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Combined Gas/Steam Cycle Power Generation
from a User's Viewpoint

Supervisor: R. Singh

May 1990

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for the degree of Master of Philosophy

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ABSTRACT

Gas turbines have been used since the 1940's to provide electric power at times of peak demand, but this is due to low initial cost and short starting times. Historically low efficiency and a higher quality fuel requirement has kept the gas turbine on the high end of the load curve. The combined cycle uses a gas turbine as the prime engine, then uses an exhaust heat recovery boiler to raise steam for a steam turbine cycle. High efficiencies are achievable and the gas turbines can now burn fuels from natural gas through to crude oil and coal via gasification. High efficiency, short construction time, competitive installation costs and smaller plant power ratings, has led to increasing and extensive use of the combined cycle. During the next two decades large numbers of gas turbines will be purchased for all load curve requirements with combined cycles in the mid to base load range.

Gas turbines and combined cycles are studied in a holistic way, from performance theory to issues of procurement and maintenance. Gas turbines are discussed more fully than steam turbines as they are a less mature technology and dictate the operation of the combined cycle. A mathematical model of a hypothetical engine is used to find performance trends. This model includes the gas turbine, single pressure boiler and steam turbine. Off design of the gas turbine is studied using the Cranfield program Turbomatch. A mathematical model of the off design of the boiler is developed to give off design trends for the combined cycle. Other issues discussed are applications, fuels, rating, operation, maintenance and various configurations. Past development, current state-of-the-art and future prospects are also discussed.

AUTHORS COVERING NOTE

The author is employed by an electricity and gas utility in New Zealand to study combined cycle power generation. This thesis is for the qualification of Master of Philosophy, which is a 12 month course assessed 100% on this thesis. This is unlike the 12 month Master of Science course which is assessed 50% on thesis and 50% on exams. This thesis is fuller and provides some background not expected in a standard Masters thesis.

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Notation and AbbreviationsAbbreviations

ABB	A.S.E.A. Brown Boveri
AC	Alternating Current
AFB	Atmospheric Fluidized Bed
ANSI	American National Standards Institute
API	American Petroleum Institute
ASTM	American Society for Testing and Materials
CC	Combined Cycle
CHP	Combined Heat and Power
CV	Calorific Value
DOD	Domestic Object Damage
DOE	Department Of Energy
EB/PVD	Electron Beam Physical Vapour Deposition
EPRI	Electric Power Research Institute
FAR	Fuel Air Ratio
FBC	Fluidised Bed Combustion
FOD	Foreign Object Damage
GE	General Electric
GT	Gas Turbine
HCV	Higher Calorific Value
HEX	Heat EXchanger
HIP	Hot Isostatic Pressing
HP	High Pressure
HRSG	Heat Recovery Steam Generator
HTGR	High Temperature Gas cooled Reactor
LCV	Lower Calorific Value
IGCC	Integrated coal Gasification Combined Cycle
IGV	Inlet Guide Vane
IP	Intermediate Pressure
ISO	International Standards Organisation
KE	Kinetic Energy
LP	Low Pressure
MHD	MagnetoHydroDynamics
MTBF	Mean Time Between Failure
NDT	Non-Destructive Testing
NGV	Nozzle Guide Vane
NTU	Number of Transfer Units
O & M	Operation and Maintenance
OEM	Original Equipment Manufacturer
PC/FGD	Pulverised Coal and Flue Gas Desulphurization
PF	Pressure Factor
PFBC	Pressurised Fluidised Bed Combustion
RAM	Reliability, Availability and Maintainability
SFC	Specific Fuel Consumption
ST	Steam Turbine
TET	Turbine Entry Temperature
TIT	Turbine Inlet Temperature (same as TET)
TX	Time eXpired
UHC	Unburnt HydroCarbons
WG	Water Gauge

Notation

'	Ideal
A	Availability
C _p	Heat capacity at constant pressure

Notation

C_v	Heat capacity at constant volume
C	A constant
E	Ratio of cold to hot stream heat capacities in HEX
h	Heat transfer coefficient or specific enthalpy
H	Total enthalpy
i	Fraction of area
k	Ratio of specific heats C_p/C_v (also known as γ)
K	Number
m	Constant
$\dot{m}$	Mass flow rate
n	Constant
p	Number of poles
N	Rotational speed or Number
Nu	Nusselt number
p	Pressure
P	Total pressure or Probability
Pr	Prandtl number
Q	Heat flow or Volume Flow
R	Gas Constant
Re	Reynolds number
r_p	Pressure ratio
r_T	Temperature ratio
s	Specific entropy
S	Total entropy and part load efficiency factor
t	Static Temperature
T	Total temperature
v	Specific volume
V	Velocity
W	Work or rate of work, i.e. power
x	Fraction, e.g. steam quality or e.g. x_{p12} pressure loss from 1 to 2
z	Fouling factor
β	Fluid property function used in HRSG calculations
Γ	Geometric function used in HRSG calculations
ϵ	Effectiveness (Heat exchangers)
θ	Gas temperature, used to differentiate gas from steam in HRSG
η	Efficiency
μ	Ratio of steam to exhaust gas mass flows
ϕ	Phase angle

Subscripts

ABS	Absolute	HR	Heat Recovery
AV	Average	i	Isentropic or summation index
B	Boiler or Blade	k	Index
BC	Bottoming Cycle	$L1 \text{ \& } L2$	Low
CC	Combined cycle	MAX	Maximum
E	Electrical	MIN	Minimum
f	Saturated liquid	N	Nozzle
fg	From saturated liquid to saturated vapour	P	Process or Polytropic
F	Fuel	$PINCH$	Pinch point
FL	Full load	S	Steam
g	Saturated vapour	ST	Steam Turbine
G	Gas	TH	Thermal
GT	Gas Turbine	TOT	Total
$H1 \text{ \& } H2$	High	W	Water
		$*$	Minimum load

Glossary of Terms

Aero derived;	A ground based gas turbine engine derived from an aircraft engine.
Availability;	Also known as 'potential load factor'. There are various definitions used but generally the ratio of the time the unit is available to be used to the total time, or; $\text{Availability} = \frac{(\text{hours in period}) - (\text{all downtime hours})}{(\text{hours in period})}$
Base-load;	The power production level that is required continuously, thus it is the minimum power level on the load profile. A base-load power plant is required to operate nearly continuously. The definition of a base-load power plant is formalised by ISO as 6000 to 8760 hr/yr.
Black start;	Starting without external power supply, such as during a power blackout.
Bleed;	Compressed working fluid that is bleed from the compressor such as cooling air for hot components or for auxiliary purposes.
Bottoming;	A bottoming cycle is a process that extracts useful heat from the rejected heat of a higher temperature cycle. The steam turbine cycle is the bottoming cycle to the gas turbine in a combined cycle.
Brayton Cycle;	See Joule-Brayton
Buckets;	Another term for the turbine rotor blades.
Calorific value;	The heat released per unit mass of fuel burnt, for example expressed as kJ/kg. Generally standardized as from 15 °C, and 101.3 kPa. The two main types are; <i>HCV - Higher Calorific Value</i> , including the latent heat of vaporisation of the water formed during combustion, thus products are cooled. <i>LCV - Lower Calorific Value</i> , excluding the water latent heat.
Closed cycle;	A cycle having a recirculation of working fluid independent of the atmosphere.
Compressor;	The component that increases the (total) pressure of the working fluid either; <i>Axial</i> - Flow is essentially parallel to the shaft axis <i>Radial</i> - Flow is essentially radial from the shaft (Centrifugal type).
Core Engine;	The gas turbine engine, compressor/combustor/turbine within the gas generator or driving the fan in a turbo-fan engine.
Cycle;	A sequence of processes performed by an engine on some working mass. In a normal heat engine the sequence of thermodynamic processes are performed on a fluid to convert heat into work.
Dead band;	The range through which a controller input can be varied with no resulting measurable corrective action by the controller.
Deaerator;	A device to remove the air and carbon dioxide in the water of a steam cycle.
Economiser;	A heat exchanger used to heat the feedwater of a steam generator to boiling point
Effectiveness (ε);	The fraction of available temperature difference that is recovered in a heat exchanger (regenerator or recuperator), or; $\text{Effectiveness } \epsilon = \frac{(\text{Fluid temperature rise})}{(\text{Available temperature difference})}$
Engine health monitoring;	The technique of using measured variables of an engine to directly or indirectly obtain component condition.
Exhaust heat recovery;	Extracting heat from the hot working fluid as it leaves the engine.
Fault tree;	A logic path used to establish the engine component or subsystem causing a deviation from expected performance.
Free turbine;	Another term for power turbine (see power turbine).
Fuel Cell;	A direct conversion process from fuel to electricity, similar to a battery but a fuel and oxidiser are used rather than consuming the electrodes.
Gas generator;	A gas turbine that produces hot pressurized gasses with no mechanical output.
Gas turbine (GT);	A rotary engine which uses a gaseous working fluid in a cycle which approximates the Joule-Brayton cycle.
Governor;	A control system for control of parameters such as speed, temperature or power.
Heat engine;	A machine to convert heat energy (temperature) to mechanical energy.

Heat rate;	The amount of fuel energy consumed to produce a unit of net output energy, for example an electrical plant in kJ/kWhr. It is usually based on the lower heating value of the fuel as if burnt at 15 °C.
High Temperature Corrosion;	Corrosion in the hot section of the gas turbine due to melting of the combustion ash products.
Hot Section;	The components subjected to the hot gas such as the combustor and turbine.
Hot Start;	A gas turbine start where an excessive amount of fuel is needed and the engine is abnormally overheated.
HRS6;	The Heat Recovery Steam Generator is the boiler that transfers exhaust waste heat into steam heat.
Hung Start;	A gas turbine start where the engine fails to accelerate or takes an excessive amount of time.
Intercooler;	A heat exchanger located between two compressors to reduce the inlet temperature into and hence the power required to drive the second compressor.
IRAN;	A maintenance term for Inspect and Repair As Necessary. It is synonymous with on-condition maintenance as opposed to hard time maintenance
ISO conditions;	The standard ambient operating conditions used for comparison of performance: Total Temperature = 15.000 °C Total Pressure = 101.325 kPa Humidity = 60 %
ISO rating;	A standardised set of terms for categorising the hours per year and number of starts per year that an engine is subjected to, e.g. ISO peak-load.
Joule-Brayton cycle	A thermodynamic cycle consisting of an isentropic compression followed by an isobaric heating, an isentropic expansion and isobaric cooling.
Light off;	The point of successful ignition of the gas turbine combustor.
Light round;	The successful transfer of ignition from the igniters's combustor to the others via the connecting ignition tubes.
Magnetohydrodynamics (MHD);	A generator with no moving parts using the principle of passing a hot ionised (conducting) gas through a magnetic field to induce an electric current.
Multi-spool gas turbine;	A gas turbine with mechanically independent shafts each with their own compressor/turbine pair.
On condition monitoring;	A term used for a maintenance scheme based on condition of the engine as opposed to a scheduled fixed time-based maintenance.
Open cycle;	A cycle in which the working fluid enters from the atmosphere and discharges back to the atmosphere, without recirculation.
Overhaul;	A term synonymous with complete disassembly, cleaning, testing, repair and replacement of components as required. 'Overhaul' is virtually extinct as on-condition maintenance is used.
Pattern Factor;	A measure of the uniformity of the temperature profile of gases leaving the combustor.
Peaking;	A term used in electricity generation referring to generation of the additional power needed during short periods of high demand.
Plant efficiency;	The efficiency of an entire electrical plant from fuel calorific value input to electrical output.
Power (free) turbine;	A turbine mounted on a separate shaft from the compressor, from which power is extracted. Usually associated with a gas generator.
Precooler;	A heat exchanger or evaporative device which cools the working fluid prior to initial compression.
Rated power;	Two main types: <i>Normal rated power;</i> Power of the engine at ISO conditions. <i>Site rate power;</i> Power of the engine at specified 'site' conditions.
Recuperator;	See regenerative cycle.
Regenerative cycle;	A cycle which uses the exhaust gas temperature to preheat the compressor discharge prior to the combustor. Using either of two types of heat exchanger; <i>Recuperator,</i> uses thin conductive walls. <i>Regenerator,</i> uses a matrix alternately exposed to the hot and cold gas streams.

- Reheater;** A second or subsequent combustor located between turbine stages to increase the power output and/or efficiency of the engine.
- Reliability;** Two main types are used, starting reliability and operating reliability. There are variations in definitions between users, especially to create some commercial advantage. Starting reliability is generally the ratio of successful starts to attempted starts. Operating reliability is generally the ratio of the time the unit is not down due to forced outage to the total time, or;

$$\text{Reliability} = \frac{(\text{hours in period}) - (\text{unplanned outage hours})}{(\text{hours in period})}$$
- Repowering;** Replacing a steam turbine's boiler heat input completely or in part by a gas turbines exhaust. This creates a form of combined cycle.
- Reverse Engineering;** Duplication of parts without the original engine manufacturer's drawings.
- Rogue Components;** Components that are failing at a much higher frequency than expected or recurrent problems usually due to design deficiencies.
- Simple cycle;** A cycle consisting of compressor, combustor and turbine only, without exchangers, reheaters or exhaust heat recovery.
- Single shaft cycle;** A gas turbine with all rotary components, compressor and turbine on the same shaft. This is opposed to a gas generator and a mechanically free power turbine. In a combined cycle sense this means the steam turbine is also on the same shaft.
- Sour Gas;** Fuel gas that contains sulphur, such as hydrogen sulphide H_2S .
- Specific fuel consumption;** Quantity of fuel used per unit work or thrust output, usually using the lower calorific value. It is expressed in kg/kWhr or in aircraft terms kg/kNhr.
- Specific Work;** The power output per unit flow of working fluid. This is a measure of the engine size for a required output.
- Stall;** A compressor condition resulting from an excessive angle of attack of the blades and flow separation, resulting in a loss of efficiency and unwanted vibration.
- Steam (or water) injection;** Steam and/or water injected into the working fluid to increase power and efficiency, or to reduce the formation of nitrogen oxides.
- Supplementary firing;** Firing of more fuel in the gas turbine exhaust to increase the temperature or power of the gases entering the HRSG.
- Surge;** An unstable and often very damaging compressor condition characterised by violent fluctuations in compressor flow.
- Surge Margin;** The latitude between the engine operating point and surge. It is usually measured as the ratio of the pressure ratio to the pressure ratio required at that same mass flow to cause surge.
- Thermal efficiency;** The ratio of the net power output of the engine to the fuel heat consumption, usually based on lower calorific value of the fuel.
- Turbine;** When the term is used on its own then it refers to the turbine component. This is the component in which energy is extracted from the pressurized working fluid and converted to mechanical energy by a rotating blades.
 Axial - Flow is essentially parallel to the shaft.
 Radial - Flow is essentially radial into the shaft.
- Turbine entry (inlet) temperature (TET);** The mean temperature of the working fluid immediately upstream of the first stator vanes of the turbine.
- Turbomachinery;** Engines and machinery that use rotating elements for processes, such as rotary compressors and turbines.
- Twin shaft gas turbine;** A gas turbine containing two separate turbines on separate shafts. Usually one with a gas generator and one with a power turbine creating export power.
- Unit Exchange;** Exchange of an engine, engine module or accessory with an overhauled one. The component is a different one than the customers one.
- Utility;** A company which distributes and/or generates electricity.
- Working fluid;** The fluid, whether liquid or gas, which is used by the engine in its thermodynamic cycle.

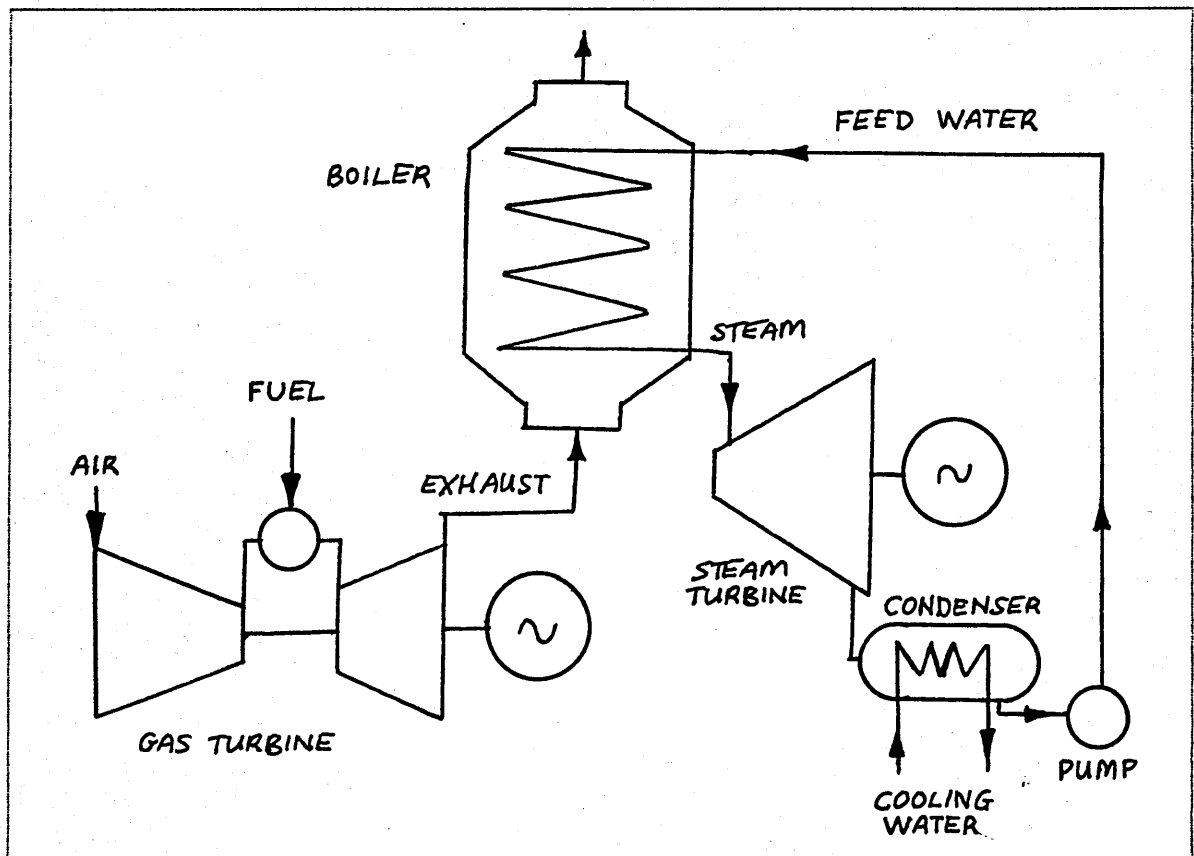
1 INTRODUCTION

The aim of every electricity producer is to generate the lowest cost power at an acceptable level of reliability. A combined cycle consists of a gas turbine exhausting into a waste heat boiler, which in turn is used to power a steam turbine. Figure 1.1 has a schematic diagram. This thesis is a study of the combined cycle as used in power generation.

