Gatric", was proved successful in the 4 year service which followed. Two years after the introduction of "Gatric", another Metrovick boost gas turbine, "G2", was tested by the R.N. It should be mentioned that these early gas turbines, still very much in the infancy of its design, were basically for trial purposes and operating at very uncompetitive conditions, when comparing with marine diesels at that time. [Trewby(1955)]

Meanwhile, the development in the second category — main propulsion gas turbine for naval and commercial vessels — was carried out by some other companies, including Rolls-Royce and British Thomson-Houston. Unlike the naval boost engine, which is basically operated at rated condition to bring the ship to full speed, the main propulsion gas turbine must be capable of meeting the naval and commercial requirement of economical low-power cruising. It was found then that the simple cycle gas turbines can no longer meet this requirement, complex cycles, for example, regenerative cycle or intercooled and regenerative(ICR) cycle, must be used to recover some exhaust heat or at the same time lower the compression work of the compressor in order to improve the cycle efficiency. Rolls-Royce started to design the first ICR cycle marine gas

turbine for R.N. coastal craft in 1946. This ICR installation was named RM60, and the shore trial of the complete engine began in July 1951. The diagrammatic layout of the RM60 was given by Trewby(1955). Since the quality of this old picture is not good, a schematic sketch of RM60 is illustrated in Figure 2.6. The engine performance is listed in Table 2.1.

Fig.2.6 Schematic Layout of the Rolls-Royce RM60 [From Ref. 2]

Table 2.1 RM60 Characteristics

Maximum Power at 60 °F (bhp)	5400
Duty	Main Propulsion
Turbine Inlet Temperature (°F)	1521
Overall Pressure Ratio	18.5
Air Flow at Max Power (lb/s)	64.6
Thermal Efficiency at Max Power	20.4
Specific Weight (lb/hp)	5.3
Designed Life of Hot Parts (hr)	1000
Low-pressure Compressor	11 stage axial
High-pressure Compressor	2 stage centrifugal
High-pressure Turbine	1 stage axial
Low-pressure Turbine	2 stage axial
Intercooler Effectiveness	2 lp in parallel:86
(%)	1 hp: 64
Regenerator Effectiveness (%)	tubular cross-flow :48
Combustion Chamber	2 cans

The Rolls-Royce RM60 was installed in the R.N. gunboat Grey Goose in 1953-54. The sea trials was commenced shortly after and continued for four years. The sea trials showed the new ICR RM60 had a much better specific fuel consumption than other simple cycle installations used by R.N. at that time. The SFC comparison of these early engines is illustrated in Figure 2.7. However, the trial had also experienced some problems associated with the new power system. The engine control seemed to be too complex, the engine maneuvering was much slower than that of simple gas turbines, the noise level was reported too high, and severe erosion of the HP compressor and regenerator was also experienced. Criticism also went to the complexity of the concentric-shaft arrangement. At the end, the R.N. dropped the idea of using an ICR gas turbine for main propulsion and, instead, adopted an installation comprising a number of simple gas turbines, each of which is designed to operate over its peak

efficiency range. The latter was actually the "combined" concept.

Fig.2.7 Fuel Consumption of the Early Gas Turbine Used in Royal Navy
[From Ref.2]

Another example, which must be mentioned in the ICR history, is the BTH ICR gas turbine in the Shell Tanker "Auris". The early history of the "Auris" is well known and she started with four diesel engines, and later in 1951 one of the four original diesels was replaced by a BTH regenerative cycle gas turbine. The success of the first gas turbine is unquestioned. It saw four years of sea duty and amassed over 20510 hr operating time without an involuntary stop between ports. Encouraged by the success of the first gas turbine, it was decided to installed a 5500 hp ICR gas turbine, replacing all of the four old installations. Basin trials on the new ICR engine took place in July, 1958 and the ship was in service a year later until it was withdrawn and laid up in July, 1960. This was not because the engine was defective but because in the prevailing circumstances of the time, a 12000-ton tanker was too small and had become

uneconomical. The layout of this ICR gas turbine is shown in Figure 2.8 and the engine performance is tabulated in Table 2.2. The fate of the second ICR gas turbine was not as good as the first regenerative system. Production of the main engine was unfortunately retarded by manufacturing difficulties and labour disputes, no shop test was carried out and finally sea service was terminated by the shifting economical environment. The criticisms of the ICR engine were levelled at the complexity of the system and high initial cost.[Duggan & Howell (1962)]

Table 2.2 Specifications of BTH ICR Gas Turbine

Power output (bhp)	5500
Ambient Air Temperature (°F)	75
Overall Pressure Ratio	6.1
Regenerator Effectiveness (%)	65
Turbine Inlet Temperature (°F)	1200
High Pressure Turbine speed (rpm)	5800
Low Pressure Turbine speed (rpm)	3840
Specific fuel Consumption at Max Power(lb/hphr)	0.505
SFC at Cruising	0.667
Overall Thermal Efficiency (%)	28.0
Specific Weight (lb/hp)	68.3

Fig.2.8 Configuration of the British Thomson Houston Gas Turbine Used in Ship "Auris"[From Ref.(4)]

(1) Air Inlet Pipe	(11) High-Pressure Turbine
(2) Horizontal Duct	(12) Low-Pressure Turbine
(3) Low-Pressure Compressor	(13) Short Exhaust Duct
(4) Intercooler	(14) Exhaust Gas Boiler
(5) High-Pressure Compressor	(15) Exhaust
(6) Pipe	(16) Alternator
(7) Inlet Drum	(17) Speed Reducing Gear
(8) Heat Regenerator	(18) Auxiliary Alternator
(9) Outlet Drum	(19) Electric Starter
(10) Combustion Chamber	(20) Gear System

Chapter Three

Cycle Analysis for ICR Gas Turbines

3.0 INTRODUCTION

For the ideal gas turbine cycle, either simple or complex, an expression for thermal efficiency can be deduced. Take the simple (Joule, or Brayton) cycle for example, its schematic cycle sketch and temperature-entropy diagram are shown in Figures 3.1. The work done in compressor (W_{12}) and in turbine (W_{34}), the heat transfer in combustor (Q_{23}) can be obtained using

$$W_{12} = -(h_2 - h_1) = -c_p(T_2 - T_1)$$

$$Q_{23} = (h_3 - h_2) = c_p(T_3 - T_2)$$

$$W_{34} = (h_3 - h_4) = c_p(T_3 - T_4)$$

Fig.3.1 Simple (Joule or Brayton) Cycle and its T-s Diagram

The cycle efficiency η of the simple cycle can be defined as

$$\eta = \frac{\text{net work output}}{\text{heat supplied}} = \frac{c_p(T_3 - T_4) - c_p(T_2 - T_1)}{c_p(T_3 - T_2)}$$

Making use of the isentropic relation, $T_2/T_1 = r^{(\gamma-1)/\gamma} = T_3/T_4$, where r is the compressor pressure ratio, the cycle efficiency is then shown to be given by

$$\eta = 1 - \left(\frac{1}{r}\right)^{(\gamma-1)/\gamma}$$

where γ is the ratio of specific heats of working fluid. Thus, the elementary cycle analysis leads to the major conclusion that overall efficiency is governed by pressure ratio; this holds for real cycles as well, but when the losses in an ideal cycle are accounted for it is found that the efficiency becomes dependent upon the maximum cycle temperature as well as pressure ratio.

For the ideal intercooled and regenerative (ICR) cycle, the cycle efficiency depends not only on the compressor pressure ratio, but also on the maximum cycle temperature, because temperature difference between the outlets of compressor and turbine determines the amount of energy recovered by the regenerator. It is true to say that no very high pressure ratio is required for an ICR cycle, because a high pressure ratio would result in a high compressor delivery temperature, which would reduce the function of the regenerator. In the real ICR gas turbine, in addition to all kinds of component losses, e.g. frictional pressure losses and "incomplete" heat exchange, other factors such as the location of intercooler also affect the cycle efficiency. In this chapter, the general characteristics of the ICR cycle will be studied using the basic thermodynamic relations.

3.1 ICR GAS TURBINE CYCLE

As mentioned earlier, most marine gas turbines are adapted from aero gas turbines. Modern ICR gas turbine designs are often based on twin-spool aero gas turbines, with an intercooler fitted between the low-pressure compressor (LPC) and high-pressure compressor (HPC), and a hot-end-positioned heat regenerator heating up the HPC delivery air with the exhaust gas. The schematic

diagram of such a two-spool ICR gas turbine and the station numbering system is illustrated in Figure 3.2. The corresponding process in each component involved is shown in Figure 3.3.

Fig.3.2 Schematic Layout of a Typical Twin-Spool ICR Gas Turbine and its Numbering System

Fig.3.3 T-s Diagram of a Typical ICR Gas Turbine

Since different ICR designs originate from different aero engines, to examine the general

characteristic of ICR cycle, it is necessary to carry out the cycle calculations based on varying governing parameters. The basic assumptions for the cycle calculation are shown in Table 3.1.

Table 3.1 Basic Assumptions For ICR Cycle Analyses

Ambient Condition	Standard ISO Condition(15 °C,1 Bar)
Flow Condition	Steady/One-dimensional
Air Properties	Constant C_p , $\gamma(C_p=1.005 \text{ kJ/kgK}, \gamma=1.4)$
Gas Properties	Constant C_p , $\gamma(C_p=1.147 \text{ kJ/kgK}, \gamma=1.333)$
Intake/Exhaust Duct Losses	"0" Duct Pressure Losses
LP/HP Compressors	$\eta_c=0.905$, OPR=6–20, LP/HP Pressure Ratio Split=0.2–2.0
Intercooler	$\epsilon_{ic}=80\sim90\%$, Pressure Loss=2–3%
Combustion Chamber	$\eta_{cc}=99\sim99.5\%$, Pressure Loss=5.0%
HP Turbine	Air-Cooled Nozzle/Rotor Blades, $m_n=6\%$, $m_r=3\%$ $\eta_{tt}=89\%$
LP Turbine	Uncooled, $\eta_{tt}=90\%$
Power Turbine	Variable Nozzle, $\eta_{tt}=89\%$
Regenerator	$\epsilon_{HE}=80\sim90\%$, Pressure Losses on Both Sides=4.0%
Mechanical Transmission Efficiency	$\eta_m=99\%$ For All Rotors

3.2 FUNDAMENTAL THEORIES OF CYCLE CALCULATION

The cycle calculations are carried out by observing the conservation of mass and the first law of thermodynamics on the air as it is put through a gas turbine cycle. The flow compatibility is illustrated in Figure 3.4. Figure 3.5 shows the energy balance of an ICR cycle. The work compatibility follows the Law of Conservation of Energy.