A communities electrical load during a day varies from some maximum to a minimum. The minimum load represents the power level that is generated all the time and is called base-load. The power capacity required for only short periods is called peak-load, derived from the shape of the plotted daily load profile. Several technologies are available for meeting the power generation requirements, but the majority of larger utility plants are based on a turbine. The main turbines used are steam turbines (ST), gas turbines (GT) and combined cycles (CC) using both steam and gas turbines. Section 2 gives a description of power production and plant considerations. Available plant types are also introduced but only the three main plants of gas turbine, steam turbine and combined cycle are compared in detail.

The gas turbine technology is still rapidly developing and relatively recent. Gas turbines also represents the largest part of a combined cycle power output and dictates the cycle performance. For these reasons the study has a significant proportion dedicated to the gas turbine. The steam turbine on the other hand is a mature technology and has only a brief covering.

Figure 1.1 The Combined Cycle



The turbine is a heat engine and as such operates under the temperature limitations of the second law of thermodynamics. Vigorous effort has and is being put into increasing the maximum temperature, or turbine entry temperature (TET). The performance of the turbine is also affected by the pressure ratio, and these performance issues are studied in depth for gas turbines in section 3.

In the early days of electricity the steam turbine dominated. The steam turbine as we know it now is usually credited to C.A. Parsons for his 1884 turbine, although steam power had been used and patented in various forms over the previous years. The first commercial power generation was in 1890, thus there has been 100 years of steam power generation. Early pioneers included, de Laval who in 1889 patented his convergent-divergent turbine, and Curtis who in 1896 patented the velocity compounded turbine. Rateau introduced his impulse turbine in 1894-96. Parsons lead the steam turbine world in the early days and licensed his turbine to companies such as BBC and Westinghouse. Steam turbines have been used in various applications but in particular dominated electricity generation. A deeper look at steam turbine background can be found in section 6.

The gas turbine has been used for power since the 1940's. The principle has been known for many years and the first patent of a gas turbine is attributed to John Barber in 1791. This used a reciprocating compressor and water cooled turbine blades. The first working model only appeared in 1905 by Armengand and Lemale, with a pressure ratio of four and a TET of approximately 750 K. This gas turbine used an exhaust heat recovery boiler and steam injection for blade cooling and power boost, but only provided sufficient power to turn itself.

Practical industrial gas turbines have been operating since 1939. The design of gas turbines initially developed out of steam turbine technology, but since the early days it has followed it's own path of technology. The first utility generating gas turbine is a 4 MW plant installed in a cave at Neuchatel and has operated for 50 years (since 1939). Military aircraft engine gas turbines were operating by the end of the second world war and commercial aircraft engines started in 1953 with the Rolls Royce Dart in the Viscount. Heavy duty gas turbines started with BBC in 1930, GE in 1948, Westinghouse in 1949 and Ruston in 1950. The aero-derived industrial engines started in the late 1950's based on mainly fighter engines. The first generation of aero engines had pressure ratio's of about 12 and efficiency of 23-27%.

In the 1960's new lessons had been learned and a second generation of aero engines were produced with increased performance but most significantly addressing the reliability and maintenance issues. These engines had or attempted to have modular designs, borescope inspection holes, and used on-condition maintenance. Repair is by replacement of an entire module giving a quick change-out time and later workshop repair. The second generation aero engines had high efficiencies, which are currently up to 36%. Many of the current successful engines were designed around this period and are still sold. Examples are the LM2500 developed from the TF39 military aircraft engine and the RB211, both developed in the 1960's.

Early electric power gas turbines were peaking units. They were inexpensive and used only for short periods under demanding conditions.

Priorities in maintaining the peaking gas turbine were often minimal relative to the large base-load plants and the gas turbine gained a reputation as unreliable. Gas turbines were also inefficient, which was only a minor concern as fuel was cheap. In 1964 gas turbines accounted for only 0.3% of the US utility system (700 MW), but the capacity grew to 28000 MW in 1973 (7%), and over 50000 MW in 1986. As a comparison the UK had in 1971 1700 MW of aero-derivative gas turbines.

The expansion of gas turbines in the 60's was due in part to the great US North East power black outs of November 1965 and severe system problems in the UK in 1961. These problems pointed to the need for emergency and black start capability. The gas turbine can be started in 2-3 minutes at short notice and most gas turbines then were for peaking, emergency and black start purposes.

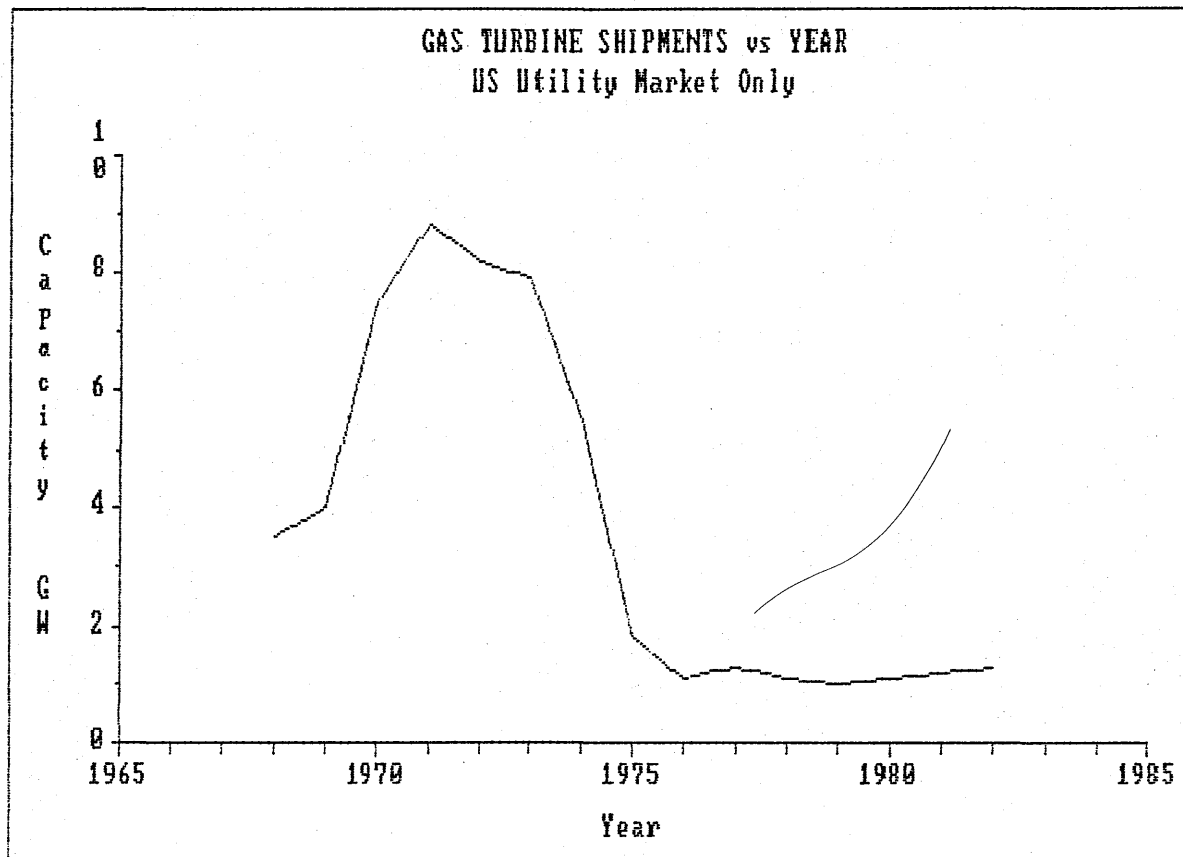
In the 1950's and 60's utility gas turbines were typically remodelled aircraft engines of 10-25 MW and an efficiency of 23-30%. The aero-derived engine was low cost, compact, and quick starting. Low efficiency can be propped up with complex cycles, but the gas turbine loses its inherent simplicity, compactness and low cost. By 1970 individual industrial GT capacity reached 50 MW, and by 1980 it was up to 90 MW at an efficiency of 32%. Currently a capacity of over 200 MW is available and efficiencies are claimed of up to 36%. The evolution cycle is 10-15 years, for both technical reasons and that utilities tend to favour established technology when spending large sums of money. Also small incremental improvements are required to be proven before additional improvements are carried out.

Repairs to gas turbine components are possible, starting in the 1960's but until the 1970's most "repairs" were by part replacement. The oil crisis in the mid 1970's changed the industrial repair industry and the use of component repair upsurged. It has attracted increasing attention over the years due to the high cost of spare parts and/or long lead times for ordering, although often discouraged by manufacturers as a loss in their profit. Modern repair techniques have improved and it is now possible to repair many components that were traditionally unrepairable. Typically the repaired item will cost 30 to 60% of a new item, and may have an equivalent life. Repair and maintenance are briefly discussed in section 5.

In the late 1970's fuel prices rose dramatically. The US Congress passed the Power Plant and Industrial Fuel Act of 1978, banning utilities after 1992 from installing any additional intermediate or base-load capacity fired with oil or natural gas (for more than 1500 hours a year). The rising fuel prices and this law caused the bottom to fall out of the gas turbine utility market. Sales to the US utility market fell from 9000 MW/yr in 1971 to only 1000 MW/yr from 1980 to 1982 (figure 1.2). The US market is the worlds largest electricity market. It is no exaggeration to say that the combined cycle sustained the utility gas turbine market in the early 1980's [179].

Currently fuel is cheap again and capital is expensive. Gas turbine improvements have pushed the combined cycle lower calorific value (LCV) efficiency to over 50% on the best plant. Also the technology for coal gasification is proven and can be added as a retrofit and future option. Gas turbines can be used alone, to repower old steam plant as a combined cycle or in cogeneration schemes.

Figure 1.2 Shipments of Gas Turbines vs Year



The 1987 amendment to the US Fuel Act of 1978 eased restrictions on gas turbine use. In the US there has been little new installed electric capacity since the oil shock (1978-80), and growth has largely been sustained by using up reserve capacity. Gas turbines sales are predicted to boom (see section 2 and reference [8]), with gas turbines and combined cycles very dominant in future electric power generation. Latest estimates of the US department of energy expect combined cycles to account for 50% of the growth in US power. Predictions are for 45 GW of new combined cycle capacity by the year 2000 [8]. EPRI believe these are conservative estimates. In the U.K. the CEEGB have announced plans for 4 GW of new combined cycles. Growth in world combined cycle capacity up until 1980 is shown in figure 1.3.

Combined cycles have not always been so successful. They date back to the 1960's but it was not until the advent of high temperature gas turbines in the 1970's that combined cycles became competitive. Early experiences were reasonable but not totally satisfactory, as they were very low efficiency and poor reliability. Early gas turbines had efficiencies of about 25% and gave combined cycle efficiencies of about 40%, which is less than the comparable steam cycle efficiencies. During the 1970's improvements to gas turbine efficiencies caused the combined cycle efficiency to increase dramatically, and most plant installed after this period have efficiencies of 40 to 50%. Even the best modern steam plants only achieve practical efficiencies of about 36 to 40%.

Reliability was a problem for early base-load plants. One of the earliest large plant, the 1974 Wharton 300 MW combined cycle station,

had an availability of 79% [180]. But the plant managed to maintain high availability when the demand was highest obtaining 91.6% in 1980 to 1981 when the demand factor was 92.7%. Subsequent plant have benefited from improvements. During the 1980's a Product Reliability Program was carried out by the Electric Power Research Institute (EPRI) aiming to raise availability of the US large gas turbines to 95%. Each cause of failure was addressed and recommendations made. The mean time between failure (MTBF) was improved over this period from about 350 hours to the current 28000 hour average claimed by GE [180]. This mainly involved developing controls and accessory reliability. Currently many engines operate for extended periods at close to the maximum availability, due to planned outages, of about 95% to 98%. Forced outage rates can be maintained within 2 to 4% [180].

One of the lessons learned during the efforts to improve reliability was that good reliability starts from the procurement stage. It is intuitively obvious that many problems can be solved or reduced by getting things right first time. Retrofits are very expensive and cost severely in terms of availability. The topic of procurement is too big for a thesis of this size, hence only a small amount is included in section 9. Here various references are listed.

The efficiency and reliability of combined cycles are now more than competitive with conventional steam turbine plants, but some of the main attractions to combined cycles are in flexibility and finance. Combined cycles offer smaller unit sizes, shorter delivery times and flexibility of staged construction. A simple cycle gas turbine can be installed and later made into a combined cycle. The smaller incremental capacity, lower construction site content and short lead times reduces debt and forward planning risks. These debt reducing features are very important to a utility.

In this thesis, as part of the study of the combined cycle, a mathematical model is developed. Firstly in section 3.4 the gas turbine design point is modelled. Section 6.2 develops a simple steam turbine model for a limited pressure and temperature range. Under the combined cycles section a model is developed for the heat recovery steam generator (HRSG) for both design point and off design performance. These

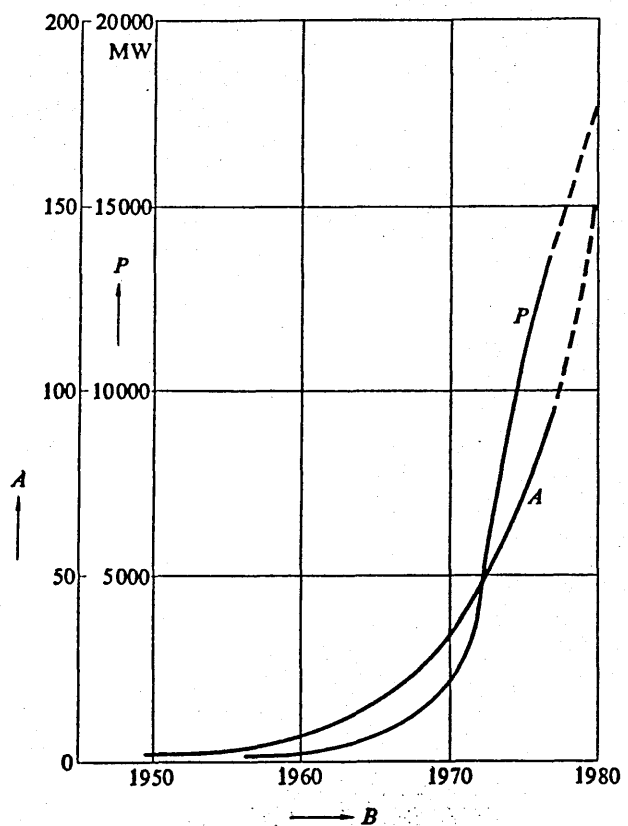
Figure 1.3 Combined Cycle Growth

Fig. 1 - Combined gas/steam power plants in the western world

A = Number of plants installed

B = Year of construction

P = Total rated output of the plants installed



various models are brought together to model a single pressure combined cycle in section 7.6. Using the Cranfield off design program for gas turbines the off design for the combined cycle is also included. Various configurations of both the gas turbine and steam cycle are presented and discussed.

The future is always of interest to utilities as predictions are necessary for planning. Section 8 investigates current technology and developments.

2 POWER GENERATION

2.1 ELECTRICAL PRODUCTION BACKGROUND

Electricity cannot be practically stored in bulk so must be generated on an as-demanded basis. This means the generating plant must follow the consumption pattern of the consumers as they draw power. Typical daily load profiles are given for a high load and a low load day in figure 2.1. These are for the authors utility company where the load is temperature sensitive.

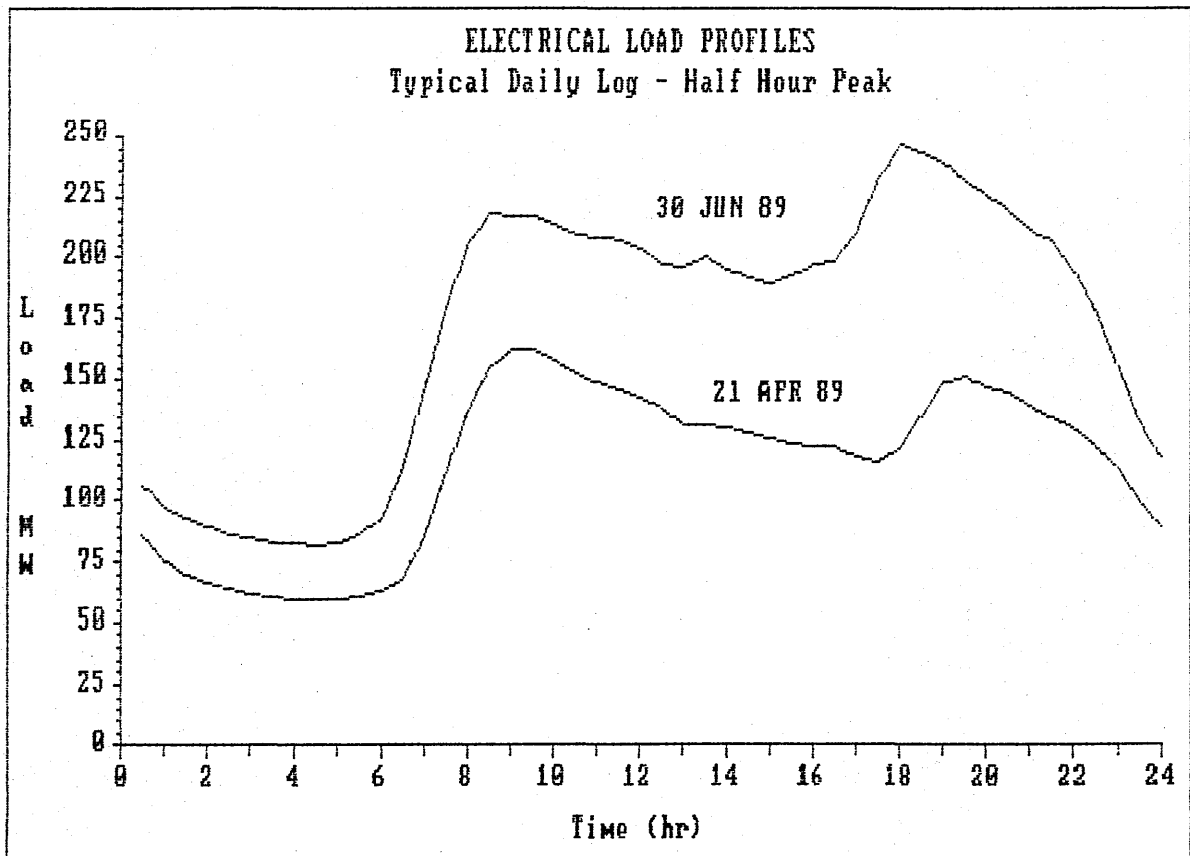
Electricity is also usually generated as alternating current or AC. This means there is a frequency, and a phase angle between the voltage and current. Any generating plant must be synchronised to the waveform and this means the engine is at a constant speed. A reduction in load tends to cause turbomachinery to speed up and this is compensated by reducing the fuel, steam or water in the turbines, or shedding generation plant from the grid.

As the generator must be synchronised there are only discrete speeds at which the generator can be designed to turn. The number of 'poles' in the generator can be used to reduce the speed, but the maximum speed is when the number of poles is at a minimum.

$$\text{Frequency} = pn \quad \text{where frequency is in Hz}$$

p is the number of poles
n is the revolutions per second

Figure 2.1 Typical Utility Daily Load Profile



Most turbines, except some very large types, require high speeds for best design optimisation thus two pole machines are often used. For 50 Hz this means a maximum speed of 3000 rpm and for 60 Hz 3600 rpm. 50 Hz is the standard frequency in New Zealand and Europe, while the North Americans use 60 Hz.

The power factor is the cosine of the phase angle θ between the voltage and the current. The phase difference comes as a consequence of the type of load. The voltamperes (kVA) is the product of the averaged (RMS) current and voltage but if the current is not in phase with the voltage then the actual power in Watts will be less. The losses are proportional to the square of the current and the power factor must be kept as high as possible. The practical importance of power factor is as follows. When an alternator is rated at say 2000 A at 400 V, these are the highest safe values and this alternator is then rated at 800 kVA. But if the power factor $\cos(\theta)$ is say 0.8 the generator can only produce $0.8 \times 800 = 640$ kW. Also the transmission conductors can only transmit this reduced load, and due to increased current the losses are 1.57 times those losses occurring when using a unity power factor. An unloaded but synchronised generator can be made capacitive or inductive depending on the excitation current hence can be used to improve the power factor. Gas turbines are suited to controlling power factor.

2.2 UTILITY OBJECTIVES

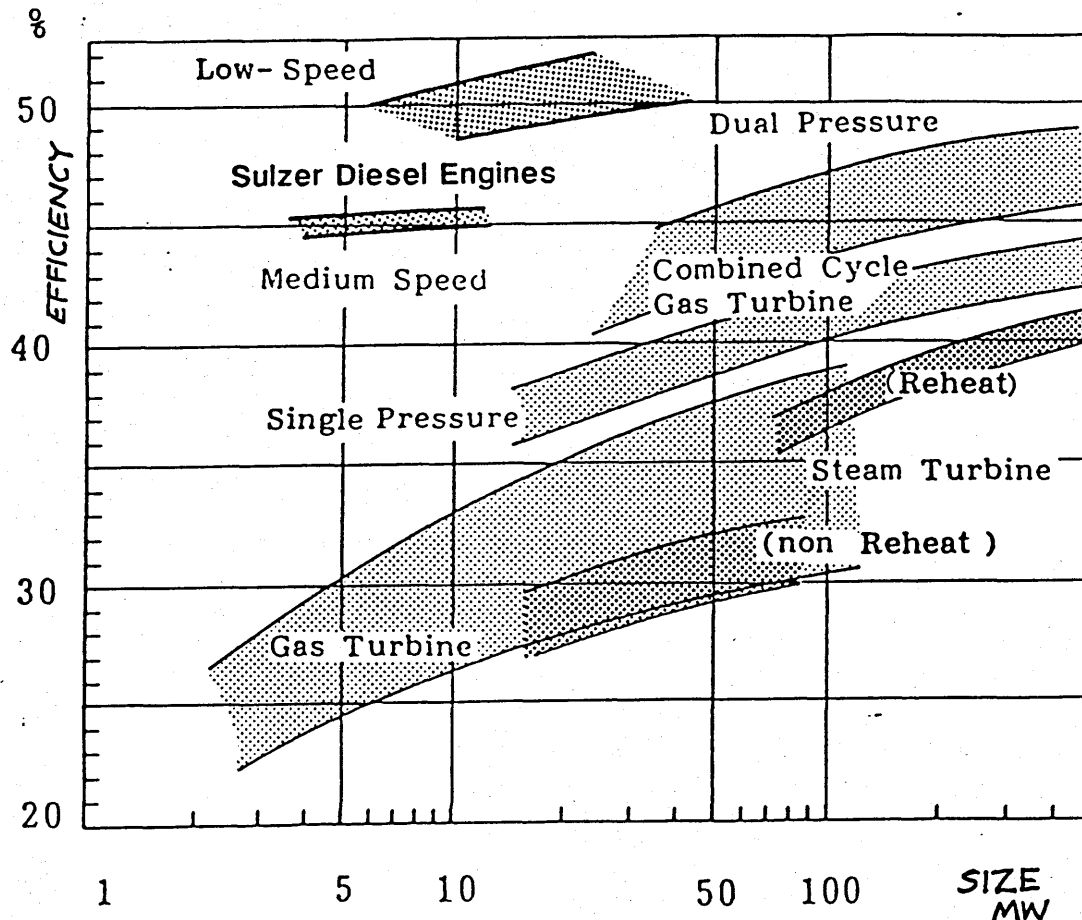
The aim of every electricity producer is *lowest cost power at an acceptable level of reliability*. The 'lowest cost' can be over a shorter or longer term depending on the commercial set up. Low cost requires;

- * low production costs (efficiency, fuel cost, maintenance cost)
- * low transmission costs (size, location, interconnection, reliability)
- * low administration costs (simple engines, flexibility)
- * good financial position (debt, capital, construction time, unit size)

Different engines are appropriate for different situations, as each has it's own clear balance of advantages in certain applications. The main current engine types for utility power generation are, Steam Turbines, Gas Turbines, Combined Cycles, and Diesel engines. The main fuels are natural gas, light and heavy distillates, residuals and coal. Hydroelectric power is not included as it is not a heat engine and is a special case. Nuclear is also not included due to it's special considerations. Figure 2.2 is an attempt by "Sulzer" to show the efficiency verses size of various alternatives. Sulzer is a manufacturer of diesels with a bias to show. The diagram does tend to show low combined cycle efficiencies relative to manufacturers claims but in reality the average efficiency obtained in an engine over it's life is somewhat reduced from the design point claims. The different engine types are advantageous for different load sizes, fuel types, load characteristics and interest rates. More comparison information follows later in this section.

Cost optimisation requires extensive knowledge of performance, materials, economics, manufacturing, people etc. It involves a trade off of initial cost or capital and ongoing cost related to performance and maintenance requirement. In many cases rigorous cost optimisation is not possible during the engine design stage as this would need cost functions such as capital cost to thermodynamic variables such as

Figure 2.2 Plant Comparison - Efficiency vs Size



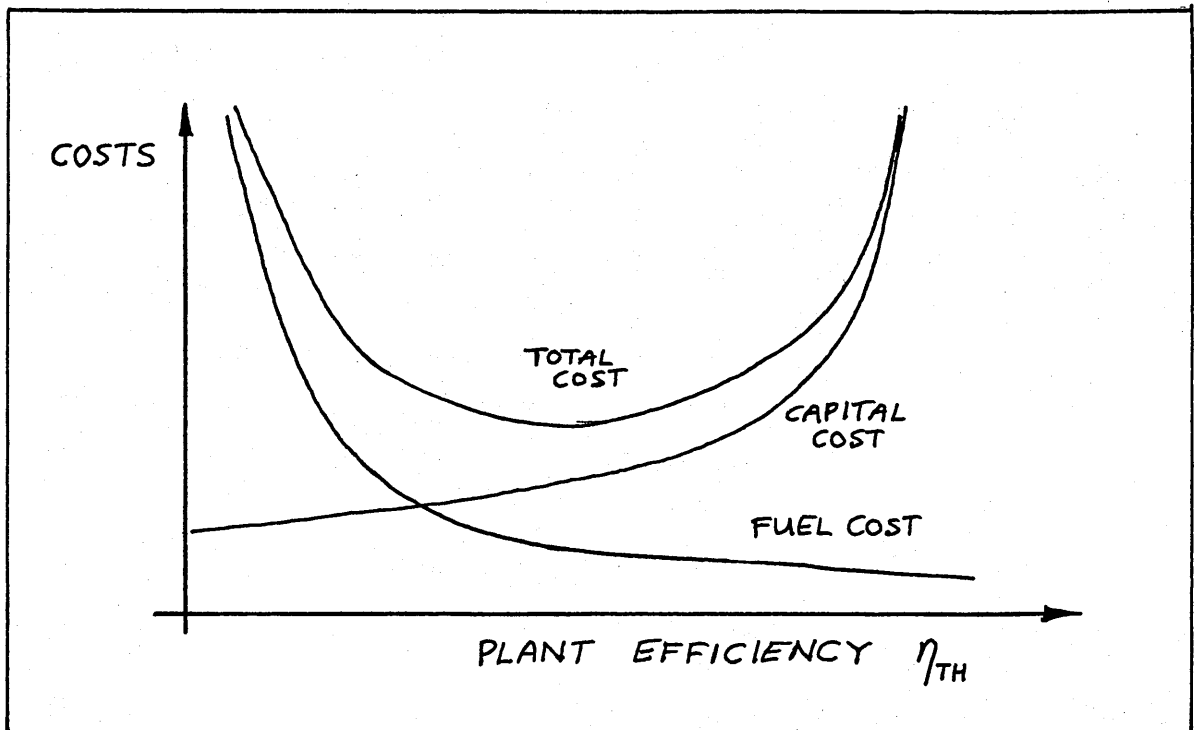
temperature, which is often unavailable. Engine optimisation is based, among other things, on the experience and intuition of the design engineer. For a purchaser the optimisation is based on available plant, and the conditions in which it will be used.

Inevitably the capital and maintenance costs increase with higher efficiency or some compromise is made with part load performance. The increase in costs would look something like the exponential curve in figure 2.3, increasing to infinity just less than some ultimate ideal maximum. A fuel cost curve is also plotted and a total cost given. This is an area of 'thermo-economics'. A major problem to users are that only a limited range of engines are available in certain quantised sizes and configurations, and there will be some uncertainty in usage patterns and fuel availability for the long term future.

2.3 SELECTION CONSIDERATIONS

Analysis must include many items, including intangibles which are attributes not directly quantifiable in the cost of electricity. Some items are listed as follows; cost of preparation and tendering (procurement), time for procurement, probability of not meeting specification, plant capital, plant operating and maintenance cost, electrical performance (power, voltage and frequency control), plant

Figure 2.3 Thermo-economics



thermal efficiency, capability for load change, part load thermal efficiency, forced outage rate, planned outage rate, spare parts availability, supplier support capability and long-term survivability, plant safety, dependence on other systems, adaptability to fuels, life, flexibility in siting, waste production/disposal, plant construction time, unit rating, by-products, heat rejection, environmental intrusion, availability of fuels and additives, stand-by power requirements, start-up power requirements, tax and legislation. Each item has a greater or lesser tangible component.

To give an indication of attributes, a sample of them are expanded in the following section. The discussion as follows is based largely from [10];

Electrical performance;

Inherently stable frequency control, thus would react to support the system in an emergency. Capable of supplying reactive kVA.

Capacity for load change;

A rapid rate of response to change, and short start up or shut down times.

Part load efficiency;

Maintaining thermal efficiency at reduced load. This can be quantified using the average slope of the efficiency verses load curve over some expected load range.

$$S = (1 - \eta^*/\eta_{FL}) / (1 - W^*/W_{FL})$$

where FL is at full load and * is at the minimum load, say 50%. This factor S could be considered good for $S < 0.2$ and poor if $S > 0.5$.

Minimum Load for Continuous Output;

Good if below about 20%, and poor if above about 60%.

Availability and Outage Rate;

This effects the required reserve capacity. A ratio of time not available due to unexpected failure to total time of less than about 5% is good and more than 10% is poor. A planned outage rate of less than 10% is good and more than 17% poor.

Plant Safety;

This consists of two components; property and persons. The analysis can be divided into the probability of an unsafe condition and the consequences of the unsafe condition. For instance probability of an unsafe condition is usually considered high if greater than 10^{-3} /yr and low if less than 10^{-6} /yr. The consequences of an unsafe condition are the number of fatalities or serious injuries. This gives an overall rating of probability of fatalities or serious injuries per year. This may be poor if greater than 10^{-2} fatalities or serious injuries per yr per 1000 MW.

Life;

Typically plant life is good if more than 30 years.

Flexibility of Site;

Limited by noise, appearance, size, heat rejection, effluent, fuel delivery, construction access and availability of qualified staff.