Fig.3.4 Flow Compatibility in an ICR Gas Turbine

Fig.3.5 Energy Balance in an ICR Gas Turbine

As shown in Figure 3.4, the air flow compatibility among the components of an ICR gas turbine can be given by

$$m_1 = m_2 = m_3 = m_4 + m_b = m_5 + m_b = m_6 + m_b = m_7 = m_8 = m_9 = m_{10}$$

Referring to Fig 3.5, the work relations of the components involved in the ICR cycle can be given by

$$\text{For LP spool: } W_{lpc} = W_{lpt} \quad (3.1)$$

$$\text{For HP spool: } W_{hpc} = W_{hpt} \quad (3.2)$$

these expressions can be further expanded as

$$W_{lpc} = m_1 C_{pa}(T_2 - T_1) \quad (3.3)$$

$$W_{hpc} = m_3 C_{pa}(T_4 - T_3) \quad (3.4)$$

$$W_{hpt} = (m_6 + m_n) \eta_m C_{pg}(T_6 - T_7) \quad (3.5)$$

where m_n indicates the HPT nozzle cooling air which re-joins the gas flow before entering the HP rotor and does work.

$$W_{lpt} = m_7 \eta_m C_{pg}(T_7 - T_8) \quad (3.6)$$

$$\text{For PT spool: } \text{Work Output} = m_8 W_{pt} = \eta_m m_8 C_{pg}(T_8 - T_9) \quad (3.7)$$

The effectiveness for intercooler and heat regenerator can be defined as

$$\epsilon_{ic} = \frac{T_2 - T_3}{T_2 - T_1} \quad (3.8)$$

$$\epsilon_{HE} = \frac{T_5 - T_4}{T_9 - T_4} \quad (3.9)$$

The combustion energy balance equation used in this cycle analysis is Chappell and Cockshutt(1974) and the fuel/air ratio (f) can be expressed as

$$f = \frac{m_f}{M_s} = \frac{h(T_6) - h(T_5)}{\eta_{cc} ECV_6} \quad (3.10)$$

where $h(T_6)$ and $h(T_5)$ are the dry air enthalpy values at the turbine inlet and combustor inlet,

respectively, the ECV_6 indicates the heating value of the fuel at the temperature of T_6 . These values can be found from the Chappell & Cockshutt Thermodynamic Data Table (1974). Knowing the fuel/air ratio f , the cycle efficiency η and the specific fuel consumption can be easily calculated using

$$\eta = \frac{W_{pt}}{fQ_{net,p}} \times 100(\%) \quad (3.11)$$

$$s.f.c. = \frac{3600f}{W_{pt}} (kg/KWhr) \quad (3.12)$$

In any real gas turbines, the processes in both compressor and turbine are polytropic, rather than isentropic. Knowing the polytropic efficiencies, η_{-c} and η_{-t} , these processes can be written as
For compression, with LPC as example,

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\left(\frac{n-1}{n}\right)} \quad \text{where} \quad \frac{n-1}{n} = \frac{1}{\eta_{-c}} \times \frac{\gamma-1}{\gamma} \quad \text{with } \gamma=1.40 \quad (3.13)$$

For Expansion, with HPT as example,

$$\frac{T_6}{T_7} = \left(\frac{P_6}{P_7}\right)^{\left(\frac{n-1}{n}\right)} \quad \text{where} \quad \frac{n-1}{n} = \eta_{-t} \times \left(\frac{\gamma-1}{\gamma}\right) \quad \text{with } \gamma=1.333 \quad (3.14)$$

3.3 DESIGN POINT SOLUTION PROCEDURE

Having examined in the preceding sections all the component and cycle fundamentals, which are required for the ICR DP calculation, a cycle calculation procedure for ICR DP analysis can be programmed.

- (a) Determine a cycle inlet condition T_1 and P_1 from the ambient condition.
- (b) Select a range of proposed TIT and overall pressure ratio (OPR), and the size of increments of these two parameters, and assume the performance for all the components

involved in the ICR cycle, including η_{ac} , η_{hpt} , η_{lpt} , η_{pt} , ϵ_{ic} , ϵ_{HE} , η_{cc} , etc.

- (c) Assume a range of pressure ratio split between LPC and HPC, and use $OPR = LPR \times HPR$ to calculate LPC delivery pressure P_2 .
- (d) For each combination of TIT, OPR and pressure ratio split, use Equ.(3.13) to calculate the LPC delivery temperature, and by assuming mass flow M_1 equal 1 kg/s, use Equ.(3.3) to calculate the LPC work W_{lpc} .
- (e) With the assumed intercooler effectiveness ϵ_{ic} and pressure drop, use Equ.(3.8) to calculate the HPC inlet temperature T_3 and pressure P_3 .
- (f) Use Equ.(3.4) and the polytropic relation, referring to Equ.(3.13), calculate the HPC delivery pressure P_4 and temperature T_4 , and HPC work W_{hpc} .
- (g) With the assumed pressure loss in the combustor, calculate combustor delivery pressure.
- (h) With each assumed TIT [see (b)], use Eqs.(3.5) and (3.2), by assuming 9% cooling bleed m_6 , in which 6% is used for nozzle cooling, calculate the HPT delivery temperature T_7 , ie, use $m_3 C_{pc}(T_4 - T_3) = (m_6 + m_n) \eta_m C_{pg}(T_6 - T_7)$ to solve for T_7 . Use Equ.(3.14) to calculate the P_7 .
- (i) Use the polytropic relation and work compatibility [Equ.(3.1)], calculate the LPT delivery pressure P_8 and temperature T_8 .
- (j) With the assumed "0" exhaust loss and 4% pressure drop in regenerator, thus, P_{10} equals P_1 , use the polytropic relation and Equ.(3.7) calculating the cycle work output and T_9 .
- (k) Use Equ.(3.9) and ϵ_{HE} calculating combustor inlet temperature T_5 .
- (l) With T_5 , TIT and Chappell & Cockshutt Thermodynamics Data Table, use Equ.(3.10) calculating the fuel/air ratio f , the cycle thermal efficiency or specific fuel consumption (SFC) can then be easily calculated using Eqs.(3.11) and (3.12).
- (m) The calculation will repeat for all TIT, OPR and LPC/HPC pressure ratio split.

Note that the purpose of examining the effect of varying the pressure split, ie. the work split, is to find out the optimum region of ICR rotor arrangement in terms of the best thermal efficiency. However, it is important to note that since most marine gas turbines are adaptations of aero gas turbines, the pressure ratio split between HP and LP compressors is basically determined by overall engine design and number of turbine stages in order to minimize weight, rather than by the optimum location of the intercooler. The effects of varying the OPR and TIT are also checked

to determine their influence to the cycle performance.

3.4 RESULTS AND DISCUSSION

The first task of the DP calculation is to examine the change of cycle performance because of the addition of the intercooler and regenerator to the simple cycle gas turbine. Assume that our ICR model is adapted from a twin-spool plus free turbine simple cycle gas turbine with a maximum TIT of 1400 K and an OPR of 12.0 evenly distributed between LPC and HPC. A moderate thermal effectiveness of 85% is chosen for both intercooler and regenerator. Other cycle parameters follows the assumptions given in Section 3.1 The performance comparison of the ICR and simple cycle is tabulated in Table 3.2.

This comparison shows clearly that an ICR gas turbine could offer a 30 % plus efficiency improvement over the simple cycle gas turbines with an associated increase in power by about 15 %. From concept, the improvement can be explained clearly that the use of regenerator helps reduce the exhaust heat loss of gas turbine, thus improves the thermal efficiency, whilst the use of intercooler lowers the compression work of the compressors, so increases the work output with a small increase in thermal efficiency due to the increased work output. This concept was also proved in the cycle calculation by investigating the effects on cycle performance when varying the effectiveness of the intercooler and regenerator. The calculated effects of varying the IC and HE effectiveness from 80 to 90 % on ICR performance are plotted in Figure 3.6 and 3.7. Figure 3.6 shows that the regenerator performance has a greater influence to the cycle efficiency, which means that the use of a regenerator results in a major proportion of the efficiency improvement. The intercooler, however, plays a less important role in increasing the fuel efficiency, but makes a major contribution in improving the work output, and regenerator effectiveness (ϵ_{HE}) will have zero effect on power output, see Figure 3.7.

Table 3.2 Performance Comparison of ICR and Simple Cycle
(Standard ISO Condition)

DP Performance	ICR	Simple Cycle
OPR	12.0	12.0
TIT (K)	1400	1400
LPC Pressure Ratio	3.46	3.46
LPC Delivery Temp.(K)	427.3	427.3
IC Effectiveness(%)	85	—
HPC Inlet Temp.(K)	308.9	427.3
HPC Pressure Ratio	3.46	3.46
Combustor Inlet Temp.(K)	778.4	633.9
HPT Pressure Ratio	1.575	1.61
LPT Inlet Temp.(K)	1256.2	1201.1
LPT Pressure Ratio	1.63	1.99
PT Inlet Temp.(K)	1134.2	1079.1
PT Pressure Ratio	3.98	3.56
PT Outlet Temp.(K)	834.9	814.6
HE Effectiveness(%)	85	—
Cycle Efficiency(%)	43.8	21.8
SFC (lb/hphr)	0.314	0.432
Work Output(KW s/kg)	339.8	300.3

Fig.3.6 Effect of Varying Effectiveness of Regenerator and Intercooler on Cycle Efficiency

Fig.3.7 Effect of Varying Effectiveness of Regenerator and Intercooler on Cycle Work Output

The cycle calculations have been based on the assumption that the pressure ratio of the low-pressure compressor and high-pressure compressor are equal. However, most real gas turbines show different pressure ratios for the LPC and HPC. In an ICR cycle, the intercooler is always located between the low-pressure rotor and high-pressure rotor. The change of pressure ratio distribution implies the shifting of intercooler position. The effects of the pressure split on the cycle performance have also been studied. Although the calculation shows that the optimum pressure ratio split varies very slightly with other engine parameters, e.g., TIT and OPR, the optimum work split, or the optimum pressure ratio split is located around the evenly distributed point. However, as shown in Figure 3.8, the cycle efficiency curve reflects a fairly flat pattern near the optimum point. This indicates that a LPC/HPC arrangement, which is not far away from the optimum point, can also offer a acceptable cycle performance. This characteristic has a happy coincidence with the aero gas turbines. Most two-spool aero gas turbines feature a higher HPC pressure ratio than that of the LPC, but the split ratio seldom goes below 0.3. This fact implies that most aero derivative ICR gas turbines can capture the full advantages of such cycle arrangements.

Fig.3.8 Effect of Varying Work Split of an ICR Cycle on Thermal Efficiency

So far, the analysis is based on an assumed ICR model, which has a turbine inlet temperature of 1400 K, and an overall pressure ratio of 12.0. Actually, one question which has not been targeted is the most important aspect of the ICR DP calculation — the cycle optimization. The cycle optimization is performed by calculating the ICR performance while varying the OPR and TIT from low values to high values, and at the same time observing some major limiting factors, e.g., TIT and regenerator inlet temperature. The cycle optimization results are plotted in Figure 3.9. This figure shows that for a given TIT, the cycle performance does not always increase with the pressure ratio. The pressure ratio should be limited within a range of 8.0 to 13.0. The range of regenerator inlet temperature considered is shown by dashed lines for 1000°F and 1200°F. The 1200°F temperature reflects the maximum allowable operating temperature for modern recuperators. The TIT lines indicate that an improved ICR performance can always be obtained at a higher TIT. The shaded area indicates the design point region.

Fig.3.9 Effect of ICR Parameters on Cycle Performance

From the ICR cycle design point analysis, it can be concluded that an ICR cycle gas turbine is capable of achieving an outstanding thermal efficiency of 43 %, with a 10 to 15 % associated increase in power, relative to a simple cycle. A moderate pressure ratio tends to be used to take full advantage of using regeneration. The anticipated growth in component efficiencies, e.g., the compressor and turbine efficiencies, the intercooler and regenerator effectiveness, and maximum material temperature, indicate that thermal efficiency in the high 40's can be achieved as the system matures. Since 1985, Roll-Royce has initiated two ICR proposals based on the highly reliable Spey and very advanced RB211 gas turbines, the thermal efficiency of both installations is projected over 43%. From the point of view of today, using the proven state-of-the-art aero engine and regenerator technologies, such a technical goal can be achieved without bearing high development risks.