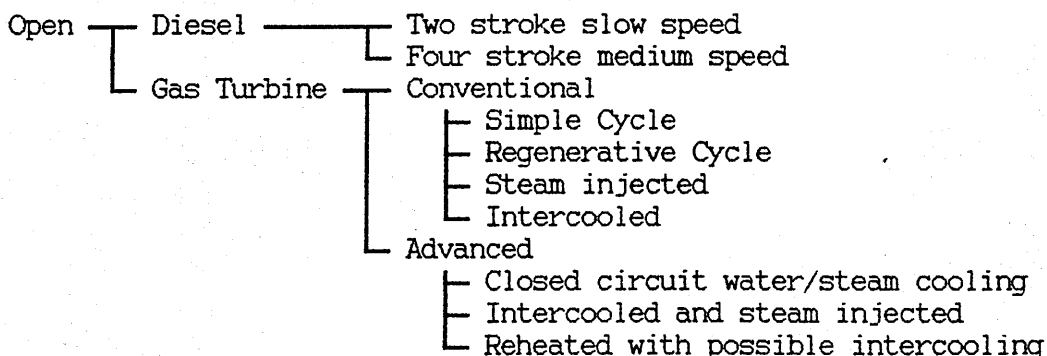
Construction time;

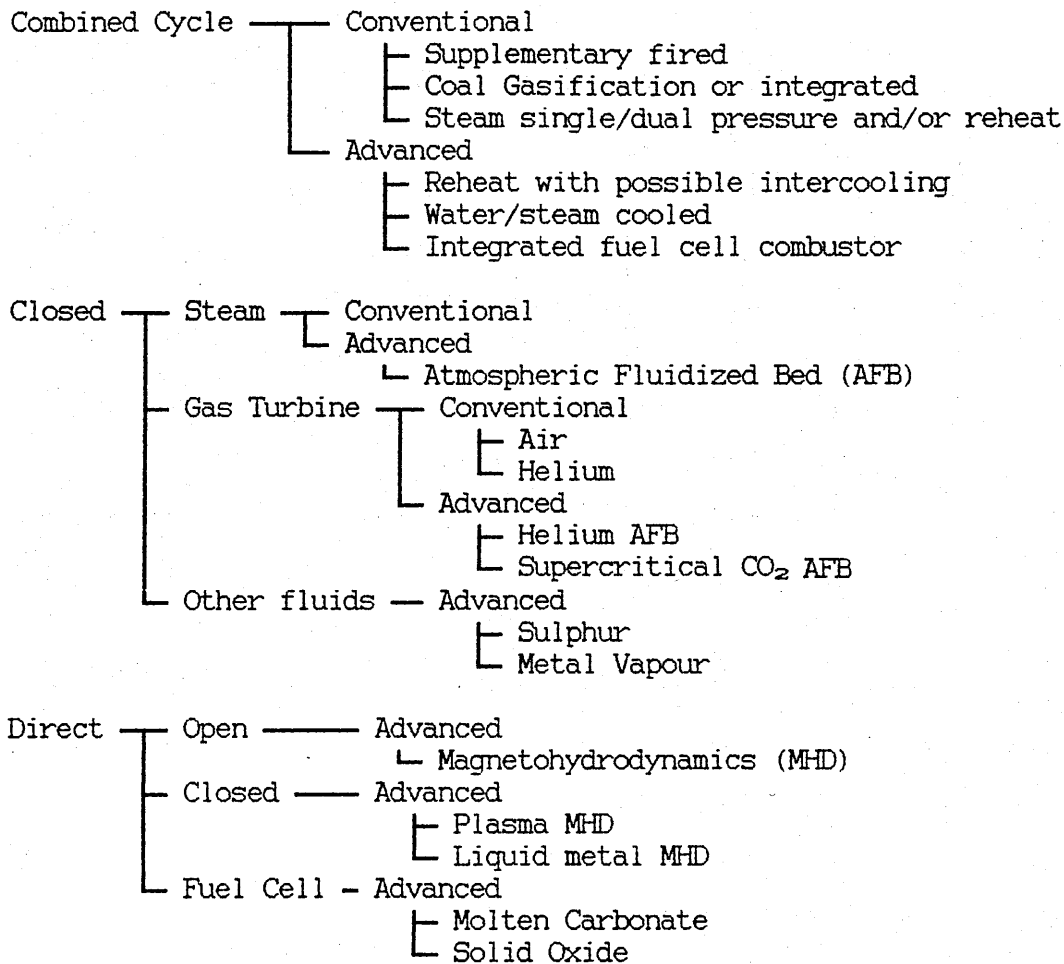
Time for procurement, delivery, erection and commissioning is good if less than 3 years and poor if more than 5 years.

The other considerations can be discussed in similar ways but a full discussion is unnecessary for the scope of this paper. An extended list is discussed in more detail in reference [10].

2.4 TYPES OF CYCLES

Many sources of energy can be used for power production, including hydroelectric, geothermal, ocean thermal, wave and wind power. But this study is confined to heat engines using a combustible fuel. The main heat engine cycles applicable to power generation currently or expected in the future are listed as follows;





In 1978 EPRI did a comprehensive and extensive study on cycles for the future [10]. This is dated but gives an idea of the predicted power producing plant 10 years ago. The study used a list of 26 tangible and intangible considerations similar to the ones listed above including research, development and manufacturing, but ignoring tax structures. It gave gradings and weightings for each so that a single figure of merit could be produced. The conclusions were for 1978 that for usage, greater than 1600 hr/yr a conventional steam while less than 1600 hr/yr a simple cycle GT. By the late 1980's the prediction was for less than 1500 hr/yr a simple cycle GT, and over 1500 hr/yr a combined cycles on coal gasification. In reality the high fuel prices of the oil shock did not last and fuel has been cheap. This has meant coal gasification has not as yet been successful commercially.

The USA is the largest user of electricity and for most of the 1980's has purchased little new capacity, using up reserve, which was 35% in 1985 and 25% in 1988 [8]. US growth predicted in the 1990's is large with a conservative estimate of 100 GW by the year 2000 [8]. Large numbers of combined cycles are planned currently in the USA and UK, and combined cycles on natural gas are predicted as the dominant new power plant of the next decade. The US DOE predict 63 GW of new US gas turbine capacity by the year 2000, with combined cycles making up 45 GW of this [8]. EPRI believe these are conservative estimates and with technology developments in electric vehicles coming on line this year (1990) and processes such as freeze concentration, even larger growth is predicted. The UK also has a 3-4 billion cubic meter per year of surplus natural

gas and with deregulation is expected to support much new plant. The CEEB have announced plans for 4 GW of new combined cycles. Natural gas has only about 40-60 years supply left and as this is used up coal is expected to be used again.

2.5 COMPARISONS BETWEEN PLANT

2.5.1 List of Plant Features

This section attempts to list some of the advantages and disadvantages of the main existing technology cycles relative to each other. It should be noted that the efficiency is not directly comparable as different configurations and fuels mean different efficiency in the same type of engine, and the use of cheap fuels such as coal allow much lower efficiency without being uncompetitive.

Gas Turbine

Advantages

- * No cooling water or cooling towers needed
- * Very small size and weight per unit output
- * Unit size small
- * Short starting times possible, particularly for aero-derivatives
- * Cost, very low in \$/kW installed
- * Prepackaged and often pretested units
- * Simple engine and minimal supervision
- * Delivery and construction time relatively short
- * Allows future change to combined cycle or cogeneration
- * Wide choice of fuels, gases and liquids
- * Low emissions possible
- * Low level of specialised operating and maintenance staff

Disadvantages

- * Low efficiency
- * Poor part load performance
- * More sensitive to fuel than steam turbine

Steam Turbine

Advantages

- * Efficiency reasonable for coal burning
- * Long life

Disadvantages

- * Slow starting times
- * Very long installation times
- * Large construction site content and little prefabrication/testing
- * Large sizes needed to be competitive
- * Cooling water or cooling towers needed (about 380 l/min per MW)
- * High cost in terms of \$/kW
- * Make-up water supply needed
- * Complex cycle
- * Large manning levels
- * Exhaust clean-up expensive, about 40% of capital

Combined Cycle

Advantages

- * Starting time medium, can have 2/3 of the power at GT start times

- * Cost, less than coal fired steam and more than simple GT
- * Often packaged units pretested and assembled on site
- * Unit size small yet still competitive
- * Delivery time of the GT can be short and run before ST delivered
- * Size of HRSG is smaller than equivalent ST boiler, say 1/8
- * Simpler than a ST to operate but more complex than GT
- * Uses 1/3 of the water than in a ST
- * Can be designed to add coal gasification in future
- * High efficiency
- * Flexible operation and possible staged construction

Disadvantages

- * Steam turbine power slow starting
- * Large manning levels, similar to ST

Diesel

Advantages

- * Quick starting
- * Good efficiency, over 50% in large sizes
- * Very good part load efficiency
- * Little water needed
- * Running costs small
- * High availability and reliability
- * Wide choice of incremental sizes by adding single cylinders
- * Short delivery time
- * Preassembled units

Disadvantages

- * Small overall unit sizes for power generation, up to about 40 MW.
- * Restricted in fuels, distillate, gas, and some use of heavy fuels
- * Very heavy in terms kg/kW, e.g. 7 - 60 kg/kW
- * High emissions

Notes;

Diesel: In large sizes the diesel engine tends to be slow speed (about 100 rpm) and two stroke. Medium sizes are often medium speed (about 600 rpm) and four stroke. Emergency generators may be high speed (750 to 1000 rpm) for compactness.

Conventional Steam: Conventional pulverized coal and flue gas desulphurization steam plant have efficiencies of about 36 to 38%. Improving the steam inlet temperature from 540-565 °C to supercritical plant of 620-650 °C with double reheat and adding additional feedheaters, the advanced steam cycle could reach efficiencies of 39-41 % [234] (1986). Conventional coal plant may include NO_x reduction, dust separation and desulphurization equipment, and are complex and expensive. The construction time is long and the unit size large so that more forward planning is needed early, increasing risk. The capital investment is also needed over the construction period increasing interest and debt, and increasing costs and risk. Construction may be over a period of up to 6 or 8 years and include significant on-site content. It is expected that development in technology will enable even better flue gas cleaning but the cost of cleanup will increase from about 38% of the capital to about 47%.

2.5.2 Comparison Discussion

For the purposes of this comparison discussion the engine types will be narrowed down to medium and large sizes, thus steam, gas turbine and combined cycle.

Construction time

Gas turbines are held as stock items by some manufacturers, and will have a short delivery relative to steam turbines. A gas turbine usually is factory assembled, delivered with the ancillaries as a number of prepackaged units and requires little in the way of water. This means it can be erected with a relatively small amount of on site labour and in a short period. Steam turbines on the other hand, especially in the very large sizes, require substantial on site construction work. The boiler may be substantially larger than an equivalent combined cycle boiler and not able to be factory assembled. The relative construction times are shown in figure 2.4 [12].

Figure 2.4 Construction Times

Construction times of various types of power plant

P = Installed capacity
 B = Construction time (months)
DT = Steam turbine plant
GT = Gas turbine plant
KA = Combined gas/steam turbine plant

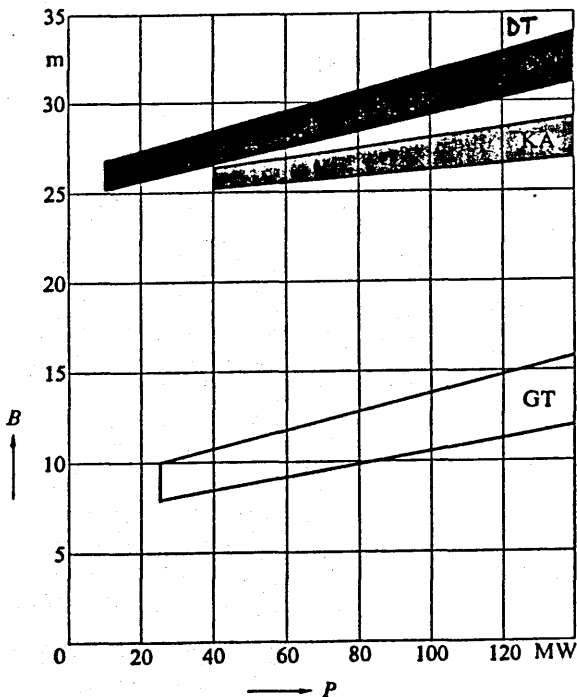
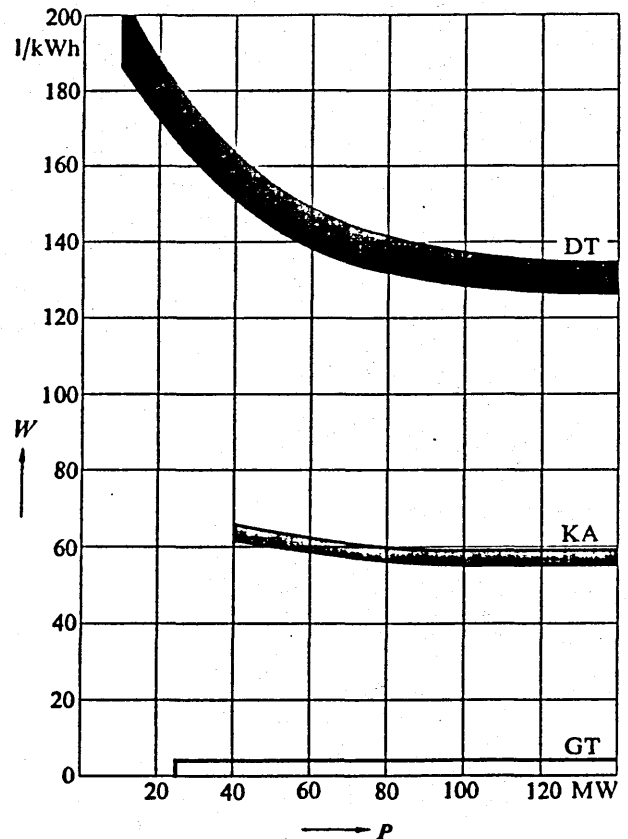


Figure 2.5 Cooling Water Requirements

Cooling water requirement W of power plants, with once-through cooling

P = Installed capacity
DT = Steam turbine plant
GT = Gas turbine plant
KA = Combined gas/steam turbine plant



Cooling water requirement

When considering steam Rankine cycles the problem of large scale heat rejection is important. Gas turbines have the advantage by needing practically no cooling water, and rejecting it's waste heat directly up the exhaust stack to atmosphere. The cooling of the generator can also be rejected to atmosphere. In steam plant cooling of the condenser for heat rejection can be by one of several means. It can be direct by river or sea water, by wet cooling towers, or if there is a shortage of water by air-cooled condensers at the cost of additional capital outlay. The relative cooling water requirements are shown in figure 2.5 [12].

Starting times

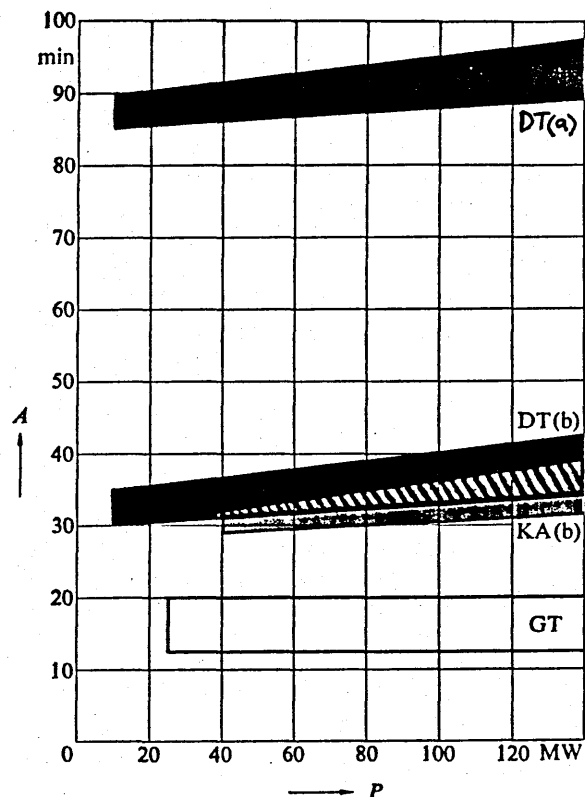
Aero-derivative gas turbines generally have the shortest starting times, in the order of two minutes. Heavy duty gas turbines may require times in the order of 20 minutes while steam plant may require 40 to 180 minutes depending on how long since last shut down.

Start up times of steam plant not only effect load control but also the losses incurred during the start-up. The magnitude of starting losses following an overnight shutdown for a natural circulation steam generator are about 1600 to 7500 MJ/MW. In contrast a heavy duty gas turbine may require only 800 MJ/MW, and an aero-derivative less than 80 MJ/MW. Combined cycles have the advantage that the gas turbine portion of the power can be available in gas turbine starting times, even if when the steam turbine power is available much later. Even the steam power portion of the combined cycle is quicker than an equivalent steam plant where the steam turbine and boiler are three times larger. Starting times are compared in figure 2.6 [12].

Capital Outlay

The costs of a power plant is a function of the capital, fuel, and usage. Figure 2.7 [12] is an attempt by one manufacturer to compare only the specific capital costs. These are based on a turnkey installation in Western Europe, not including subsidiary facilities such as offices and fuel storage. Interest on capital, depreciation, taxes and insurance have an impact on these capital costs as well but are site specific. As

Figure 2.6 Start-up Times



- Start-up times A of power plants
- P = Installed capacity
 - DT = Steam turbine plant
 - GT = Gas turbine plant
 - KA = Combined gas/steam turbine plant
 - (a) = 36-hour shut-down
 - (b) = 12-hour shut-down

Figure 2.7 Cost of Plants

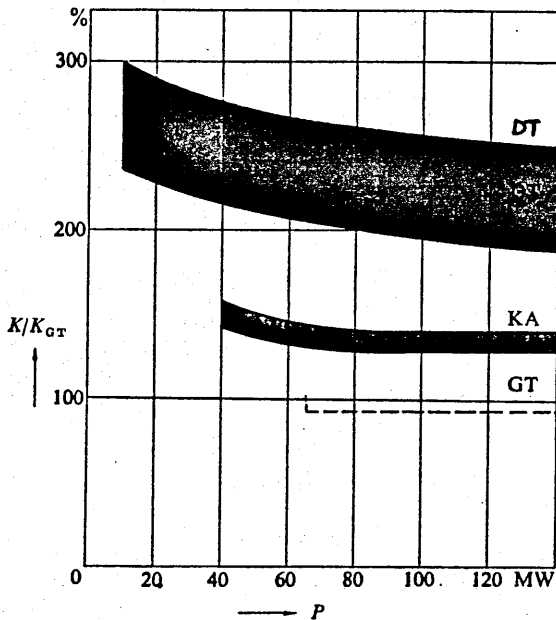
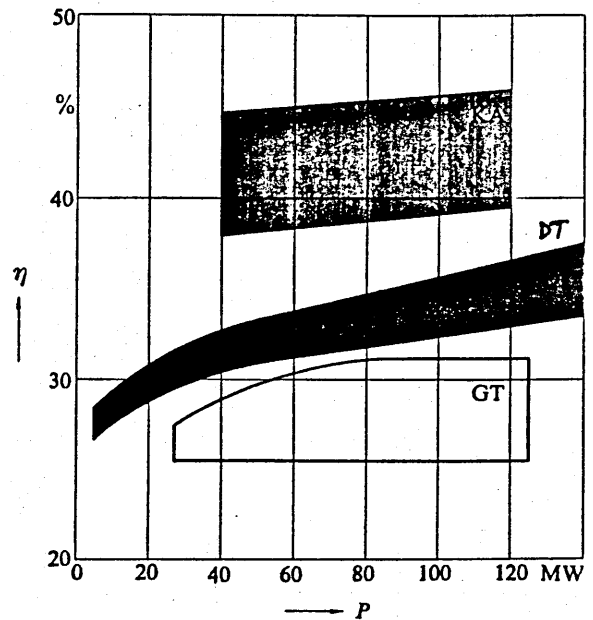


Figure 2.8 Efficiency of plants



Cost of power plants, referred to the cost of a gas turbine plant

P = Installed capacity
 K/K_{GT} = Referred cost of plant
 DT = Steam turbine plant
 GT = Gas turbine plant
 KA = Combined gas/steam turbine plant

Efficiency of power plants

P = Installed capacity
 η = Net efficiency
 DT = Steam turbine plant
 GT = Gas turbine plant
 KA = Combined gas/steam turbine plant

a comparison the cost of diesel plant is about \$US 833/kW for slow speed two stroke and \$US 600/kW for medium speed four stroke. A gas turbine ranges from about \$US 370/kW at 20 MW unit size to about \$155/kW at 200 MW unit size (engine cost only). Section 9.7 has more details.

Efficiency

The power station efficiency is important in calculation of the consumption of fuel, but consumption of a cheaper fuel may allow a lower efficiency to be competitive. Efficiency is also a measure of the consumption of resources and effects the need to reduce emissions like NO_x and CO_2 .

The highest efficiency of over 50% is from a combined cycle burning natural gas with a dual pressure boiler/turbine and direct fresh water cooling. A loss in efficiency of about 5% is expected when burning oil, using a single pressure boiler and air condenser. The lowest efficiency is from the simple cycle gas turbine at about 30 to 38% (on gas or liquid). Steam plant lie between these at about 36 to 38% (on coal) but require to be large sizes to compete. The comparison is shown in figure 2.8 [12], but it must be noted these figures are from 1987 and thus are conservative.

Part load efficiency in gas turbines and combined cycles is poor, $S_{50\%} \approx$

0.4 (definition of s section 2.3). It can be improved significantly in multi-engine installations where engines can be shut down one at a time. Steam turbine part load efficiency depends on the design. The design point efficiency can be compromised to improve the part load efficiency.

Operating costs

Operational costs are for items such as supervision, maintenance, consumables and in particular fuel. The discounted present value of the fuel for a base-load steam plant is about 2.5 times the capital outlay in Western Europe (1987 based on 8% interest rate and 30 year life) [12]. Fuel prices vary widely in different parts of the world, from practically zero upwards. Coupled with the large effects on costs from emissions, contaminants, storage and transport little generalised comment can be made. An important consideration is fuel flexibility but with coal gasification an established technology and fuel treatment possible heavy duty gas turbines have a wide range of possible fuels. Conventional boilers for a steam plant can fire low grade fuels without problems but gas turbines and hence combined cycles need more consideration. Fuels in gas turbines are discussed fully in section 4.

Other operating costs also include maintenance and personnel. Personnel requirements for supervision and maintenance are summarised approximately as follows for plant of medium range up to 100 MW [12];

Steam	0.38 - 0.50	persons/MW
Gas Turbine	0.22 - 0.31	persons/MW
Combined Cycle	0.38 - 0.50	persons/MW

Other costs are maintenance and consumables.

Generally the availability of gas turbines is better than coal fired steam turbines, mainly due to the reduced duration of an outage, and the reduced start times.

2.6 NEW ZEALAND SITUATION

The author is employed by an electrical utility in New Zealand. For completeness the New Zealand electrical situation will be briefly described.

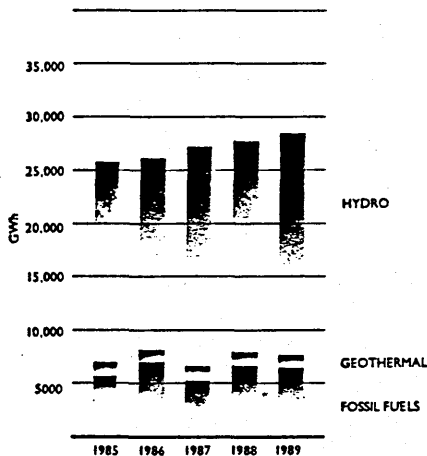
A number of electrical utilities are responsible for reticulation in their own areas within the country but as part of a "deregulation" exercise the boundaries of these areas are to be removed. A corporation called Electricorp has been formed from the old government operated power production and transmission department. The new company Electricorp no longer has a monopoly on power generation and the transmission part of the system is to be owned by a consortium of the Utilities and Electricorp.

The total highest peak load was 4876 MW for 1989 and 4765 MW for 1988 ($\frac{1}{2}$ hour peak). The predicted growth is 2% per annum. Although there is a surplus of generating capacity for the next 10 years, surplus natural gas is being flared off and new gas fields are being discovered at intervals. Proposals for new plant in competition to Electricorp have been made, including two combined cycle schemes, and Electricorp is looking at mothballing low efficiency thermal plants and converting a

couple of thermal steam plant to combined cycle. The system statistics are given in table 2.1.

The bulk electricity supply tariff is charged on a 20% peak and 80% energy basis, but the actual tariff is very complicated. The authors company charges differently for different customers ranging from 8.45 to 14.88 cents/kWhr, the overall charge per unit was 11.03 cents/kWhr at the end of 1989. The exchange rate over the last year has ranged from 2.6 to 3.0 \$NZ per pound sterling.

Table 2.1 New Zealand Statistical Information



PRODUCTION AND CONSUMPTION OF ELECTRICITY

		1985	1986	1987	1988	1989
Production						
Fossil fuels	GWh	5468	6189	5192	5424	5052
Geothermal	GWh	1206	1109	1174	1183	1237
Total thermal	GWh	6674	7298	6366	6607	6289
Hydro	GWh	19078	18659	20582	20891	21900
Total generated	GWh	25752	25957	26948	27498	28189
Purchased	GWh	19	22	21	20	20
Total available	GWh	25771	25979	26969	27518	28209
Consumption						
Sales of electricity	GWh	23923	24241	25187	25772	26436
Internal use	GWh	84	87	74	79	83
Transmission losses	GWh	1764	1651	1708	1667	1690
Total consumed	GWh	25771	25979	26969	27518	28209

Notes: 1. Thermal generation figures are net of own consumption at those power stations.

2. Internal use figures shown are for usage at hydro stations, switching stations, substations and some power station villages.

Total inflows to hydro catchments						
North Island	% of mean				85	119
South Island	% of mean				101	123
Total	% of mean				96	122
Peak demand						
24 hours	GWh				90.5	95
½ hour	MW				4765	4876
Generating system load factor	%	60	64.4	61	62.3	62.7
Plant availability						
Hydro	%	88.2	87.3	87.4	90.0	91.9
Thermal	%	70.1	64.8	73.2	76.4	82.5
System reliability	%	99.8	99.9	99.9	99.8	99.9
Transmission line availability						
HVAC	%	96.9	97.6	96.9	97.7	98.0
HVDC	%	96.5	95.0	96.5	96.4	90.8
HVDC usage (North and South)	GWh	4197	3201	4762	5021	4095

3 GAS TURBINES

3.1 INTRODUCTION

The gas turbine and its history have been introduced in section 1. Its characteristics have also been compared with the steam turbine in section 2. In this section the gas turbine is studied in depth.

Firstly the principles of operation are described. Then the various configurations and gas turbine applications are introduced. This is to supply a background to the gas turbine environment. The gas turbine engine performance is then looked at in detail, with a mathematical model developed for a hypothetical engine. This model is used to show the various effects of decisions made in the engine design such as pressure ratio and turbine entry temperature. The performance of published data from real engines is also presented. The off design performance of the gas turbine is investigated with the Cranfield program TURBOMATCH. The configurations of single and twin shaft, and of single and twin spool, are compared. Instabilities, transients and starting are described. The discussion is finished off with a description of two current engines.

Section 8 contains a discussion on state-of-the-art and future speculation.

3.2 CYCLES AND CONFIGURATION

The aim of the cycles section is to describe the cycle of the gas turbine, and how it can be modified for improvement. In depth performance analysis of alternative complex cycles is not carried out and these can be found in various texts and articles (e.g. [19] and [20]).

The cycle exists for the turbine, where the work is extracted. To turn the turbine a pressure is needed and is supplied by the compressor. To get more work out from the turbine than is required by the compressor, heat is added between these devices to raise the temperature. The process relies on the principle that the work is approximately proportional to the temperature difference, while the temperature ratio is related to the pressure ratio. If the pressure ratio remains the same, and thus so does the temperature ratio, then with a raised inlet temperature to the expansion results in a larger temperature difference in the turbine than in the compressor, and thus giving a net work output. It is thus advantageous to compress the air as cold as possible, and expand as hot as possible. The limits on temperature are the material strength limits.

The cycle of the gas turbine is the series of processes and property states that the working fluid goes through to extract work from the fuel. The most basic cycle is the so called 'simple cycle', where the fluid is air drawn from the atmosphere and passed through a compressor, combustion chamber and then a turbine. Modifications to the simple cycle are generally attempts at improving the simple cycle performance, either its efficiency, specific power or part load performance.

Terminology varies from one company and user to the next. In this document the compressor and turbine stators will be called vanes and

nozzles respectively. The compressor and turbine rotor blades will be called blades, and not buckets as is also common.

3.2.1 Simple Cycle

The simple cycle is shown in figures 3.1, 3.2 and 3.3. The ambient atmospheric air is drawn in at state 1. The compressor increases the pressure with as small a temperature rise as possible, because the temperature difference represents the work required. The smallest temperature rise obtainable is an isentropic compression, which is approximated as closely as possible. The compressor is inevitably a continuous rotary compressor, either radial in small sizes or axial in larger engines.

The air passes from the compressor, at state 2, to the combustion chamber where fuel is injected and burnt to raise the temperature. Ideally heat addition would have no pressure loss, but in reality there is a fundamental heating pressure loss, and losses due to the necessary mixing processes in the combustor. The combustion must take place in a local combustion zone at near stoichiometric conditions and high temperature, thus the combustor must also mix, dilute and cool the combustion products before discharging it to the turbine.

The hot and high pressure gasses at state 3 are passed through a turbine which expands the gasses in driving the turbine, and reduces the temperature as work is extracted. An isentropic process is again the ideal as it maximises the temperature differences. Some of the turbine work is required to drive the compressor, and the remainder is useful