Chapter Four

Some Component Topics

4.0 INTRODUCTION

The ICR cycle analysis in Chapter Three was based on a group of assumptions for the components involved in an ICR gas turbine, for example, the values of polytropic efficiency were assumed for both compressors and turbines, the amount of cooling air flow was chosen for the HPT section, and effectiveness was selected for regenerator. The accuracy of all these assumptions would influence the validation of the cycle analysis. It is, therefore, necessary to carry out an assessment on these component performance, before any performance estimation is implemented on a real engine model. In this chapter, the attention will be focused on the issues of cooled turbine performance, selection of heat regenerator, and variable power turbine nozzle. The chapter ends with a brief introduction of intake/exhaust duct losses of a marine gas turbine.

4.1 TURBINE BLADE COOLING

The topic of turbine blade cooling is not within the scope of this thesis. The application and design of a cooled turbine unit involved broad studies on material, heat transfer and cooling pattern, etc. The reason that this topic is introduced is to modify the cooling assumptions, which were used in Chapter Three, and to determine the aerodynamic penalties due to the existence of turbine blade cooling.

Regarding the coolant mass flow, some data of a number of Roll-Royce engines were released by Barry (1974), see Table 4.1. It is clearly shown in Table 4.1 that the coolant mass flow is closely related to the turbine inlet temperature, but because the blade material and design varies from engine to engine, it is impossible to give a simple relation between TIT and cooling air flow rate. The coolant mass flow for the 1st stage vane is often designed marginally higher than that of the turbine blade, because the vane is subjected to higher temperature; the cooling flow from

the vane is also returned to do work in the turbine rotor. A 6% coolant flow is very typical for 1st stage vane, while 3% is quite common for HP turbine blade cooling in modern gas turbines.

Table 4.1 Typical HP Turbine Blade Coolant Mass Flow

Engine Model	RB211-22B	RB211-524	RC ₄₃
TIT (K)	1493	1548	1340
Coolant Flow	3.35%	4.0%	1.9%

Like a parasitic process, turbine cooling would inevitably result in a drop in turbine performance at the re-entry of the coolant air. Many engine manufacturers and research institutions, including RR and NASA, have carried out extensive studies and tests to evaluate this performance loss. Figure 4.1 illustrates the efficiency penalty associated with HP cooling in a number of RR engines.

Fig.4.1 Efficiency Penalty Associated With Turbine Cooling in Some RR Engines [From Ref.9]

It should be noted that the $\Delta\eta$ in Figure 4.1 indicates the drop in stage isentropic efficiency. In the thesis, however, the polytropic efficiency (η_p) is used for the most of the calculations. The isentropic efficiency drop can be converted into polytropic efficiency drop by using

$$\eta_p = \frac{1 - (HPTR)^{\eta_{is}(\gamma-1)/\gamma}}{1 - (HPTR)^{(\gamma-1)/\gamma}}$$

where the HPTR indicates the HPT pressure ratio. An advanced modern gas turbine can normally offer an uncooled turbine polytropic efficiency of 90 %. If the TIT is about 1400 K, then the calculated polytropic efficiency of a cooled turbine is 89%.

4.2 VARIABLE POWER TURBINE NOZZLE

Variable power turbine nozzle means that the area of the power turbine nozzle, which is formed by the passages between blades, is adjustable by rotating the blades simultaneously about their axes, see Figure 4.2. The function of such installations is to, through varying the nozzle area, control the swallowing capacity in order to maintain a high turbine inlet temperature at part loads. Application of variable power turbine nozzles is particular advantageous when the regenerator is also used, because a higher part-load TIT would result in a better part-load cycle efficiency, and additional exhaust heat flux due to the higher power turbine exhaust temperature would be recovered by the regenerator.

Fig.4.2 Variable Geometry Power Turbine Nozzle [From Ref.16]

Variable power turbine nozzles, as a part of the overall engine control system, have been used in some industrial gas turbines, including the GE MS-5002B, GE LM5000 and the under-development Westinghouse-RR WR-21. Each manufacturer has developed its unique means of nozzle control, the procedure of the PT nozzle control may vary significantly from engine to engine. From available information, only generalities on the single-spool GE MS-5002B will be highlighted. The variable power turbine nozzles are used in the Marine MS-5002B to keep the gas generator compressor speed constant over the 0–50% load range. This speed is sensed by the automatic nozzle angle control, and held at 80% of rated value by angle changes. A typical situation is that, for a single-spool gas turbine, the compressor operating line goes towards the surge line as the nozzle is reduced. (Woodward, 1975)

4.3 REGENERATORS

The regenerator initially found acceptance in large European industrial gas turbines, introduced about 30 years ago. The regenerator was essentially a supplement to the cycle to provide reasonable specific fuel consumption. These regenerators, of mainly tubular construction, were designed and manufactured using conventional heat exchanger practice, and this resulted in large and bulky units which were incompatible with the compact nature of modern naval power plants. With the advancements of gas turbine recuperator technology in the past two decades, the new generation of heat exchangers proposed for marine gas turbines is a type of compact plate-fin stainless steel heat exchanger. The heat exchanger used for intercooling is also of plate-fin type, but the technical and manufacturing difficulties regarding the development of such a heat exchanger appears much less challenging than those of a regenerator, because it is much easier to design a high efficiency air-to-water heat exchanger, because the operating temperature of the intercooler is much lower than that of a regenerator. A complete optimization of a regenerative installation involves the factors of heat exchange effectiveness, pressure drop, ability of resisting high-temperature oxidation, weight and size, and cost, which are beyond the scope of this thesis work. The effectiveness (ϵ) of a heat exchanger is normally defined as the fraction of heat available for transfer that is actually transferred, see Figure 4.3. The frictional pressure losses in the regenerator are closely related to effectiveness (ϵ) and regenerator cost.

Fig.4.3 Definition of Regenerator Effectiveness

Figure 4.4 indicates a typical relations of the effectiveness ϵ , the pressure drop and the regenerator cost. This group of curves was plotted by AiResearch in the late 1970s.

Fig.4.4 Typical Relations of Effectiveness, Pressure Loss and Cost of Regenerator

[From Ref.10]

In the ICR cycle analyses, it is assumed that the latest regenerator technology and the high quality ferritic (400 series) steel are used, so that an effectiveness of 90 % and a pressure drop of 4 %, an operating temperature as high as 1150 °F and a 10-to-20-year oxidation free life can be achieved.

4.4 DESIGN AMBIENT CONDITIONS

The performance of a gas turbine engine is also affected by the pressure and temperature of the air at its compressor inlet, and by its exhaust pressure. In theoretical cycle calculation, the intake and exhaust pressure losses are often omitted, and the ambient conditions is assumed to be the Standard ISO Condition, but when predicting the performance of a real power plant, the losses associated with the intake and exhaust system must be taken into account, and the design-point ambient condition should be based on its designated working condition. U.S. Navy requires that naval gas turbines be designed to operate at 100 °F inlet condition, with 4" H₂O intake and 6"H₂O exhaust duct losses. In Chapter 5, these requirements will be observed in calculating the performance of the WR-21 ICR

Chapter Five

Design Point Performance of WR-21

5.0 WR-21 ICR MARINE GAS TURBINE

The WR-21 is an intercooled and regenerative (ICR) marine gas turbine, currently being developed jointly by Westinghouse and Rolls-Royce. The new engine system has been selected to power future US Navy surface combatants and combat logistics ships. It is a "marinized" adaptation of the Rolls-Royce RB211 aero engine combined with a current state-of-the-art industrial recuperator, and a water cooled intercooler. The engine has a nominal rating of 26400 bhp. Figures 5.1 and 5.2 illustrate the engine structure by cutaway and exploded views. Figure 5.3 is the schematic diagram of the WR-21.

Fig.5.1 Cutaway View of WR-21

As shown in the figures, the WR-21 is composed of a twin-spool gas generator, an intercooler between the low pressure compressor (LPC) and high pressure compressor (HPC), a heat regenerator and a free power turbine. The engine design uses the 1990s modular design concept, which not only eliminates the need for engine removal during maintenance, but also increases the system reliability and reduces the service cost. As illustrated in Figure 5.3, the WR-21 modular components are

- Intake Plenum: Based on existing industrial/marine engines
- Low-pressure Compressor: Adapted from the RB211-535 with modified inlet
- Intercooler: Compact aircraft modular heat exchanger design
- Heat Exchanger: Off-the-shelf freshwater/seawater subsystem
- High-pressure Compressor: RB211-535 unchanged except for casing adaptations
- Combustor: Based on SPEY marine combustor and existing TAY design
- High-pressure Turbine: RB211-524 with minimal changes for revised capacity
- Low-pressure Turbine: RB211-535 intermediate turbine with minimal changes to blade profile
- Power Turbine: Adapted from RB211 TRENT low-pressure turbine with addition of a variable area nozzle
- Exhaust Collector Diffuser: Based on existing industrial and marine designs
- Regenerator: High reliability, industrial based plate-fin heat exchanger

WEIGHT AND DIMENSIONS

Total ICR Weight (Dry) 109,751 Pounds

Fig 5-1 Cutaway View of WR-21

ICR CYCLE

Fig.5.3 Schematic Flow Diagram of WR-21

Fig.5.4 Specific Fuel Consumption of WR-21

According to the WR-21 brochure released by the manufacturer, on a 100 °F day, with 4" H₂O inlet and 6" H₂O exhaust losses, the engine has an air flow rate of 142 lb/sec, a rated power of 26400 bhp and a growth power of 29000 bhp. The exhaust gas temperature is 660 °F. Figure 5.4 illustrates the published specific fuel consumption of the WR-21.

At the ASME conference in June, 1992, some further details on the WR-21 were released in an oral presentation by Rolls-Royce. The new ICR system has an overall compressor pressure ratio of 12.6, with a pressure ratio split between LPC and HPC of 2.8 and 4.5. The intercooler in the WR-21 provides a cooling range of about 200 °F with a pressure loss of 2-3 %. The temperature at the power turbine inlet is reported to be 1535 °F at rated condition. It should be mentioned that some new information shows contradictions with the old data, for example, the intake air flow is reported to be 130 lb/sec, rather than 142 lb/sec as in the product brochure, and rated s.f.c. is 0.33 lb/hp-hr, rather than 0.325 lb/hp-hr.