Figure 3.1 Gas Turbine Schematic

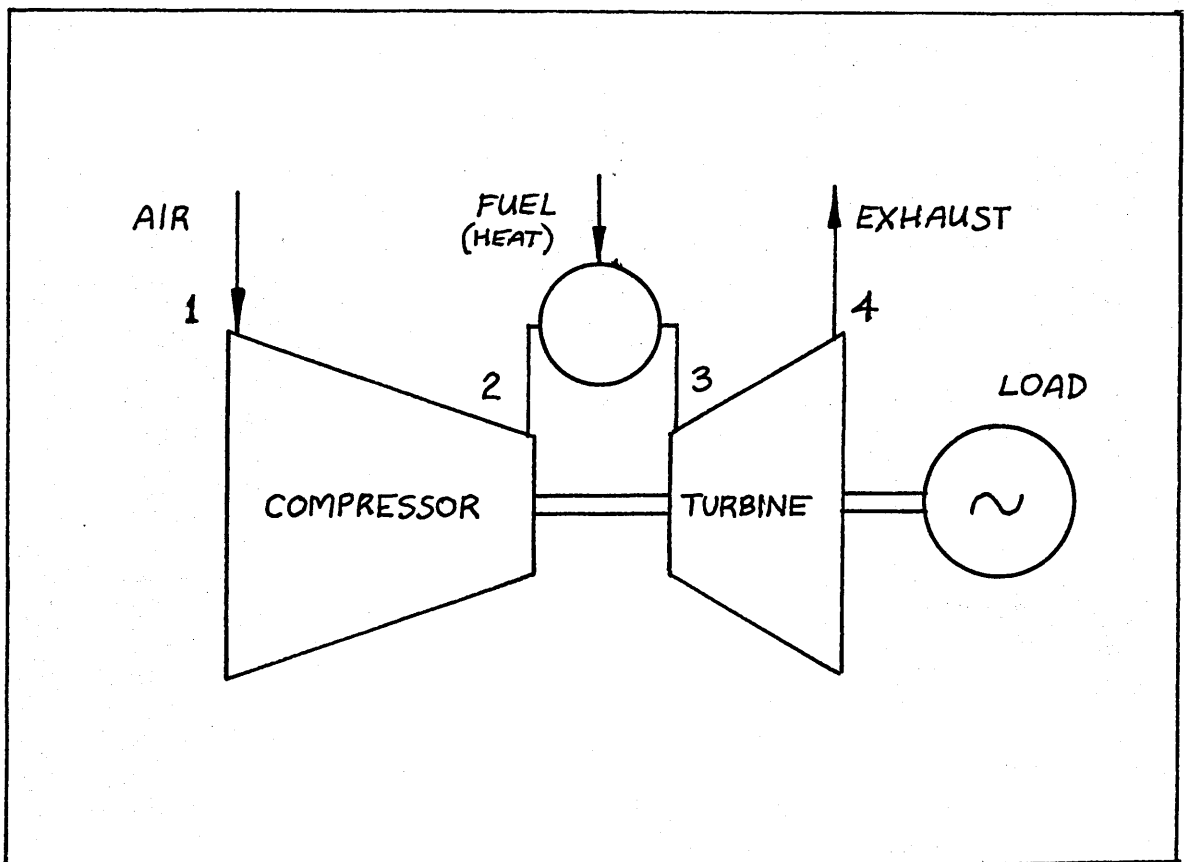


Figure 3.2 Gas Turbine Cycle - Pressure vs Temperature

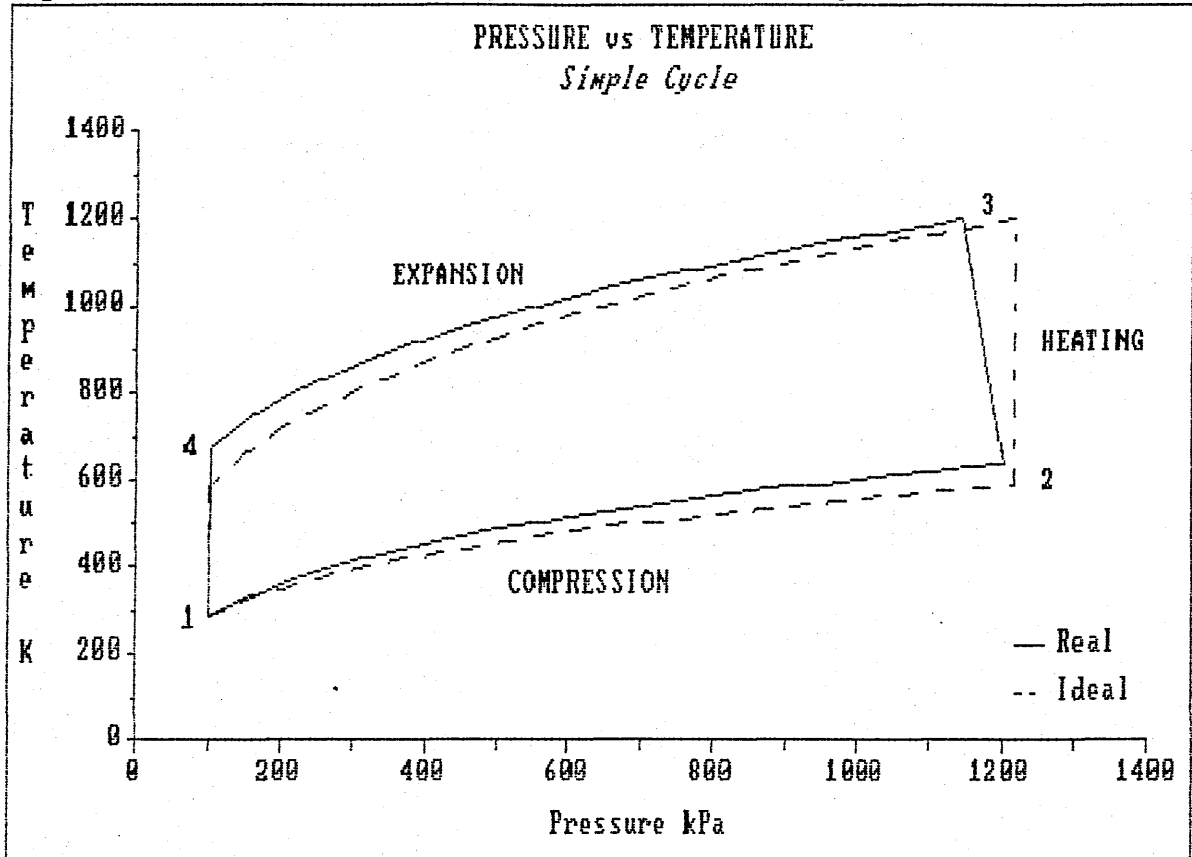
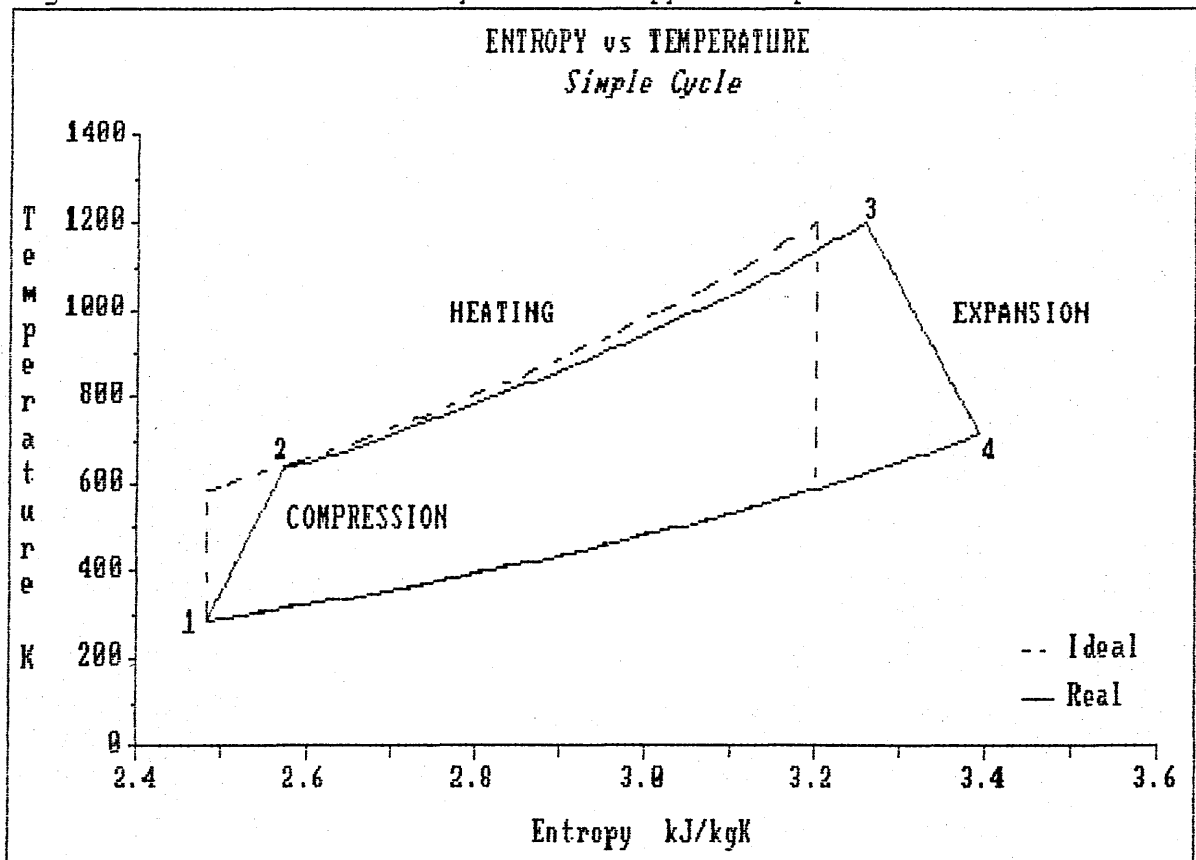


Figure 3.3 Gas Turbine Cycle - Entropy vs Temperature



output. The exhaust gas is at ambient pressure but elevated temperature, and heat is rejected at state 4.

Real cycles are rarely simple. They usually involve some extra air bled off the compressor and used to cool the turbine blades and to seal disk spaces. This air does not get heated in the combustion process and has mixing losses and dilutes the temperature of the working gas. Also the turbine air mass flow is increased by the fuel flow. Other cycle differences are in the arrangement of shafts, but these cycles are still called 'simple' cycles. Complex cycles can be used to prop up the performance, such as regenerators, reheaters and intercoolers.

The two major variables in the cycle are the pressure ratio (r_p) and the Turbine Entry Temperature (TET). However different definitions are used. Typically the pressure ratio is the ratio over the compressor, which is more than the turbine pressure ratio due to the combustor pressure loss. This loss is typically 4 to 5%. The peak cycle temperature is sometimes taken as the average temperature of the diluted and mixed combustion products as they enter the turbine, called the TET. But for cycle analysis the highest temperature at which work is extracted is the first rotating blade entry temperature, called the Turbine Rotor Inlet Temperature (TRIT). The TRIT is lower than the TET due to cooling flow in the first stationary nozzles, and often the TRIT is also quoted as the TET.

Simple cycle gas turbines can be configured on one continuous shaft, and therefore all stages operate at the same speed. These units are well suited to electrical generation where speed variation is not required, and especially in base-load where power variation is low. They are called 'single shaft' engines and are shown diagrammatically in figure 3.4.1.

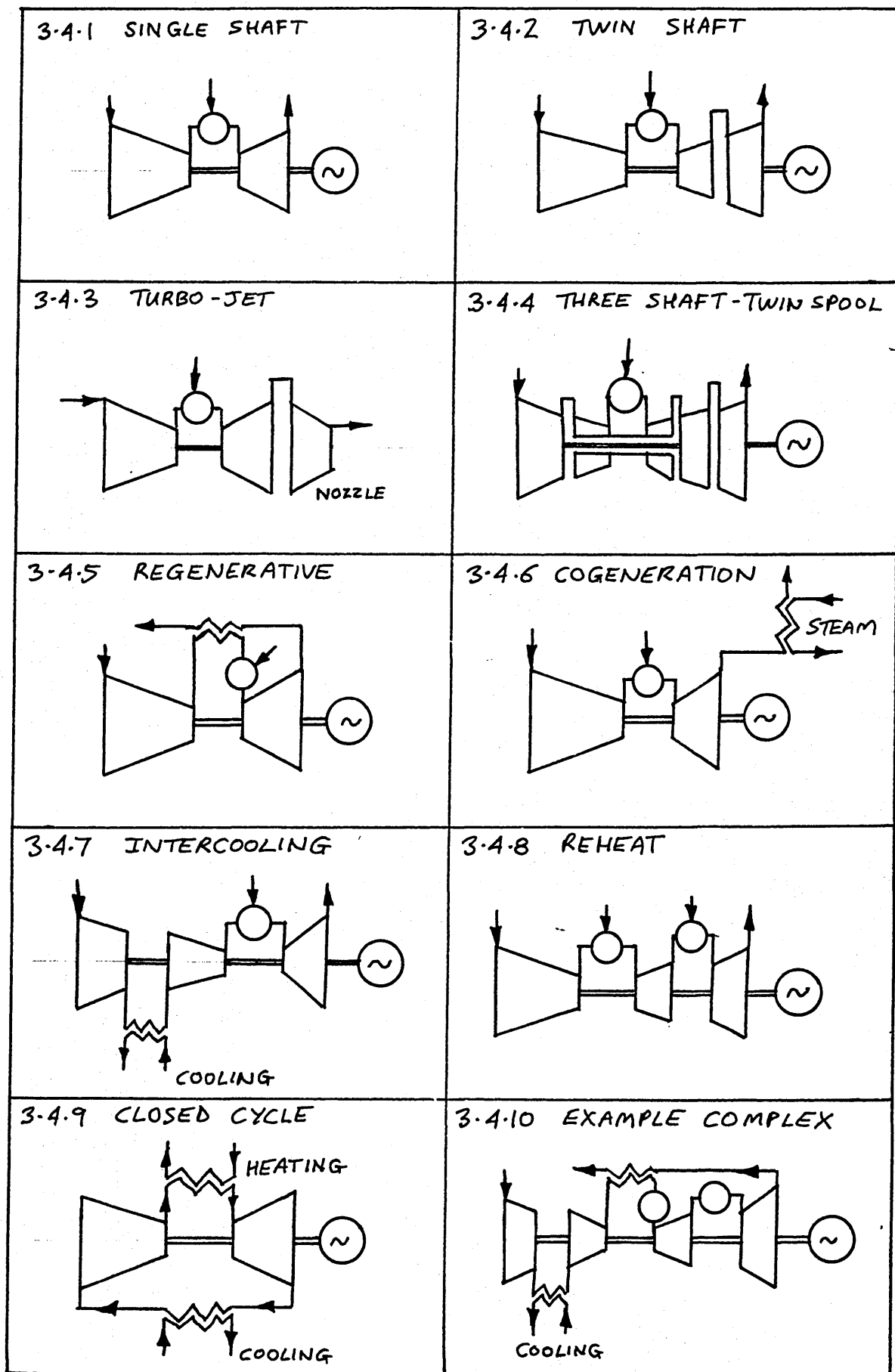
A diagram for a simple cycle twin shaft engine is shown in figure 3.4.2. Here the turbine is mechanically separated into two sections, one which drives the compressor and one which drives the output shaft. Here the compressor and high pressure turbine combination is called the 'gas generator' while the low pressure turbine connected to the load is called the 'power turbine'. Two shaft engines are particularly suitable for mechanical drive situations such as pumps and propellers where speed variation is required, although they are also suitable for power generation.

A two spool engine is where the gas generator is made up of two shafts, typically concentric, each with it's own compressor and turbine pair. The high pressure (HP) compressor is connected to the HP turbine. This is shown in figure 3.4.4. With the power turbine there are now three shafts.

3.2.2 Regenerative Cycle

The regenerative cycle, or sometimes just known as the heat exchanger cycle, makes use of the still high temperature of the exhaust gas. If the exhaust gas temperature is significantly above the compressor discharge temperature then the compressor discharge can be passed through a heat exchanger with the exhaust (figure 3.4.5). In this way some of the exhaust heat is recovered, heating the compressor discharge flow and reducing the temperature rise required by the combustor. Both

Figure 3.4 Gas Turbine Cycle Configurations



the efficiency and specific work is improved.

The exhaust temperature is only significantly higher than the compressor discharge at relatively low pressure ratios, and the peak performance is at a lower pressure ratio than for a simple cycle. This means that the increase in size due to the heat exchanger is partially offset by the reduced compressor size. The gains in reality are reduced by the additional pressure drops across the heat exchanger, and it's high cost. Two types of heat exchanger are used, either a conductive wall separating the flows or a matrix of material alternately exposed to the hotter and cooler flows. This second type is used in most of the land based vehicle designs to reduce size, but it suffers from mass leakage between flows and the need for parasitic power to move the matrix.

3.2.3 Reheat Cycle

Reheat is the process of extracting the gas part way through the expansion, burning more fuel in it to raise the temperature again, and then completing the expansion (figure 3.4.8). This increases the temperature differences in the expansion and thus the work extracted. In an aircraft this is done downstream of the turbine, but before the nozzle expansion and is sometimes called "after burning". The result of reheat is a substantial increase in specific work, thus power, at the expense of efficiency. This is of no virtue in utility power generation without regeneration or some other exhaust heat recovery.

Reheat with a regenerator gives some improved specific work and efficiency, but the performance gain is reduced by pressure losses in the reheat chamber and heat exchanger, and increased cooling required for the increased surfaces exposed to high temperature. Reheating in a combined cycle with exhaust heat recovery gives significant gains in both specific work and efficiency. Reheated combined cycles are not currently available but are discussed fully in section 7.7.2. A form of reheat is used in combined cycles where fuel is fired in the exhaust duct but this is more correctly called supplementary firing. It is discussed in section 7.7.1.

3.2.4 Intercooling Cycle

Intercooling is the process of extracting the air part way through the compression and cooling it in a heat exchanger (figure 3.4.7). The result is colder compression temperatures and thus less work required for the compression. It is easier to accomplish than reheat, and has been used in compression for many years. It can give improved efficiency and specific work for simple cycles, but the gains do not always justify the added expense, and the large volumes of cooling water that are also required. The two main methods are using noncontact heat exchangers or using water spray evaporative coolers. An evaporative cooler has a continuous loss of high quality treated demineralised water. Intercooling is seldom used except for processes requiring unusually high pressure ratios, such as air storage. With heat recovery intercooling may reduce the overall efficiency, and intercooling in a combined cycle is discussed fully in section 7.7.5.

3.2.5 Complex Cycles

Combinations of reheat, regeneration and intercooling together give potentially greater improvement than each individually, but the plant is then bulky and expensive. A further major problem is development costs. These cycles have in the past shown insufficient advantage for standard power generation compared to the increased costs, complexity and reduced reliability, and these modifications are not used regularly by electric utilities. Combined cycles have tended to use unaltered simple cycle gas turbine with a separate heat recovery steam cycle.

Currently some interest is being shown in the complex cycles for marine use and naval engines using intercooled and regenerative cycles are being produced. These have higher efficiency and very much improved part load characteristics. It is claimed that the part load efficiency at 50% load is the same as the full load efficiency. The Japanese have also investigated a high pressure ratio, intercooled and reheated gas turbine for combined cycle power generation which is discussed in section 8.5.5. Steam injection is also being used but is more a form of combined cycle and is included in section 7.12.

3.2.6 Closed Cycles

Closed cycles are where the same gas is recirculated and used repeatedly (figure 3.4.9). The heat addition is by heat exchanger so as to maintain the internal gas composition constant. The exhaust must be cooled by heat exchanger before entering the compressor again. The main advantage is that there is now no limiting ambient atmosphere and thus the overall cycle can operate at a different inlet pressure, with a different gas and is unaffected by weather. The compressor inlet pressure is raised to reduce plant size, and the pressure levels are adjusted at part load to keep the TET and speed constant so there is no reduction of efficiency at part load. The other main advantage is that the working fluid is contained and can be kept clean, not fouling the compressor or turbine. The combustion is external, thus any heat source or fuel can be used. These cycles are discussed more fully under the applications section.

3.3 APPLICATIONS

3.3.1 Aircraft

The gas turbine is a large success in aircraft, largely due to its high power to weight ratio. The cycle is similar to the industrial turbine, especially in a turboprop where the shaft power drives a propeller. In a turbojet engine the turbine only expands enough to drive the compressor and the remaining expansion is through a nozzle to impart momentum to the exhaust. The main difference to power generation applications is that the weight is very important and that the engine is optimised for different economics.

The aircraft engine weight, size and efficiency is so important that the engine can be operated at higher loadings and temperatures at the expense of a shorter life. A rule of thumb is that 1 kg of engine weight saved is equivalent to 5 kg of aircraft weight, thus payload. British Airways stated recently that a saving of 0.45 kg (1 lb) in engine weight is worth 500 £/yr to the airline. Also fuel carried in a subsonic transport aircraft ranges from 20-35% of its total weight thus an improvement in efficiency is also important to civil aircraft performance.

The aircraft engines are useful to industrial users as they are readily modified for industrial use. The nozzle is replaced by a turbine and the engine modified for the different conditions. They are smaller, lighter and of higher efficiency than equivalent 'heavy duty' engines, and are often in a highly modular form which assists in maintenance. Aircraft engines are in general quicker starting and loading, with shorter response times. An engine designed for aircraft but used on land is called an aero-derivative. The different characteristics are discussed more fully in section 9.3.

3.3.2 Road and Rail

Land vehicles powered by gas turbines have been used in a number of occasions but this application has not been successful except in isolated examples. The main advantage is the high power to weight ratio and the small volume. There are also low emissions relative to piston engines. Disadvantages are the poor efficiency, slow response to throttle and poor torque profiles. There is only a modest gain in torque with reduced speed and little engine braking. Cycles are usually two shaft with regeneration. A notable current use is the M1 battle tank. Past uses have included trains and cars for limited periods. Currently ceramic vehicle engines are being developed and their success or otherwise remains to be seen. It is interesting that a combined gas and steam cycle high speed train locomotive was developed in the 1950's and was competitive technically in certain areas but not successful. Using modern technology it would be even more competitive today. Some references of interest are the M1 tank [36] and ceramic vehicle [37].

3.3.3 Marine

In the naval area the gas turbine has been very successful in recent years, currently representing about 50% of the US naval warships. The main problem is the large power variations in a ship with cruise power say only 1/8 of the power at full speed. To combat this the gas turbines are often used in conjunction with other engines such as diesel. Notation is used to denote the configuration, for example CODOG, meaning Combined Diesel Or Gas turbine. Other arrangement examples are COSAG, Steam And Gas turbine, and COGOG, Gas turbine Or Gas turbine.

The US navy first used gas turbines in Patrol Gunboats in 1964. They were CODOG with an LM1500. The boats could accelerate from standing to full speed (over 40 knots) in 1.5 boat lengths. Currently ships of all sizes can be gas turbine powered, for instance the Invincible class aircraft carriers with 4 Olympus aero-derived engines. Non-military applications are rare, but include ships such as liquified natural gas tankers which may be powered by boil off gas. Complex cycles such as intercooled and regenerative cycles are produced to improve efficiency and part load performance. Some references are [108] and [158].