5.1 OTHER ASSUMPTIONS

Although some of the design point parameters of WR-21 are known, as introduced in the preceding section, they are not enough for the D.P. calculation. Some additional assumptions are made as follows:

- 1) The D.P. ambient condition is 100 °F for temperature, and 1.0 bar for pressure.
- 2) The polytropic efficiencies (η_{hpc} or η_{lpc}) of LPC and HPC are 0.905.
- 3) The combustion efficiency (η_{cc}) is 0.99, with a 5.0 % pressure loss.
- 4) The regenerator effectiveness (ϵ_{HE}) is 0.90, with 4.0 % pressure loss on both the air side and the gas side.
- 5) The polytropic efficiencies (η_{hpt} , η_{lpt} or η_{pt}) of HPT, LPT and PT are 0.89, 0.90 and 0.89, respectively.
- 6) The mechanical transmission efficiency (η_{m}) of all rotors is 99%.
- 7) The compression and expansion processes are considered as adiabatic.
- 8) The average values of fluid properties (C_p and γ) are used. For air, $C_{pa}=1.005$ kJ/kg K, $\gamma_a=1.4$; For gas, $C_{pg}=1.147$ kJ/kg K, $\gamma_g=1.333$.
- 9) The bleed flows for the HPT nozzle and rotor cooling are 6.0 % and 3.0 %, respectively;

The 6.0 % nozzle cooling air re-enter the main gas stream and does work in the HPT rotor.

5.2 WR-21 DESIGN POINT CALCULATION

The purpose of the design point (D.P.) calculation is to reproduce the rated performance of the WR-21 with available engine information and reasonable assumptions for component performance. The accuracy of such a reproduction with respect to the real engine is of great importance, not only because an accurate D.P. itself is important, but also because a poor D.P. reproduction will distort the off-design calculation.

In order to obtain a good D.P. performance, accurate engine data and assumptions are indispensable. However, as introduced above, available engine data are far from complete, and furthermore, there is some disagreement among them, plus some assumptions are necessary. This D.P. calculation is actually a performance estimation. The D.P. calculation resulted in two different set of results based on two different sets of engine data, see Table 5.1.

Table 5.1 WR-21 Engine Data

WR-21 specifications	Model 1	Model 2
Inlet Conditions	100°F, 4" inlet and 6" exhaust losses (for both sets)	
Intake Air Flow (lb/sec)	142.0 ^{**}	130.0
Pressure Ratio (LPR × HPR)	2.8 × 4.5	2.8 × 4.5
Rated Power (bhp)	26400 [*]	26400 [*]
LPT Turbine Outlet Temperature (°F)	1535	not specified
Specific Fuel Consumption (lb/hp-hr)	0.325	0.325

(^{**} Ratings are at 100 °F, 40% relative humidity with 4" inlet and 6" exhaust losses and 0 customer bleed; ^{*} Characteristics at rated power condition)

The engine parameters shown in Table 5.1 are obtained from different sources, the data with the same superscript indicate the performance at the same operating condition. However, it is not known whether all these data are acquired at the same operating condition. The combination of known parameters for the calculations in Model 1 and Model 2 were not selected randomly. The

major difference between Model 1 and Model 2 is that the intake air flow used in the calculation is based on two different source data. In the calculation, checking of calculated s.f.c. with published s.f.c. offers a major criterion for the D.P. calculation. As with the first set of data, it is needed to assume a proper group of component performance data, such as compressor and turbine efficiencies, intercooler and regenerator effectiveness, and pressure losses in these components, so as to meet the published s.f.c.. In the calculation with the second set of engine data, a turbine inlet temperature is assumed, instead of using the LP turbine outlet temperature of 1535 °F, together with other component assumptions, in order to obtain the same s.f.c. and similar exhaust gas temperature.

5.3 DESIGN POINT CALCULATION PROCEDURE

The theories used in gas turbine cycle calculation have been introduced in Chapter 3. The station numbering system is given in Figure 3.1. The calculation for WR-21 D.P. performance proceeds as follows:

- a) With ambient conditions (T_a, P_a) and intake pressure loss (ΔP_{intake}), calculate the engine inlet temperature (T_1) and pressure (P_1) using $T_a = T_1$ and $P_1 = P_a - \Delta P_{\text{intake}}$.
- b) Calculate the polytropic parameter $(n-1)/n$ for compressors and turbines by using

$$\text{For compression} \quad \frac{n-1}{n} = \frac{1}{\eta_{\infty}} \left(\frac{\gamma-1}{\gamma} \right)_a, \quad \text{For expansion} \quad \frac{n-1}{n} = \eta_{\infty} \left(\frac{\gamma-1}{\gamma} \right)_g$$

- c) With LPC pressure ratio (LPR), T_1 and P_1 , calculate P_2 , T_2 and LPC compression work (W_{lpc}) using $P_2 = P_1 \times \text{LPR}$,

$$T_2 = T_a \left(\frac{P_2}{P_1} \right)^{\frac{n-1}{n}} \quad \text{and} \quad W_{\text{lpc}} = C_{pc} T_a \left[\left(\frac{P_2}{P_1} \right)^{\frac{n-1}{n}} - 1 \right]$$

- d) Knowing the intercooler effectiveness (ϵ_{ic}) and pressure loss (ΔP_{ic}), calculate P_3 and T_3 using $T_3 = T_2 - \epsilon_{\text{ic}}(T_2 - T_1)$ and $P_3 = P_2(1 - \Delta P_{\text{ic}})$.
- e) Knowing HPC pressure ratio (HPR), calculate HPC delivery temperature (T_4) and pressure (P_4) and HPC compression work (W_{hpc}) using $P_4 = P_3 \times \text{HPR}$,

$$T_4 = T_3 \left(\frac{P_4}{P_3} \right)^{\frac{\gamma-1}{\gamma}} \quad \text{and} \quad W_{hpc} = C_{pa} T_3 \left[\left(\frac{P_4}{P_3} \right)^{\frac{\gamma-1}{\gamma}} - 1 \right]$$

- f) Knowing the pressure losses on the air-side of the regenerator (ΔP_{HEA}) and in the combustion chamber (ΔP_{cc}), the combustor delivery pressure can be calculated using

$$P_6 = P_4(1 - \Delta P_{HEA})(1 - \Delta P_{cc}).$$

- g) From work compatibility of HP and LP rotors, with the LPT outlet temperature ($T_8=1535^\circ\text{F}$), calculate temperature drop in HPT and LPT, and turbine inlet temperature (TIT or T_6), using $W_{lpc} = W_{lpt}$, $W_{hpc} = (1 - m_n)W_{hpt}$, where $W_{lpt} = C_{pg}(T_7 - T_8)$, $W_{hpt} = C_{pg}(T_6 - T_7)$ and m_n is cooling flow for HPT nozzle blades.

- h) With T_6 , T_7 and T_8 , use the polytropic relations to find out P_8 and P_7 ,

$$\frac{T_6}{T_7} = \left(\frac{P_6}{P_7} \right)^{\frac{\gamma-1}{\gamma}} \quad \text{and} \quad \frac{T_7}{T_8} = \left(\frac{P_7}{P_8} \right)^{\frac{\gamma-1}{\gamma}}$$

- i) Knowing the air flow of engine (m) and rated power output (W_{pt}), T_9 and P_9 can be calculated using $W_{pt} = mC_{pg}\eta_m(T_8 - T_9) = mC_{pg}\eta_m T_8 [1 - (T_9/T_8)^{(\gamma-1)/\gamma}]$.
- j) With calculated T_9 and P_9 , and ϵ_{he} , find T_5 using $T_5 = T_4 + \epsilon_{HE}(T_9 - T_4)$.
- k) With TIT, T_5 and W_{pt}/m , using the Chappell & Cockshutt tables, the fuel flow can be found and the cycle s.f.c. can be obtained.
- l) Check calculated s.f.c. with published engine data; if the same results are obtained, terminate the calculation, otherwise, adjust the component performance assumptions, and go through a) to l) again.

Only the step g) needs some revision for the second group of calculations, when TIT becomes a known parameter, whilst T_8 is treated as unknown.

5.4 RESULTS OF D.P. CALCULATION

The calculated results of both design model calculations are tabulated in Table 5.2.

Table 5.2 WR-21 D.P. Calculation Results (Set 1 and 2)

Input	Model 1	Model 2
Ambient Condition(°F,Bar)	100.0/1.0	100.0/1.0
Intake Pressure Loss(inch-water)	4.0	4.0
Intake Air Flow(lb/s)	142.0	130.0
Polytropic Efficiency of LPC (%)	91	90
Cooling Range and ΔP_{ic} of Intercooler (°F,%)	200.0/2.0	200.0/2.0
Polytropic Efficiency of HPC (%)	91	90
Combustion Efficiency and ΔP_{cc} (%,%)	99.5/5.0	99.5/5.0
Overall Pressure Ratio(LPR $\times$ HPR)	2.8 $\times$ 4.5	2.8 $\times$ 4.5
PT Polytropic Efficiency(%)	89	89
Cooling Air for HPT nozzle/Rotor(%)	6.0/3.0	6.0/3.0
LPT Polytropic Efficiency(%)	90	90
Power Turbine Efficiency(%)	89	89
LPT Delivery Temperature(°F)	1535.0	—
Turbine Inlet Temperature(°F)	—	2150.0
Exhaust Pressure Loss(inch-water)	6.0	6.0
Regenerator Effectiveness/Pressure Loss(%)	90/3.0 both side	86/3.0 both side

Results	Model 1	Model 2
Pressure/Temperature at LPC Outlet(Bar/K)	2.77/430	2.77/431
Pressure/Temperature at IC Outlet(Bar/K)	2.72/319	2.72/320
Pressure/Temperature at HPC Outlet(Bar/K)	12.2/511	12.2/516
Pressure/Temperature at CC Inlet(Bar/K)	11.8/806	11.8/824
Pressure/Temperature at HPT Inlet(Bar/K)	11.1/1386	11.3/1450
HPT Pressure Ratio	1.826	1.80
Pressure/Temperature at LPT Inlet(Bar/K)	6.16/1212	6.21/1272
LPT Pressure Ratio	1.49	1.47
Pressure/Temperature at PT Inlet(Bar/K)	4.14/1108	4.23/1167
PT Pressure Ratio	3.49	3.69
Pressure/Temperature at PT Outlet(Bar/K)	1.18/839	1.15/874
Temperature at Exhaust(K)	581	604
Work Output (bhp)	26400	26400
Specific Fuel Consumption(lb/hp-hr)	0.3253	0.3254

5.5 VALIDATING ANALYSIS

From the above discussions and calculations, it is clear that not only do the component assumptions influence the results, also the accuracy of published engine data dictates the accuracy of cycle modelling. Since the WR-21 is a new system under development, it is unlikely for the time being to obtain accurate engine specifications. The two sets of D.P. calculations are based on engine information from different sources. It is necessary then to distinguish which model represents the WR-21 best.

A number of input data can be classified as more important than others in determining the engine performance, they include compressor and turbine efficiencies, heat regenerator effectiveness, overall pressure ratio and turbine inlet temperature. From source information, it is believed that the overall pressure ratio of 2.8×4.5 is reliable. Regarding the compressor and turbine performance, the design and manufacturing technology of today is the result of a long process of evolution, to the point where further significant increase in efficiency or working temperature can only be achieved after long and costly design and development programmes, resulting in higher cost and often, more complexity. It is believed that the technologies used in RB-211 reflects the current state-of-the-art component design, so the selections of compressor and turbine polytropic efficiencies ($\eta_{c,p}$ and $\eta_{t,p}$) are limited to a range of 0.89–0.92, rather than a free selection. The selected values of $\eta_{c,p}$ for both Model 1 and Model 2 in the D.P. calculation can be considered relevant to the current technology.

The watershed between the Model 1 and Model 2 is the determination of turbine inlet temperature (TIT). Modern techniques are capable of pushing the TIT up to the 1500+ K region. Figure 5.5 illustrates the TIT development from 1950s to 1975. Although the test results are taken 15 years ago, the maximum TIT of RB-211 has shown small changes in the later versions of the engine family.

Fig.5.5 Improvement of Turbine Inlt Temperature Over the Years

In the maritime or industrial application, however, the TIT is normally set lower to ensure a longer service life. The TIT of some typical main-stream marine gas turbines and their correspondent aero engines are tabulated in Table 5.3. Data in Table 5.3 shows a aero gas turbine normally has a TIT of 60–130 °C higher than that of its "marinized" derivative. It should be noted that the data of TIT quoted in Table 8 represents the performance of a particular model in that model series, and does not apply to other models in the same engine family.

Table 5.3 Turbine Inlet Temperature of Some Marine Gas Turbines

(From Jane's All World Aircraft,etc.)

Marine	FT9	Coberra6356	LM2500	Spey	LM1500	FT4
TIT(K)	1466	1389	1441	1300	1151	1352
Aero	JT9D	RB211	CF6	Spey	J79	J75
TIT(K)	1550	1515	1543	1356	1281	——

The D.P. calculation of WR-21 with Model 1 shows that, because of the use of available gas generator delivery temperature of 1535 °F which is basically decided by TIT, calculated TIT is 1386 K (2035°F), which appears to be a bit too low for the capacity of RB211-524 high-pressure turbine, a further consequence of such a low TIT makes it necessary to use a highly effective heat regenerator in order to meet the s.f.c. requirement of 0.325 lb/hp-hr, resulting in a higher first cost on regenerator. Based on an intake air flow rate of 142.0 lb/s, it is difficult to establish another calculation model by dropping the known 1535 °F PT inlet temperature and assuming a higher TIT, because a higher TIT would result in a smaller pressure drop in the turbines for the same compression work and an exhaust pressure marginally higher than the ambient condition, which is not the case in a real gas turbine. The results from Model 1 appears to be a possible engine design corresponding to published WR-21 data.

The second model of WR-21 design point calculation is based on the information of 130 lb/s air flow, which was obtained form an oral presentation on WR-21 by the engine manufacturer. This model allows a higher TIT and lower component efficiencies, such as the polytropic efficiency of compressor and heat regenerator effectiveness. A 86% regenerator effectiveness appears to be more realistic than 90 %. The assumed TIT of 1450°K can be successfully achieved using the up to date turbine technologies without sacrificing the engine life.

One thing in common for the two calculation models is that the power turbine exit pressure is controlled at about 1.16 bar, of which a pressure drop of 0.16 bar represents the losses

associated with the regenerator on the gas side, the exhaust ducting system and exhaust kinetic energy.

5.6 CONCLUSIONS

The design point calculation of the WR-21 is based on published information about this engine and some other source information. The main reference is the manufacturer's sales brochure(Roll-Royce, 1992).

Although it has been shown that some oral information of engine could end up with a more reasonable D.P. performance, the engine design point parameters will be based on Model 1, which is fully supported by available information.

Chapter Six

Off-Design Calculation of WR-21

6.0 INTRODUCTION

As mentioned earlier in this thesis, most turbines do not always run at their design point. The marine propulsion gas turbine could spend 70% or more of its service life cruising at low power. When talking about the *Design Point*, the gas turbine is running at the particular speed, pressure ratio and mass flow for which the components were designed. The *Off-Design* operation, however, means the engine is operating at other conditions, e.g., 50% of the rated power, or a hot or cold day. The problem then remains to find the variation of performance over the complete operating range of speed and power output, which is normally referred to as off-design performance. A further problem is how the individual components, particularly the compressor and turbine, behave when the engine operation shifts from the design point. Unlike the design point calculations, which can simply be carried out by following the flow path and doing the basic thermodynamic calculations, the off-design calculations depend on satisfying the essential conditions of compatibility of mass flow, work and rotational speed between the various components.

In this chapter, the off-design performance of WR-21 will be studied, firstly based on the assumption of fixed power turbine nozzle, and then with the variable power turbine nozzle introduced.

6.1 GENERALIZED COMPRESSOR AND TURBINE CHARACTERISTICS

The performance characteristics of compressors and turbines are normally estimated on the basis of previous experience or obtained from actual tests. However, because most engine manufacturers are unwilling to release the detailed test results of their products, the off-design studies in most cases have to depend on available performance characteristics of the individual components, e.g. generalized compressor and turbine characteristics.

A complete characteristic for each compressor or turbine can be plotted with respect to pressure ratio (P_2/P_1 or P_3/P_4), non-dimensional flow ($m\sqrt{T_1}/P_1$ or $m\sqrt{T_3}/P_3$), non-dimensional rotational speed ($N/\sqrt{T_1}$ or $N/\sqrt{T_3}$) and isentropic efficiency. An example for both compressor and turbine is shown in Figure 6.1 and 6.2.

Fig. 6.1 A Typical Compressor Characteristic

[From Ref.16]

Fig.6.2 A Typical Turbine Characteristic [From Ref.16]

It is apparent that for a different compressor or turbine, a different set of performance characteristic diagram can be drawn. This fact forms a major obstacles in the off-design performance studies, because on most occasions, the detailed component characteristic diagrams are not available. The solution goes to generalizing the component characteristics, hoping that after generalization the results from experiments based on different components will fall approximately on a single set of curves. Then, research studies can be carried out using generalized component characteristic diagrams. The generalized compressor and turbine characteristics used in our cycle analysis, which are shown in Figure 6.3 and 6.4, were obtained by normalizing the early test results based on many early engines. These two performance diagrams proved very reliable and accurate in previous use, especially when the pressure ratio is below 6.0.

Fig.6.3 Generalized Compressor Characteristic [From Ref.14]

Fig 6 4 Generalized Turbine Characteristic [From Ref 14]

It is seen that the efficiency plot in Figure 6.2 does not appear in the generalized turbine characteristics (Figure 6.4). This is because, firstly, the turbine efficiency is sensibly constant over a wide range of rotational speed and pressure ratio, and secondly, the turbine operation is normally restricted to a relative narrow running range, with exception in the case of power turbine. The turbine efficiency is treated as a constant in the generalized turbine characteristics. It should also be noted that the efficiency plots in the generalized diagrams are based on isentropic efficiency, rather than polytropic efficiency. Since the engine design-point efficiency was specified in polytropic efficiency, the relations between isentropic and polytropic efficiencies must be established in order to use the generalized performance diagrams. The relation between η_c and η for the compressor can be given by

$$\eta_c = \frac{(P_2/P_1)^{(\gamma-1)\eta_c} - 1}{(P_2/P_1)^{(\gamma-1)\eta_c} - 1} \quad \text{where} \quad \eta_c \equiv \frac{T_2' - T_1}{T_2 - T_1} \quad (6.1)$$

For a turbine, this relation is

$$\eta_t = \frac{1 - (P_4/P_3)^{\eta_t(\gamma-1)/\gamma} - 1}{1 - (P_4/P_3)^{(\gamma-1)/\gamma}} \quad \text{where} \quad \eta_t \equiv \frac{T_3 - T_4}{T_3 - T_4'} \quad (6.2)$$

6.2 OFF-DESIGN OPERATION OF THE ICR SYSTEM

Referring to the two-spool ICR system shown in Figure 3.1, it can readily be seen that the high-pressure rotor, the low-pressure rotor and the free power turbine are mechanically independent with one another. However, these three sub-systems, together with the intercooler and the regenerator, are aerodynamically coupled. Although the basic thermodynamic theories, which have been introduced in the preceding design point analysis, are still applicable in the off-design calculation, they are not enough in solving all the problems arising with the off-design operation. It is noted that in off-design operation, as the rotors move to another non-dimensional shaft speed $N/\sqrt{T}$, a new equilibrium among the rotors is established based on changed performance of the

individual components, resulting in a change of the whole engine performance. So the main task of the cycle off-design analysis is to determine the correspondent equilibrium for each off-design working condition.

For a twin-spool ICR system, the establishment of an equilibrium running point involves not only the matching between compressors and turbines, but also the matching among the LP, HP and PT rotors. Based on the model of the WR-21, the procedure for obtaining an equilibrium operating point can be given as follows, in general terms.

(a) With calculated results of WR-21 DP performance and Generalized Compressor and Turbine Characteristics, produce the performance characteristics of HPC, LPC, HPT, LPT and PT.

(b) Since the LPT of the WR-21 is not choked at the design point, the HP turbine operation is restricted by the swallowing ability of the LP turbine, the operation of which is also determined by the swallowing ability of the power turbine. Based on a fixed or an assumed power turbine nozzle position, find the HPT operation and establish the matching of HP rotor.

(c) Having matched HP rotor, find the equilibrium running line on LP characteristics.

6.3 COMPONENT CHARACTERISTICS OF WR-21

The design-point calculation of the WR-21 has resulted in the temperatures and pressures at the inlet or outlet of each individual component. With these DP parameters, generalized compressor and turbine characteristics can then be converted into actual component performance diagrams.

These characteristics are shown in Figure 6.5~6.9.

Fig.6.5 Low-Pressure Compressor Characteristic of WR-21

Fig.6.6 High-Pressure compressor Characteristic of WR-21

Fig.6.7 Low-Pressure Turbine Characteristic of WR-21 At Rated PT Nozzle Condition

Fig.6.8 High-Pressure Turbine Characteristic of WR-21 at Rated PT Nozzle Condition

Fig.6.9 Power Turbine Characteristic of WR-21 for Standard Nozzle Area

6.4 COMPATIBILITY OF MASS FLOW

Although the flow condition in a gas turbine always follows the Continuity Equation, the expressions, however, should be given in terms of non-dimensional flow by the identity so as to establish the inherent relation between LPC and HPC characteristics, as

$$\text{For LPC / HPC} \quad \frac{m_3 \sqrt{T_3}}{P_3} = \frac{m_1 \sqrt{T_1}}{P_1} \times \frac{P_1}{P_2} \times \frac{P_2}{P_3} \times \sqrt{\frac{T_3}{T_1}} \times \frac{m_3}{m_1} \quad (6.3)$$

where $m_1 \sqrt{T_1}/P_1$ and $m_3 \sqrt{T_3}/P_3$ are the non-dimensional flow at the inlet of the LP compressor and HP compressor, respectively; P_2/P_1 is the LPC pressure ratio; P_3/P_2 can be obtained from the intercooler pressure loss, i.e. $P_3/P_2 = 1 - (\Delta P_i/P_2)$; m_3/m_1 is the mass flow ratio, which equals 1 hereby, and the temperature ratio $\sqrt{(T_3/T_1)}$ can be obtained from the calculation on the intercooling.

For HP rotor, the flow relation can be written as

$$\text{For HPC / HPT} \quad \frac{m_6 \sqrt{T_6}}{P_6} = \frac{m_3 \sqrt{T_3}}{P_3} \times \frac{P_3}{P_4} \times \frac{P_4}{P_5} \times \frac{P_5}{P_6} \times \sqrt{\frac{T_6}{T_1}} \times \frac{m_6}{m_1} \quad (6.4)$$

where P_4/P_3 can be obtained from the air-side pressure loss of the regenerator; the P_4/P_6 reflects the pressure loss inside the combustion chamber; and m_6/m_1 can be calculated by considering the cooling air bleed flow. Given the HPC inlet and the information of flow conditions of the LPT and PT turbines followed, the equilibrium working conditions of the HP rotor can be determined using the Equ.6.4 and work compatibility between the HPC and HPT, which will be discussed later in this chapter.