3.3.4 Pumping and Oil Platforms

Gas turbines are often used in pumping for oil and gas pipelines. Also the electricity supply of offshore platforms comes from gas turbines. The engine may be able to use the fuel from the pipeline and only requires minimal supervision. On offshore oil platforms the weight and size of the engine is paramount and the fuel is often cheap, or even

waste fuel which would be otherwise flared. This suits a gas turbine characteristics well.

3.3.5 Electricity Generation

Gas turbines have been used since 1939 and currently they are widely applied. The simple cycle is used for reserve and peaking, often aero-derivative, and as a combined cycle the gas turbine is used for medium to base-load. Uses have included 'repowering' of existing steam plant, which is discussed in section 7.3.2.

Gas turbines are well suited to emergency generator service. Examples are telephone exchanges, banks, airlines, factories, computers, nuclear power plants and many others. The main features required in emergency power are high start reliability, quick start, acceptance of full load on synchronous speed, stable frequency and voltage control under rapid load fluctuations and fully automatic starting. These features are possible in gas turbines.

3.3.6 Cogeneration

Cogeneration is the use of rejected heat for another purpose, and the Gas Turbine has found a valuable niche. The gas turbine is usually used to supply electricity and its exhaust supplies the heat, which is termed Combined Heat and Power or CHP. Cogeneration includes district heating, process steam in industry, desalination, drying and other varying uses. This topic is discussed in its own section 7.3.3.

3.3.7 Air Storage

Electricity cannot be practically stored, and peaking capacity is usually of low efficiency as its low usage cannot justify much expense. It is therefore advantageous to convert the fuel energy in large base-load plant during low demand and then store this for later use at high demand. This can be done by storing the compressed air from a gas turbine compressor, and expanding it through the turbine only when needed. Storage would be in large underground caverns such as mined or natural cavities, or porous formations as used in natural gas storage. Possible configurations are shown in figure 3.5. The compressor and turbine are aligned but separated by a synchronous generator/motor, which is both a motor for charging the caverns and a generator for peaking.

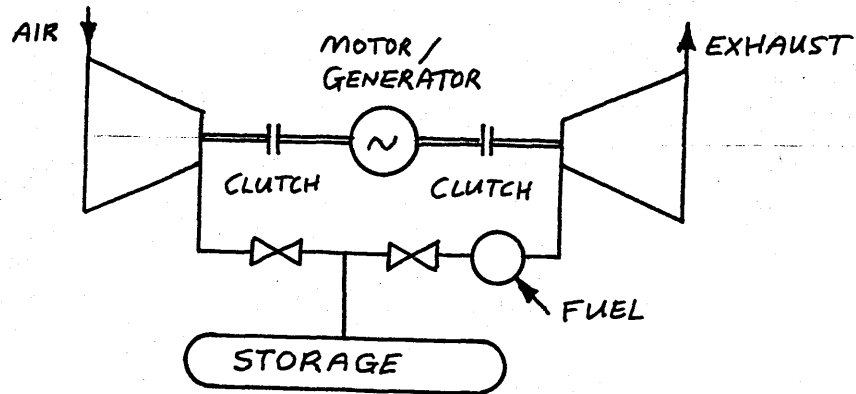
Air storage plant is either an adiabatic plant with no fuel burnt, or a diabatic plant where fuel is burnt at the turbine. The storage efficiency of an adiabatic plant is about 75%, which is the same value as pumped hydrostations. The incremental efficiency of conversion of an adiabatic plant to a diabatic plant gives very high efficiencies for the additional power.

Air storage is not widely used but it is past the demonstration stage, the first plant being 290 MW at Huntorf in Germany. This plant uses mined salt caverns of 300,000 m³ at 50-70 bar. The air storage plant has a high output for a gas turbine as the compression work is separated, and this effect is amplified for emergency operation as the compression power draw-off can be reduced further and the turbine power added giving an effective backup in the case of Huntorf of 320 MW. The concept has

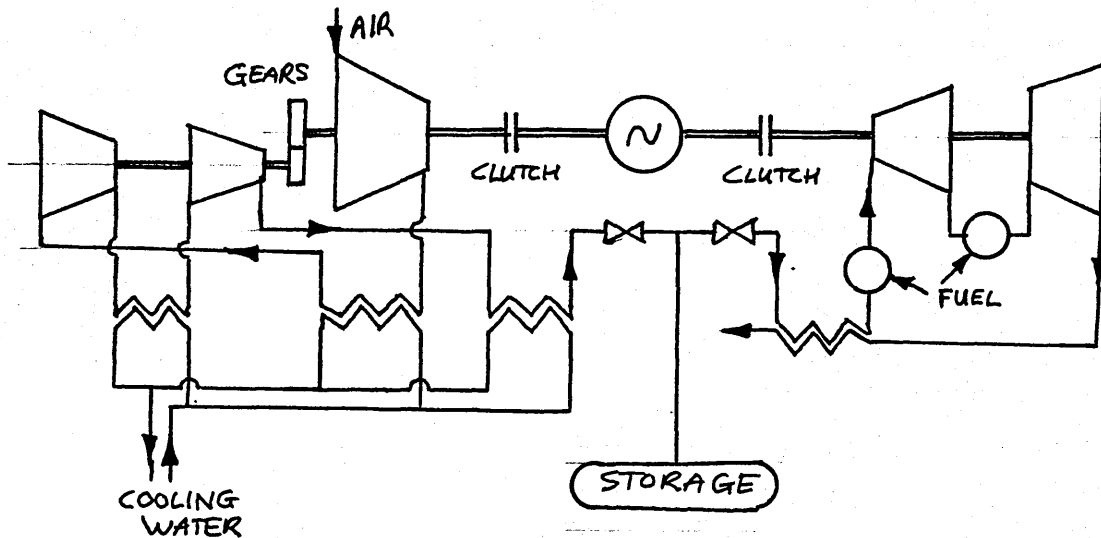
Figure 3.5

Air Storage Power Plant Configuration

3-5-1 CONCEPTUAL LAYOUT



3-5-2 PRACTICAL LAYOUT



proved functional and cost-effective. Additional references are [31], [32] and [33].

3.3.8 Closed Cycles

Closed cycles have been used for the last 30 years, notably five in Germany. These plant have been operated successfully and profitably, but have never moved into large scale use. In 1977 about a dozen plants existed in the 2-20 MW range, with compressor inlet pressure up to 40 bar.

The advantages as listed under the cycles section 3.2.6 are; smaller size, maintaining part load performance, utilisation of any fuel or heat source, no fouling or performance deterioration, and the potential of using a better property gas. The efficiency in practice can be maintained down to approximately one third of the rated output. The disadvantages are cost and complexity. This is especially true of the heat exchangers, although the necessary heat exchanger is smaller than

in a comparable regenerative cycle due to higher pressures. Another major disadvantage is that the TET is limited by the heat exchanger materials and the peak temperatures will be significantly lower than in an internal combustion turbine, penalising efficiency. Generally below about 30-50 MW the cycle would be air for cost reasons, and over this size using helium as it improves efficiency and more importantly reduces heat exchanger size. The heat exchanger would be about 35-40% of the total cost.

The cooling water is hot enough for cogeneration, although the gas turbine may still require further cooling. Germany's closed cycles are mainly used in cogeneration roles. In a cogeneration role the heat to power ratio is variable. The closed cycle also can be used integrated in a gas cooled nuclear reactor, or with fuels such as coal, peat, refuse, solar and chemical. Closed cycles are used or proposed for special purposes such as solar powered space stations. In a space station a phase change material would be used to store solar energy as latent heat for continuous operation over day and night. This is discussed in reference [40].

The reason the closed cycle has not been used more widely is that the capital cost is high, as much as two times that of a steam plant, and efficiency is low. Other problems are that closed cycles are not available in large capacity and that there is little data available on operating experience. Experiences and discussion can be found in references [42] and [219].

3.4 PERFORMANCE

The gas turbine market has many engines optimised for different operating conditions which all have their own advantages and disadvantages. The aim of this section is to investigate the behaviour of the gas turbine, firstly under different design conditions, and then how the performance varies under part load for a particular engine. The reason for the analysis is to understand the compromises in engine design and to provide background for the later combined cycle analysis. The performance of the engine under design conditions is called the 'design point' performance, while the performance of an engine under various other loads and conditions is called 'off-design' performance. The study will first look at design point and then off-design.

As has been discussed in the previous sections, there are a multitude of engine configurations and cycles. To analyse all possible cycles would produce an enormous number of performance curves and is beyond this thesis. As power generation, both gas turbine only and combined gas/steam generally tends to use only simple gas turbine cycles, the analysis will be limited to the simple cycle. Even the simple cycle analysis can be treated to any degree of detail, and will be limited here to the most important effects on a hypothetical engine. The conclusions will be the same although the numerical results will not be for any particular installation. References for cycle analysis include [19] and [20].

3.4.1 Ideal Simple Cycle

The ideal cycle (figure 3.2) consists of an isentropic compression, a constant pressure heat addition and an isentropic expansion. The gas is

assumed ideal with constant heat capacity. This is fundamentally and practically unobtainable and thus of limited use, but gives an upper bound on the performance of the gas turbine. Other assumptions are that kinetic energy changes are negligible, and fuel mass flow is not added to the main flow.

The isentropic compression of an ideal gas is represented by the polytropic relation;

$$r_p = \left[\frac{P_2}{P_1} \right] = \left[\frac{T_2}{T_1} \right]^{k/(k-1)} \quad \dots 3.1)$$

and the efficiency;

$$\eta_{TH} = \frac{\text{work output}}{\text{heat supplied}} \quad \dots 3.2)$$

It can be shown that;

$$\text{ideal efficiency } \eta_{TH}' = 1 - \frac{1}{r_p^{(k-1)/k}} \quad \dots 3.3)$$

and nondimensionally that;

$$\text{ideal specific work } \frac{W_{21}'}{C_P T_1} = T_1 \left[1 - \frac{1}{r_p^{(k-1)/k}} \right] - [r_p^{(k-1)/k} - 1] \quad \dots 3.4)$$

The above formula are plotted in figures 3.6 and 3.7 (air $k = 1.4$).

The efficiency is seen to increase with pressure ratio which is explained in that the increased pressure ratio means a greater expansion ratio and hence more work extracted. The specific work is seen to have a maximum and thus there is an optimum pressure ratio for minimum size of machine. The specific work variations can be explained by figures 3.8 and 3.9, which show that at lower pressure ratios there is less expansion ratio to extract work, while at the higher pressure ratios the higher compressor discharge temperatures means there is too little temperature rise available to add the fuel heat. There is an obvious conflict between maximum efficiency at large pressure ratio and minimum plant size at modest pressure ratio.

The ideal cycle is misleading as it gives two major misrepresentations, that the efficiency is independent of the TET, and that the efficiency increases with pressure ratio continually. Both these effects are wrong in real cycles and thus the ideal performance curves must be seen as upper bounds only. The effect of inefficiencies are shown later in figure [] where an attempt is made at showing the size of each effect.

3.4.2 Real cycles Design Point

In real cycles there are various inefficiencies also effecting performance. Mathematical models can include these losses to any degree of detail and get increasingly complex. A model is developed here of one particular engine configuration with the major losses taken account of. This model is used to investigate the effects of varying the design parameters of the gas turbine, and is later included in the combined cycle calculations. The following is a summary of the mathematical model used for each component for the design point calculation.