The flow compatibility for turbines in series features a very important aspect of gas turbine operation. The highly efficient modern gas turbines are often so designed that at some portion of their turbines, very often at the nozzle path, the flow speed reaches Mach number 1, which means the flow path is "choked". As shown in Generalized Turbine Characteristics, as turbine pressure ratio increases, the non-dimensional flow curve gradually levels off and becomes a horizontal line, which indicates that after a certain pressure ratio, the turbine path has been choked and no further increase in mass flow is allowed. This aerodynamic behaviour sets up a major restriction on the operation of the turbines in the front. This argument can also be proved correct by the flow compatibility of turbines. The flow compatibility between the HPT and LPT can be given by

$$\frac{m_7 \sqrt{T_7}}{P_7} = \frac{m_6 \sqrt{T_6}}{P_6} \times \frac{P_6}{P_7} \times \sqrt{\frac{T_7}{T_6}} \times \frac{m_7}{m_6} \quad (6.5)$$

$$\text{Where} \quad \sqrt{\frac{T_7}{T_6}} = \sqrt{1 - \frac{\Delta T_{67}}{T_6}} \quad \text{also} \quad \frac{\Delta T_{67}}{T_6} = \eta_t \left[1 - \left(\frac{P_7}{P_6} \right)^{(\gamma-1)/\eta_t} \right]$$

From this equation, it can be seen that as the HPT pressure ratio increases, the calculated HPT exit mass flow ($m_7 \sqrt{T_7}/P_7$) keeps increasing, the trend of which is represented by the specified dashed line on Figure 6.10. However, the turbine characteristics show that the choking of the LPT limits the increase of $m_7 \sqrt{T_7}/P_7$, hence, also limits the HPT pressure ratio. This relation will be shown more clearly after the flow relation between the low-pressure turbine and power turbine is written.

$$\text{For LPT / PT} \quad \frac{m_8 \sqrt{T_8}}{P_8} = \frac{m_7 \sqrt{T_7}}{P_7} \times \frac{P_7}{P_8} \times \sqrt{\frac{T_8}{T_7}} \times \frac{m_8}{m_7} \quad (6.6)$$

This equation shows that the LPT inlet flow $m_7 \sqrt{T_7}/P_7$ is related with the power turbine inlet flow $m_8 \sqrt{T_8}/P_8$, and as shown in Figure 6.10, the choking of the power turbine nozzle will limit and fix the operation and pressure ratio in the LPT, the swallowing capacity of which in turn determines the operation and pressure ratio of the HPT. The point "C" in Figure 6.10 indicates the onset of the choking for each turbine. If the use of variable area nozzles is considered, the geometry area of the turbine path can be increased or reduced, which means the PT swallowing ability can be controlled. The matching of three turbines in series based on the WR-21 engine is shown in Figure 6.10. The point "A" illustrates the operating point of the HP and LP turbines for the design-point PT nozzle position. The point "B" indicates the new operating point of the HPT and LPT for the reduced part-load PT nozzle position.

6.5 WORK COMPATIBILITY

The HPC and the HPT, and the LPC and LPT are mechanically coupled together, their work compatibility can be expressed as

$$(m_6 + m_n) \eta_m C_{pg} \Delta T_{67} = m_3 C_{pa} \Delta T_{34} \quad (6.7)$$

where m_n represents the cooling flow for HP turbine stators.

$$m_7 \eta_m C_{pg} \Delta T_{78} = m_1 C_{pa} \Delta T_{12} \quad (6.8)$$

The net work output corresponding to the selected operating point is calculated by

$$\text{net work output} = \eta_m C_{pg} m \Delta T_{89} = \eta_m \eta_t C_{pg} m T_8 \left[1 - \left(\frac{P_9}{P_8} \right)^{(\gamma-1)/\eta} \right] \quad (6.9)$$

Fig 6.10 Flow Compatibility of WR-21 Turbines

6.6 EQUILIBRIUM RUNNING OF HP ROTOR

The off-design calculations for a twin-spool ICR gas turbine can be lengthy but they may be considerably simplified if the operation of the HP turbine is fixed by the throat area of the LP turbine stators. In the case of the WR-21 engine, the design-point calculations reveal that the pressure ratio of the power turbine is about 3.5, which means the PT nozzle is always choked. If no variable power turbine nozzle were used, the operation of the HP turbine would be fixed at the rated condition. However, it is known that a variable power turbine is used in the WR-21 to control the off-design operation, thus, a series of fixed LPT and HPT operating points can be established for different PT nozzle settings. Since the studies on the issue of variable geometry requires detailed information of nozzle control, that is not available here, to simplify the matching procedure, it is assumed that the engine is operated at the design-point PT nozzle area. Then, the position of the unique running line on the HP characteristic can be determined by considering the HP rotor alone with respect to the specified nozzle position. The issue associated with the variable power turbine nozzle will be studied separately later in this chapter.

Considerations of compatibility of flow and work have been discussed in the preceding sections. The equation (6.4) is still applicable. The equation (6.7), however, has to be re-written in terms of non-dimensional groups

$$\frac{\Delta T_{67}}{T_6} = \frac{\Delta T_{34}}{T_3} \times \frac{T_1}{T_6} \times \frac{C_{pa}}{C_{pg} \eta_m (1 - m_c)} \quad (6.10)$$

Equations (6.4) and (6.10) are all linked by the temperature ratio T_6/T_3 , where T_6 is the turbine inlet temperature. It is then possible to determine, by trial and error, the temperature ratio T_6/T_3 required at any arbitrary point on each constant speed on the HPC characteristic. The procedure is as follows:

- (a) Having selected a point on a constant speed line on the HPC characteristic, the values of P_4/P_1 , $m/T_4/P_1$, and η_c are determined, and $\Delta T_{34}/T_3$ can be calculated using

$$\frac{\Delta T_u}{T_3} = \frac{1}{\eta_c} [(P_4/P_3)^{(\gamma-1)/\gamma} - 1] \quad (6.11)$$

(b) The HPT pressure ratio (P_4/P_3) is fixed by choking of the power turbine nozzle, the value of $m\sqrt{T_4/P_4}$ can be obtained from the HP turbine characteristic, enable the temperature ratio T_4/T_3 to be obtained from the flow compatibility equation (6.4).

(c) Knowing the HPT pressure ratio (P_4/P_3), it is possible to calculate the $\Delta T_u/T_3$ using

$$\frac{\Delta T_{67}}{T_6} = \eta_t [1 - (P_7/P_6)^{(\gamma-1)/\gamma}] \quad (6.12)$$

and from equation (6.10), another value of T_4/T_3 can be calculated.

(d) This second value of T_4/T_3 will not, in general, agree with the first value obtained from equation (6.4), indicating that the selected point on the constant speed line on the HP compressor characteristic is not an equilibrium operating point.

(e) A new point on the same constant speed line on the HPC characteristic must be assumed and the above calculations repeated until the same value of T_4/T_3 is obtained from both equation (6.4) and (6.10).

(f) Agreement signifies that the compressor operating point is compatible with the fixed HPT operation.

It would be possible to carry out this iteration for each constant speed line on the HPC characteristic. By joining up all the calculated equilibrium running points, an operating line can be established on the HPC characteristic. Figure 6.11 illustrates the HPC operating line of the WR-21 at fixed PT nozzle condition. Figure 6.12 shows the HP compressor pressure ratio at calculated equilibrium operating points.

Fig 6 11 HPC Operating Line of WR-21 for Fixed PT Nozzle

Fig.6.12 HPC Pressure Ratio at Equilibrium Operating Points

6.7 EQUILIBRIUM RUNNING OF LP ROTOR

Having established the equilibrium operating lines of the HP rotor, the operating point of the LP rotor can be calculated by satisfying compatibility of flow between the spools.

Iteration is also required in this calculation, the procedure of which can be summarized as follows.

- Determine the inlet conditions T_1 and P_1 from the ambient conditions and the intake duct loss.
- Select any point on a line of constant non-dimensional speed ($N_1/\sqrt{T_1}$) on the LP compressor characteristic, which will specify values of $m\sqrt{T_1}/P_1$, P_2/P_1 and η_{cL} . The compressor temperature ratio T_2/T_1 can then be calculated using the isentropic relation.

$$\frac{T_2}{T_1} = \frac{1}{\eta_{cL}} \left[\left(\frac{P_2}{P_1} \right)^{(\gamma-1)/\gamma} - 1 \right] + 1$$

- The non-dimensional flow at LP compressor exit, $M\sqrt{T_1}/P_1$, is obtained from the identity

equation(6.3) and calculated temperature and pressure drop inside the intercooler.

(d) With the operating line on the HP compressor characteristic determined by the swallowing capacity of the LPT for specified power turbine nozzle area, the known value of $m\sqrt{T_3/P_3}$ defines the operating point on the HP operating line. This gives the values of P_4/P_3 , $N_H/\sqrt{T_3}$ and η_{cH} , similar to (b), the values of ΔT_{34} and T_4/T_3 can be found.

(e) The overall compressor pressure ratio can then be calculated by

$$OPR = \frac{P_2}{P_1} \times \frac{P_4}{P_3}$$

(f) Subtracting the combustion pressure loss and pressure loss on the air side of regenerator, the turbine inlet pressure can be found.

(g) The non-dimensional flow at the HP turbine inlet ($m_6\sqrt{T_6/P_6}$) has been determined by flow compatibility among the HP, LP and power turbines. The m can be found from $m\sqrt{T_1/P_1}$, T_1 and P_1 . Knowing the cooling flow arrangement, the m_6 can be found.

(h) The turbine inlet temperature (TIT or T_6) can now be found from

$$TIT = \left(\frac{m_6\sqrt{T_6}}{P_6} \times \frac{P_6}{m_6} \right)^2$$

(i) From the flow compatibility of turbines in series, at the fixed PT nozzle condition, the HP and LP pressure ratios, $\Delta T_{67}/T_6$ and $\Delta T_{78}/T_7$ are all fixed at the rated values; (At other PT nozzle conditions, these parameters can also be determined by flow compatibility of the turbines.)

(j) Knowing the pressure ratios of the HPT and LPT, $\sqrt{T_{67}/T_6}$, $\Delta T_{78}/T_7$, P_6 and TIT, the temperature and pressure at the outlet of HPT and LPT can be calculated, furthermore, the non-dimensional flow at the power turbine inlet ($m\sqrt{T_8/P_8}$) can be calculated.

(k) Having obtained two values of $m\sqrt{T_8/P_8}$, (one of them is defined by choking of the power turbine), these two values will not, in general, agree with one another, indicating the selected point on the LP compressor characteristic is not a valid operating point.

(l) It is necessary to select a new point on the same constant speed line on the LPC characteristic

- and repeat the steps from (a) to (k).
- (m) Agreement signifies that the chosen point on the LP compressor characteristic is an equilibrium operating point. Repeat the procedure for all constant speed lines on the LPC characteristic and link up all the valid operating points, a LP compressor running line can be established.
- (n) Having found the LP operating point on a constant speed line, it is possible to find the power turbine pressure ratio by considering the pressure losses in the exhaust system and regenerator(gas side). Knowing the T_8 from step (j), it is possible to calculate the power output of the power turbine and T_9 .
- (o) Knowing the ϵ_{HE} , the combustor inlet temperature can then be determined. The cycle efficiency or specific fuel consumption(S.F.C) can be calculated.

6.8 RESULTS AND DISCUSSIONS

At fixed PT nozzle condition, the calculated LPC equilibrium running line is illustrated on the LPC characteristic in Figure 6.13. Knowing the LPC and HPC operating lines, detailed engine parameters for off-design operations can be established. The off-design performance of the WR-21 is tabulated in Table 6.1.

Table 6.1 WR-21 Off-Design Performance at Fixed PT Nozzle Condition

$N_L(\%)$	$N_H(\%)$	LPCR	HPCR	TIT(K)	ITT(K)	PTTT(K)	Wpt(%)	SFC(lb/hphr)
105	97	3.02	4.38	1386	1212	1108	109	0.3190
100	97	2.82	4.38	1376	1204	1100	98	0.3254
95	97	2.53	4.37	1366	1195	1092	81	0.3419
90	97	2.26	4.37	1360	1190	1088	66	0.3615
80	98	1.81	4.42	1367	1196	1093	43	0.4166
70	97	1.41	4.37	1336	1169	1068	23	0.5339

Fig 6 13 LPC Operating Line of WR-21 for Fixed PT Nozzle

The engine parameters at the off-design conditions can be illustrated more clearly using graphic plots. Figure 6.14 indicates one of significant differences between a single-spool ICR and a twin-spool ICR installation, the turbine inlet temperature of a twin-spool ICR at part-loads reflects a quite flat a. 1 stabilized pattern, rather than a declining trend as that in a single-spool ICR, like the TF-10. This is primarily because, in the case of the twin-spool ICR, the intercooler stabilizes the HPC inlet $m\sqrt{T_3}/P_3$ by keeping T_3 relatively constant, a direct result of that is the HP rotor will operate at roughly constant speed over a wide operating range, as shown in Figure 6.15. The effect of intercooler on $m\sqrt{T_3}/P_3$ is shown in Figure 6.16 in comparison with $m\sqrt{T_3}/P_3$ at the LPC outlet. Since the operating condition of the HPT and the LPT is fixed by the choking of the PT nozzle, temperature at the LPT inlet and the PT inlet shows a similar trend to the TIT curve, see Figure 6.16.

Fig.6.14 Comparison of Turbine Inlet Temperature Between Single and Twin Spool ICR Gas Turbines

Fig 6.15 LP and HP Rotor Speed Vs. Power

Fig 6.16 Effects of Intercooler on HPC Inlet Non-Dimensional Flow

Fig.6.17 TIT, ITT and PTIT at Off-Design Conditions for fixed PT Nozzle

Figure 6.18 illustrates the effect of varying LP rotor speed on the compressor pressure ratios.

Figure 6.19 shows the relation between overall pressure ratio and power output.

Fig.6.18 LP Rotor Speed vs. Compressor Pressure Ratios

HIGH PRESSURE COMPRESSOR CHIC of WR-21 ICR Cycle Engine

6.9 EFFECT OF AMBIENT CONDITIONS

The effects of ambient conditions on maximum power output is often an indispensable part in the off-design analysis. Because a higher atmospheric temperature will result in an increased compression work for the same pressure rise, as a result, the useful power output will be lower on a hot day, and on the other hand, a cold ambient temperature might result in an excessive power output that could cause over-load of the gear system of the engine. The variation of the maximum power with ambient conditions is clearly of prime importance to the engine users. Take a naval warship for example, its power system may operate between ambient temperatures of $-40\text{ }^{\circ}\text{C}$ in the Arctic and $40\text{ }^{\circ}\text{C}$ in the tropics. U.S. Navy requires that naval gas turbines should be designed based on a $100\text{ }^{\circ}\text{F}$ ($37.8\text{ }^{\circ}\text{C}$) hot day.

The effects of ambient conditions can be studied from the off-design characteristic of the gas turbine. If it is assumed that a gas turbine operates at the rated mechanical shaft speed, a different ambient temperature would produce a different $N_1/\sqrt{T_1}$, and from the established relations of $N_1/\sqrt{T_1}$ vs. power, a correspondent engine performance can be found for a given ambient temperature.

The effects of ambient temperatures on WR-21 performance can be obtained using the curve in Figure 6.21. The results of such effects are plotted in Figure 6.22. The fixed PT nozzle assumption has little influence on the ambient condition calculations, because the purpose of this study is normally to find the effect of the ambient temperature on engine performance at the design point, where the PT nozzle is kept at rated value.

Fig.6.22 LP Rotor Speed vs Power Output

Fig.6.23 Effects of Ambient Temperature on Power

6.10 VALIDATING ANALYSIS OF OFF-DESIGN CALCULATIONS

In this section, the accuracy of the off-design calculations will be studied. It can be clearly seen in Table 6.1 that calculated equilibrium values of TIT, LPC and HPC pressure ratios, etc., at rated $N_1/\sqrt{T_1}$, differ slightly from the input data from the DP calculation. This indicates small errors are involved in the off-design calculations.

If it is assumed that the engine parameters from the design-point calculation are accurate, then, the investigation can be carried out in two areas: the reliability of generalized component characteristics, and the accuracy of the cycle calculations.

The component characteristics used in the cycle calculations are produced based on generalized compressor and turbine characteristics. As mentioned early in this chapter, these generalized component performance characteristics were normalized based on numerous early component tests. Although these diagrams were found very reliable in the previous uses, it is found, in nearly all cases, that there were some places that were incompatible with the actual engine model being studied. Any corrections can only be achieved based on extensive component tests, that is not realistic with the school resources. Certain simplifications associated with generalized turbine characteristic can be seen clearly by comparing the generalized curve (Figure 6.4) and the typical test curve (Figure 6.2). On the generalized turbine characteristic, the unchoked region is simplified by a single curve, rather than a narrow region as in the typical turbine characteristic, and furthermore, the variation of turbine efficiency for different speed settings is neglected. Choking of a turbine may occur in the nozzle throats or in the annulus at outlet from the turbine depending on the design, which would result in different choking pressure ratios. On the generalized turbine characteristic, a common choking pressure ratio of 1.8 is fitted for all cases, whilst the characteristic on Figure 6.2 indicate a choking pressure ratio of 2.5 for the particular turbine. Similarly, some contradictions have also been found on the generalized compressor characteristic, the gradient of the equal efficiency islands appear too steep for modern gas turbines of today.

Another source of errors come from the graphic iterations between the component characteristics. With thousands of iteration steps between the HPC and the LPC characteristics, computerized interpolation must be used. Since the constant speed lines, especially those

representing the higher speeds, feature by almost vertical straight lines, a very small step iteration has to be used. The stability of the computerized iteration will reduce when the step number selected too high.

The magnitude of these error can be studied by comparing the off-design performance at the design point, with the engine parameters from the design point calculation. The estimated errors in terms of calculated TIT, OPR, LPCR and HPCR are about 5 %.

6.11 EFFECTS OF VARIABLE POWER TURBINE NOZZLE

It should be noted that adjustment of the power turbine nozzle is actually a part of the engine overall control system. In the case of twin-spool ICR gas turbines, the variable power turbine nozzles are used on both the WR-21 and the proposed Spey ICR, while their original simple cycle versions are not featured with variable geometry. It is believed that such variable installations, together with the fuel control, perform an indispensable duty of determining the spool speeds and the pressure ratio split between the LP and HP compressors. However, each engine producer has developed its unique schedules of nozzle control and so far little information on the variable PT nozzles is published. The only known PT nozzle schedule is of the GE MS-5002B single-spool gas turbine, the power turbine nozzles in the MS-5002B are used to hold the rotor speed at 80% of the rated value over the 0-50% load range. It seems impossible to estimate the accurate nozzle arrangement of a real engine model, because the scope of possible nozzle arrangements is too broad, particularly in the case of the twin-spool engine. Only a conceptual investigation based on the WR-21 will be described.

As has been discussed in section 6.4, choking of the power turbine nozzle will place a major restriction on the operation of the turbines in the front. Controlling of the operation of the HP and LP turbines can be accomplished by changing the power turbine nozzle areas. The flow relations between the HPT, LPT and PT of the WR-21 has been illustrated in Figure 6.10

It can be seen that reducing the PT nozzle area moves the LP turbine pressure ratio to a lower level, and opening of the nozzle allows higher LPT pressure ratio. Since the LP turbine of the WR-21 is not choked, closing or opening of the PT nozzle will produce, via the LP turbine, a similar but more gentle influence on the HP turbine. Just like a typical single spool gas turbine,

2 of/de 2

MICROCOPY RESOLUTION TEST CHART
NATIONAL BUREAU OF STANDARDS
STANDARD REFERENCE MATERIAL 1010a
(ANSI and ISO TEST CHART No. 2)

the HPC running line should move towards surge as the PT nozzle area is reduced, see Figure 6.24. The calculated results of the HP rotor based on changed PT nozzle areas have proved above argument, as shown in Figure 6.25. The behaviour of the LPC operation also shows a similar move, the operating line moves towards surge when the PT nozzle area is reduced, see Figure 6.26. It is now necessary to assume a reasonable nozzle control schedule. The benchmark of nozzle arrangements is to obtain improved SFCs with better agreement with published values and lower the HP rotor speed in low power region. As investigated earlier in this chapter, the calculations based on fixed PT nozzle show a basically constant HP rotor speed over a wide range of part-load operation, this can normally be considered as a poor arrangement for engine life. Table 6.2 illustrates a series of assumed PT nozzle positions and corresponding engine performance. Some of the engine control parameters are plotted in comparison with the fixed nozzle calculation. Figure 6.27 and 6.28 show drops in the cycle temperatures as the PT nozzles close down. Figure 6.29 illustrates the fuel flow versus engine power. Improved SFC based on assumed PT nozzle areas is plotted in Figure 6.30, showing a better agreement with published SFC. Although the overall pressure ratios remain basically the same as those at fixed nozzle, the LP compressor pressure ratio increases and the LPC operating line moves to the surge, and the HP compressor pressure ratio decreases when reducing the PT nozzle, as shown in Figure 6.31 resulting in an improved LPC performance, and moving the compressor pressure ratio split towards the optimum ICR rotor split.

Table 6.2 Engine Performance At Assumed PT Nozzle Areas

PT Nozzle (%)	$N_{11}/\sqrt{T_1}$ (%)	$N_{11}/\sqrt{T_3}$ (%)	TIT (K)	Work (%)	SFC (lb/hphr)	LPCR	HPCR	PTIT (K)
102	106	97	1370	108	0.325	2.98	4.45	835
100	100	98	1366	98	0.3262	2.82	4.47	844
98	95	98.5	1378	85	0.3300	2.54	4.43	857
95	90	96	1310	65	0.3378	2.32	4.16	836
90	80	92	1212	39	0.3630	1.91	3.75	823

Fig.6.24 Typical Effect of Reducing PT Nozzle on HPC Running Line for Single-Spool Engine

At Sea Level, 100 F, 1 Bar
Operating at PT Nozzle Area 90-105%

Fig.6.25 HPC Running Line for Various PT Nozzle Areas of WR-21

Fig.6.26 LFC Running Line for Various PT Nozzle Areas of WR-21

Fig.6.27 Rotor Speed With With Variable Power Turbine Geometry

Fig.6.28 Power Turbine Inlet Temperature with Variable PT Nozzle

Fig.6.29 Fuel Flow at Part-Load Conditions

Fig.6.30 Specific Fuel Consumption

Fig.6.31 Engine Pressure Ratios at Part-Loads

Chapter Seven

Conclusions and Recommendations For Further Work

7.0 MAJOR FINDINGS

This section is intended to reiterate certain key points that can be distilled from the discussions of this thesis. The first question about marine ICR gas turbines might be whether or not the new generation of "marinized" ICR gas turbines would become an outstanding competitor as a naval base-load engine to replace the current mainstream diesels. Unfortunately, this topic can not be treated at the moment, because early experience with machinery of a new type is always prepared with failures that are due both to inadequacies in design attributable to the lack of that experience and also to the "early-failure" defects that are unfortunately inevitable even in proven machinery. However, it is clearly seen that the newly designed ICR systems, e.g., Spey ICR and WR-21, are adapted from highly reliable aero gas turbines and proven industrial regenerators to ensure low risk. It is certain that the chances of success of these installations can be expected to be much better than the early ICRs, such as the RM60 and the BTH ICR.

Some major findings in this thesis work are derived in the cycle analysis of the ICR gas turbines. These findings are briefed as follows:

- (1)Comparatively an intercooled regenerative gas turbine can offer a fuel consumption of 30 % better than that of a simple cycle engine, with an 10-15 % increase in power.
- (2)The present status of the modern twin-spool aero gas turbines provides a suitable compressor pressure ratio and spool layout for the re-design of ICR systems.
- (3)Variable power turbine nozzles play an important role in governing the engine shaft speed and maximum cycle temperature. Reducing the PT nozzle area will result in improvement in part-load fuel consumption.
- (4)In a twin-spool ICR gas turbine, fuel flow and power turbine nozzle arrangement are the major engine control parameters.

7.1 CONCLUSIONS

Having investigated, using the basic gas turbine theories, the unique characteristics of the ICR cycle, and especially, having matched the design-point and off-design point performance of the WR-21, some key conclusions can be derived as follows:

1. The cycle calculation model used in the ICR cycle analysis is correct. The examination of effect of varying component performance, e.g., the ϵ_{ic} and ϵ_{tt} , on the cycle performance is reliable.
2. The estimation of the WR-21 design-point performance is reliable. The calculated results match with published data of the said engine.
3. The method used in calculating the off-design performance of WR-21 is correct. The plots of these calculated results provide reasonable engine behaviours as the engine speed is reduced.
4. The involvement of variable geometry in the calculation results in a better agreement of fuel consumption with the published data, which indicate the method of accounting for the variable PT nozzle is correct.
5. There is still significant work required to develop some compressor and turbine characteristics with greater accuracy, and hopefully, a component characteristic data base, in order to make the computerized calculation possible.

7.2 RECOMMENDATIONS FOR FURTHER WORK

Saravanamuttoo and MacIsaac (1983) introduced the use of generalized compressor and turbine characteristics in the off-design calculations of gas turbines. Although this appears to be the only possible means to deal with the off-design problem when the detailed component information is not known, the method of generalizing the component performance does generate additional sources of error for the cycle calculations.

For example, in generalization of the turbine performance, the variation of efficiency for different speed settings is neglected, and the narrow band of curves, which appears at the low pressure ratio in a typical turbine characteristic, is simplified by a single curve. See the comparison of a typical turbine characteristic and generalized turbine characteristic. It is also seen

that the two turbine performance diagrams give a different choking pressure ratio, 1.8 from generalized turbine characteristic and 2.5 from the sample characteristic. Similar problems can also be found from generalized compressor characteristic, for example, the equal efficiency island appears to be steep for the modern compressors, thus, certain revision was needed in the application of the diagram. It is believed that the errors associated with generalized component characteristics would influence the accuracy of the off-design calculations. It is then desirable that some new component performance characteristics be produced based on some new information from engine manufacturers. Establishment of a data base of these generalized characteristics is also necessary for the computerized cycle analyses.

The principle of variable geometry in gas turbines has been explained by Saravanamuttoo (1987). However, the engines of each manufacturer have their unique means of putting into practice. Detailed information on the nozzle control are needed in order to examine the accuracy of the nozzle simulation in the cycle off-design calculation.

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APPENDIX A

Thermodynamic Data for Increments of 10 K
(System International Units)

Table 1

Temperature K	Enthalpy kJ/kg
300.0	300.23
310.0	310.28
320.0	320.32
330.0	330.38
340.0	340.44
350.0	350.51
360.0	360.59
370.0	370.68
380.0	380.78
390.0	390.89
400.0	401.02
410.0	411.15
420.0	421.30
430.0	431.47
440.0	441.65
450.0	451.84
460.0	462.05
470.0	472.28
480.0	482.52
490.0	492.79
500.0	503.07
510.0	513.37
520.0	523.69
530.0	534.03
540.0	544.39
550.0	554.77
560.0	565.18
570.0	575.60
580.0	586.05
590.0	596.52
600.0	607.0

APPENDIX A

Thermodynamic Data for Cycle Calculations

(System International Units)

Table 1: Air

Temperature K	Enthalpy kJ/kgK
300.0	300.23
310.0	310.28
320.0	320.32
330.0	330.38
340.0	340.44
350.0	350.51
360.0	360.59
370.0	370.68
380.0	380.78
390.0	390.89
400.0	401.02
410.0	411.15
420.0	421.30
430.0	431.47
440.0	441.65
450.0	451.84
460.0	462.05
470.0	472.28
480.0	482.52
490.0	492.79
500.0	503.07
510.0	513.37
520.0	523.69
530.0	534.03
540.0	544.39
550.0	554.77
560.0	565.18
570.0	575.60
580.0	586.05
590.0	596.52
600.0	607.0

Temperature	Enthalpy
K	kJ/kgK
650.0	659.83
660.0	670.47
670.0	681.13
680.0	691.81
690.0	702.52
700.0	713.26
710.0	724.02
720.0	734.80
730.0	745.61
740.0	756.44
750.0	767.30
760.0	778.18
770.0	789.09
780.0	800.02
790.0	810.07
800.0	821.94
810.0	832.94
820.0	843.96
830.0	855.01
840.0	866.07
850.0	877.16
860.0	888.28
870.0	899.41
880.0	910.57
890.0	921.75
900.0	932.95
910.0	944.17
920.0	955.41
930.0	966.68
940.0	977.96
950.0	989.26
1040.0	1046.08
1050.0	1103.37
1100.0	1161.10

Table 2:Gas

Temperature	Enthalpy	FCV
K	kJ/kgK	kJ/kgK
950.0	960.00	41100.0
1000.0	1046.08	41327.4
1010.0	1057.50	41296.9
1020.0	1068.94	41266.3
1030.0	1080.40	41235.6
1040.0	1091.88	41204.7
1050.0	1103.37	41173.7
1060.0	1114.88	41142.6
1070.0	1126.41	41111.4
1080.0	1137.96	41080.1
1090.0	1149.52	41048.6
1100.0	1161.10	41017.0
1110.0	1172.70	40985.3
1120.0	1184.32	40953.5
1130.0	1195.95	40921.6
1140.0	1207.59	40889.6
1150.0	1219.25	40857.4
1160.0	1230.93	40825.1
1170.0	1242.62	40792.7
1180.0	1254.33	40760.3
1190.0	1266.06	40727.6
1200.0	1277.79	40694.9
1210.0	1289.55	40662.1
1220.0	1301.31	40629.2
1230.0	1313.09	40596.1
1240.0	1324.89	40563.0
1250.0	1336.70	40529.8
1260.0	1348.52	40496.4
1270.0	1360.36	40462.9
1280.0	1372.21	40429.4
1290.0	1384.07	40395.7
1300.0	1395.95	40316.9
1310.0	1407.84	40328.1
1320.0	1419.74	40294.1
1330.0	1431.65	40260.1
1340.0	1443.58	40225.9
1350.0	1455.52	40191.7

Temperature	Enthalpy	ECV
K	kJ/kgK	kJ/kgK
1370.0	1479.43	40122.9
1380.0	1491.41	40088.4
1390.0	1503.39	40053.7
1400.0	1515.39	40019.0
1410.0	1527.40	39984.2
1420.0	1539.42	39949.3
1430.0	1551.45	39914.3
1440.0	1563.49	39879.2
1450.0	1575.55	39844.1
1460.0	1587.61	39808.8
1470.0	1599.69	39773.5
1480.0	1611.77	39738.1
1490.0	1623.86	39702.6
1500.0	1635.97	39667.0
1510.0	1648.08	39631.3
1520.0	1660.21	39595.6
1530.0	1672.34	39559.8
1540.0	1684.49	39523.9
1550.0	1696.64	39487.9
1560.0	1708.80	39451.8
1570.0	1720.98	39415.7
1580.0	1733.16	39379.5
1590.0	1745.35	39343.2
1600.0	1757.55	39306.8
1610.0	1769.76	39270.4
1620.0	1781.98	39233.9
1630.0	1794.20	39197.3
1640.0	1806.44	39160.7
1650.0	1818.68	39124.0
1660.0	1830.93	39087.2
1670.0	1843.19	39050.4

APPENDIX B

Data File for Generalized Turbine

Pressure Non-dimensional

Ratio	Flow
1.1000	0.6125
1.1125	0.6625
1.1250	0.6940
1.1400	0.7250
1.1500	0.7450
1.1750	0.7800
1.2000	0.8100
1.2250	0.8380
1.2500	0.8620
1.2750	0.8810
1.3000	0.9000
1.3250	0.9150
1.3750	0.9400
1.4000	0.9500
1.4500	0.9650
1.5000	0.9800
1.5500	0.9850
1.6000	0.9950
1.7000	1.0000
1.8000	1.0000
2.0000	1.0000
3.0000	1.0000
4.0000	1.0000
5.0000	1.0000
6.0000	1.0000
7.0000	1.0000

APPENDIX C

In the cycle calculations, data interpolation is widely used for either thermodynamic air property or data acquisition from generalized component characteristics. A interpolation subroutine is written based on three nearest points of any input value. The subroutine is also applicable for extrapolation. The subroutine is quoted as follows

```

      SUBROUTINE LAQ(N,L,X,A,B,Y)
      DIMENSION A(N),B(N,L),Y(L)
      N1=N-2
      DO 1I=1,N1
      IF(X.LE.A(I+1))GO TO 2
1  CONTINUE
      I=N1-2
2  IF(I.EQ.1)GO TO 3
      IF(X-A(I).GE.A(I+1)-X)GO TO 3
      I=I-1
3  A1=A(I)
      A2=A(I+1)
      A3=A(I+2)
      U=(X-A2)*(X-A3)/((A1-A2)*(A1-A3))
      V=(X-A1)*(X-A3)/((A2-A1)*(A2-A3))
      W=(X-A1)*(X-A2)/((A3-A1)*(A3-A2))
      DO 4J=1,L
4  Y(J)=U*B(I,J)+V*B(I+1,J)+W*B(I+2,J)
      RETURN
      END
```

where A are known values of Xs, B are known values of Ys, N indicates the number of available data group, L illustrates the number of Y axis(Y1,Y2...), X is the input interpolating value, and Y are the output interpolated results.

END

0 9-1 1-9 4

FIN