Figure 3.6 Ideal Cycle - Efficiency vs Pressure Ratio

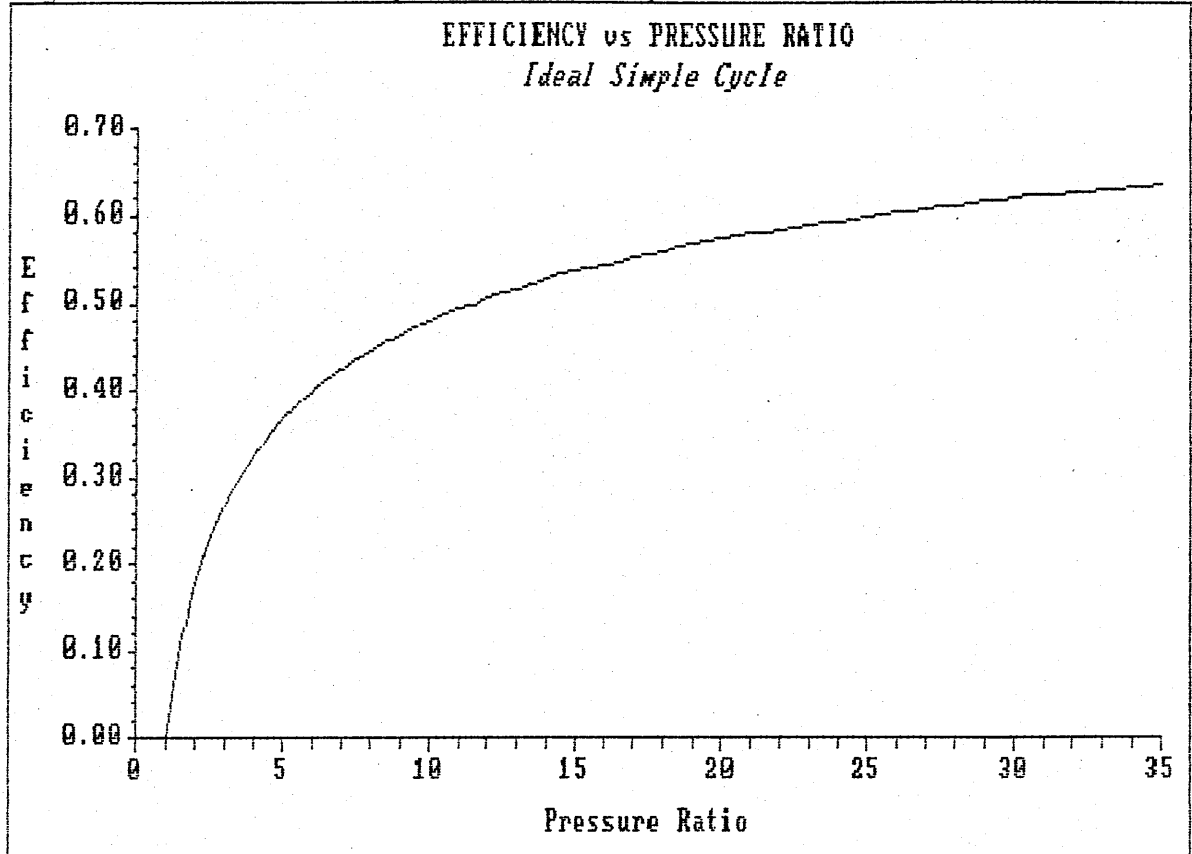


Figure 3.7 Ideal Cycle - Specific Work vs Pressure Ratio

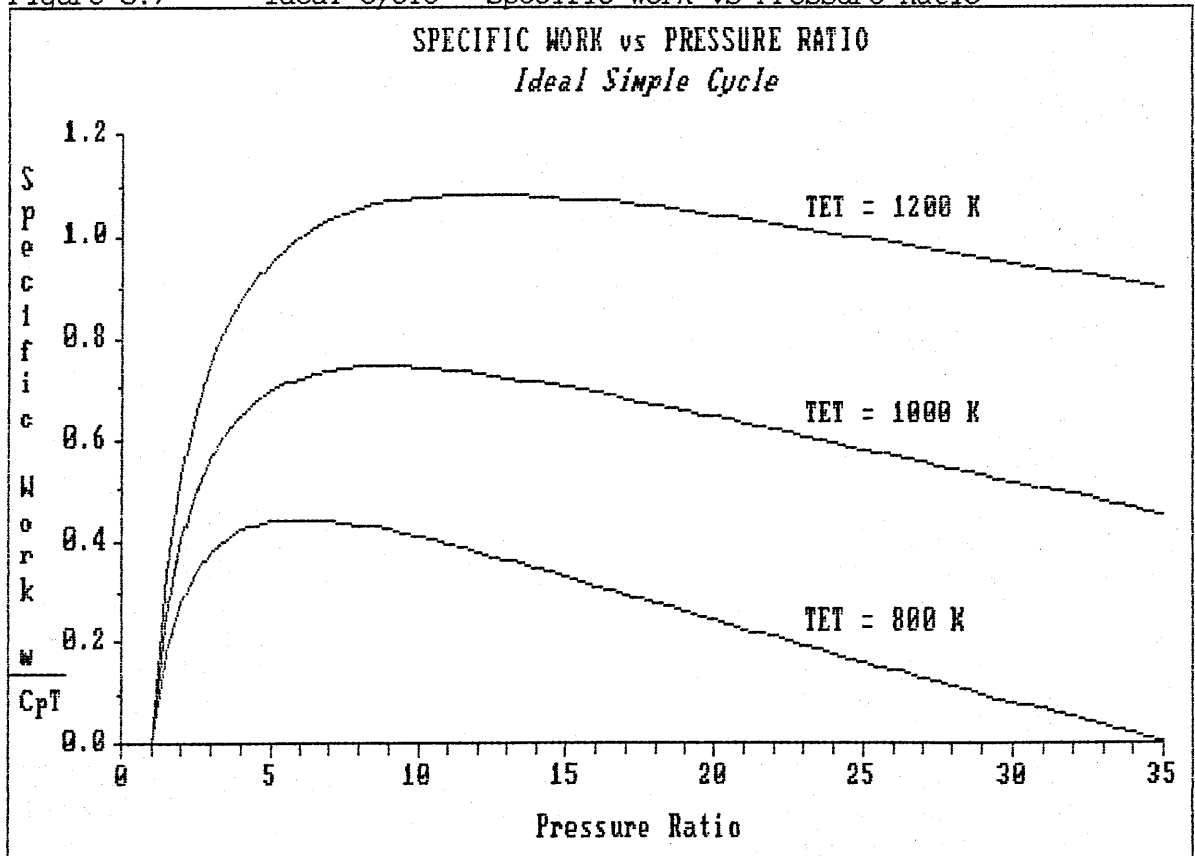


Figure 3.8 Cycle Pressure vs Temperature - Varying Pressure Ratio

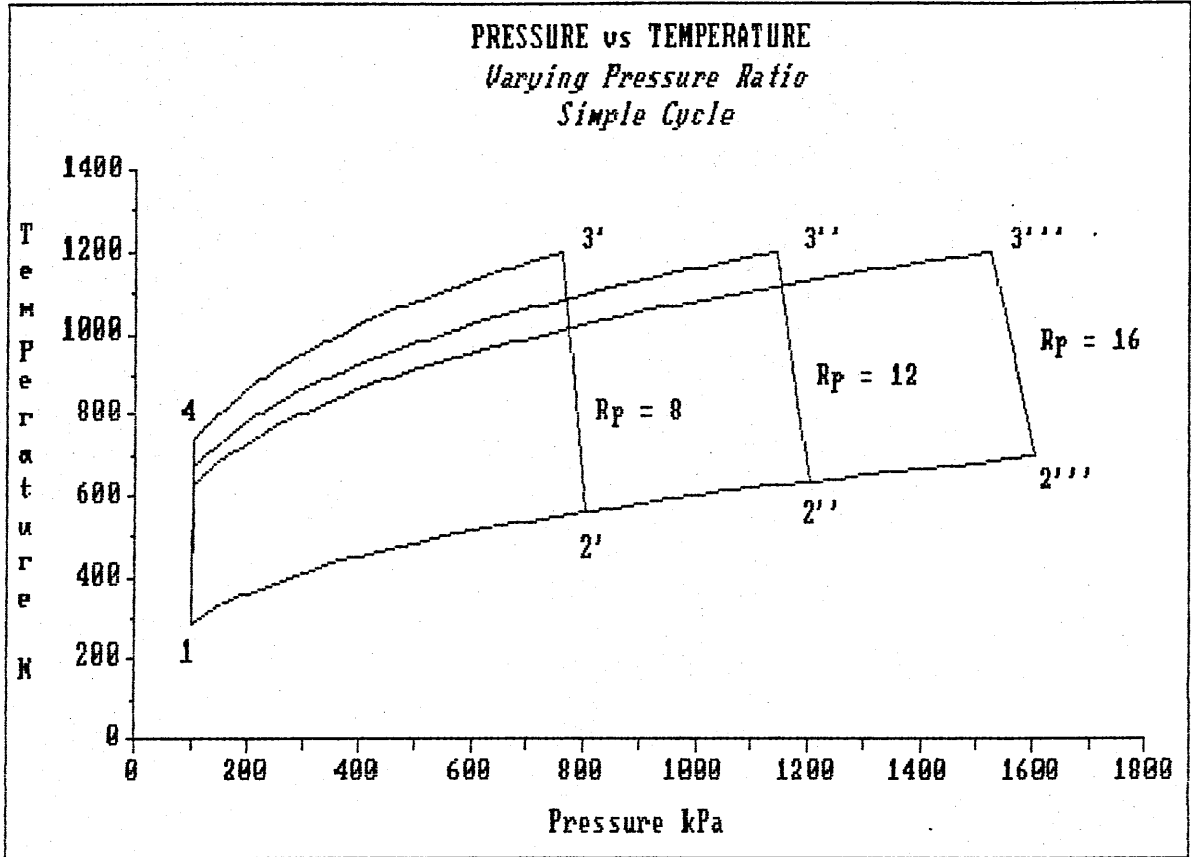
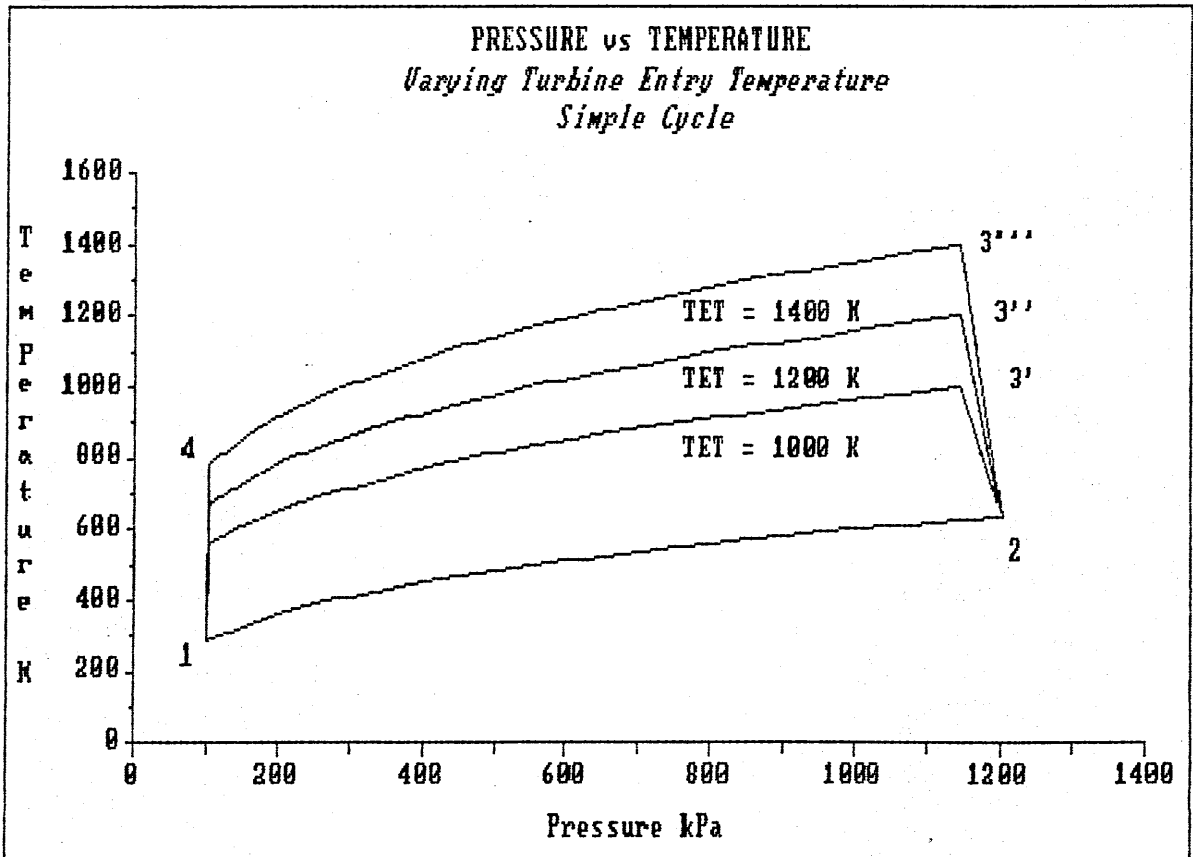


Figure 3.9 Cycle Pressure vs Temperature - Varying TET



Air. The air is assumed to exhibit near ideal gas properties, although the heat capacity is varied at different points to account for the temperature dependence. A function for the heat capacity of dry air was developed by the author from data in the tables of reference [1];

$$C_p = 991.24 - 0.0141 T + 0.000186 T^2 \quad \dots 3.5)$$

accurate to within 0.3% of the tables over the range of 280 to 800 K.

The combustion products heat capacity is assumed to follow the equation,

$$C_p = 950.0 + 0.21 T \quad \dots 3.6)$$

This is from reference [20] derived from NGTE report R74 for kerosine fuel. In reality the exhaust heat capacity depends on the fuel type, and air humidity, but the fuel correlation is also for kerosine and sufficient for this purpose.

Intake. The air intake will always have some pressure losses, and in this case it also includes the filter losses. The losses are assumed to be a fixed proportion of the ambient pressure, which is valid for design point calculations where filters and intake ducts would be sized for the flow. Temperature changes are usually small and negligible.

$$\text{Intake} \quad T_{out} = T_{in} \quad \dots 3.7)$$

$$P_{out} = P_{in}(1-x_p) \quad \text{where } x_p \text{ is the loss fraction} \quad \dots 3.8)$$

$$\dot{m}_{out} = \dot{m}_{in} \quad \dots 3.9)$$

Compressor. The compressor would have a characteristic performance 'map,' mapping the performance over the operating range. This information is not necessary at design point stage, and a single efficiency is used.

There are two common efficiencies used which are defined as follows.

η_{ic} isentropic compressor efficiency

$$\eta_{ic} = \frac{\text{isentropic work required}}{\text{actual work required}} \quad \dots 3.10)$$

η_{pc} polytropic compressor efficiency

$$\left[\frac{P_2}{P_1} \right] = \left[\frac{T_2}{T_1} \right]^{\eta_{pc}k/(k-1)} \quad \dots 3.11)$$

In this model the isentropic compressor efficiency is used which gives the following equation (ideal isentropic denoted by ' ').

$$\text{compressor} \quad (T_{out} - T_{in}) = (T_{out} - T_{in})' / \eta_{ic} \quad \dots 3.12)$$

$$P_{out} = r_p P_{in} \quad \text{where } r_p \text{ is pressure ratio} \quad \dots 3.13)$$

$$W_c = \dot{m}_{in} C_{p_{in}} (T_{out} - T_{in}) \quad \dots 3.14)$$

$$\dot{m}_{out} = \dot{m}_{in} \quad \dots 3.15)$$

Bleed. Most turbines use some compressed air for cooling or sealing. The cooling and sealing flows can be tapped off the compressor in several places, but in this model it is assumed they are all removed at the compressor discharge pressure.

$$\text{Bleed} \quad \dot{m}_{out} = \dot{m}_{in}(1 - x_m) \quad \text{where } x_m \text{ is fraction of bleed} \quad \dots 3.16)$$

$$T_{out} = T_{in} \quad \dots 3.17)$$

$$P_{out} = P_{in} \quad \dots 3.18)$$

Combustor. The combustor is treated as a temperature rise, a pressure drop and a mass increase by the fuel mass flow. The pressure drop consists of two parts, the fundamental loss, due to momentum effects, and the so called cold loss due to the friction in the necessary mixing processes. The loss is treated as a fixed fraction, which is possible in a design point case, but even at off design conditions the fraction would remain approximately constant for a particular design.

$$\text{Combustor } T_{out} = TET \quad \dots 3.19)$$

$$P_{out} = P_{in}(1 - x_p) \quad \text{where } x_p \text{ is the loss fraction} \dots 3.20)$$

$$\dot{m}_{out} = \dot{m}_{in} + \dot{m}_{fuel} \quad \dots 3.21)$$

The fuel flow is obtained from the fuel lower heating value and the required temperature rise. Due to dissociation the temperature rise is a function of the temperature and also a slight function of the pressure. Normally combustion temperature rise charts or combustion programs are available for accurate calculations, but for this comparative purpose a correlation of some typical curves is sufficient. The pressure effect is assumed negligible, for example an increase of pressure from 1 bar to 100 bar decreases calorific value by only about 1%. The correlation used is from reference [20].

$$FAR = \frac{1.1(TET - T_{in} - 50)(1 + T_{in}/3250)}{\eta_{comb}LCV} \quad \dots 3.22)$$

This is accurate for aircraft kerosine fuel to $\pm 1\%$ over the range $400 < TET - T_{in} < 900$ °C, centred on a fuel of LCV 42000 kJ/kg. The use of kerosine fuel does not effect the trends of the analysis, even though natural gas is the more likely fuel in electricity generation. The off design program TURBOMATCH also uses kerosine and thus this is consistent with later calculations. The effect of different fuels is discussed more fully in sections 4 and 3.6 and in reference [62].

$$\dot{m}_{fuel} = \dot{m}_{in} FAR \quad \dots 3.23)$$

Flow Mixing. The cooling and sealing flows are mixed back into the main flow at various points after the combustion, and some may be lost overboard. The actual location of the mixing will be at the blade tips, trailing edges and other places. It is usual in calculations to lump the flows together either before or after the turbine, or turbine stage, in such proportion as to give the same effect. In this model the flow is remixed in two places, before the fixed turbine and before the power turbine. The fraction to each can be varied, and no mass is lost overboard.

$$\text{Remixing } \dot{m}_{out} = \dot{m}_{in} + \dot{m}_c \quad \dots 3.24)$$

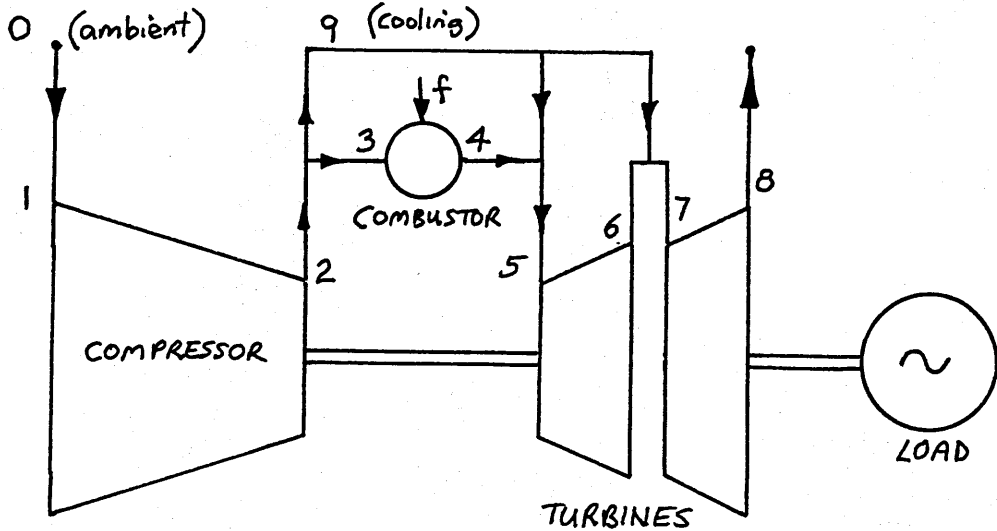
$$T_{out} = T_{in} - \dot{m}_c / \dot{m}_{in} (T_{in} - T_c) \quad \dots 3.25)$$

$$P_{out} = P_{in}$$

where $\dot{m}_c \ll \dot{m}_{in}$ and $\dot{m}_c$ is the cooling and/or sealing flow

Turbine. The turbine is similar to the compressor in that it is treated by including an efficiency on the isentropic process. As in the compressor two efficiencies are commonly used but the one in this model

Figure 3.10 Hypothetical Gas Turbine Configuration



is the isentropic efficiency (ideal isentropic denoted ').

$$\text{turbine} \quad (T_{in} - T_{out}) = \eta_{it}(T_{in} - T_{out})' \quad \dots 3.26)$$

$$W_t = \dot{m}_{in} c_{p1} (T_{in} - T_{out}) \quad \dots 3.27)$$

$$\dot{m}_{out} = \dot{m}_{in}$$

The pressure at outlet is either calculated from the required work for the first turbine or from atmospheric plus exhaust losses for the power turbine.

Exhaust. The exhaust will have exhaust stack and if applicable heat exchanger pressure losses.

$$P_{in} = P_{amb} + P_{loss} \quad \dots 3.28)$$

The example hypothetical engine used in this thesis is shown in figure 3.10 and the following is a summary of the theory.

Intake	$T_1 = T_{AMB}$
	$P_1 = P_{AMB}(1 - X_{P01})$
Compressor	$P_2 = \gamma_{P12} P_1$
	$\gamma_{T12}' = \gamma_{P12}^{(k-1)/k}$
	$\Delta T_{21}' = \gamma_{T12}' T_1 - T_1$
	$\Delta T_{21} = \Delta T_{21}' / \eta_{1c}$
	$W_{12} = \dot{m}_{1n} c_{p1} T_{21}$
	$T_2 = T_1 + \Delta T_{21}$
Cooling bleed	$\dot{m}_9 = \dot{m}_{1n} X_{C29}$

Combustor	$\dot{m}_3 = \dot{m}_{in} - \dot{m}_9$
	$\Delta T_{43} = TET - T_3$
	$FAR = \frac{1.1(\Delta T_{43} - 50)(1 + TET/3250)}{LCV \eta_{comb}}$
	$\dot{m}_f = \dot{m}_3 FAR$
Mixing	$P_4 = P_3(1 - X_{P43})$
	$\dot{m}_5 = \dot{m}_4 + \dot{m}_9(X_{C95}) + \dot{m}_f$
	$T_5 = TET - \dot{m}_9 X_{C95} / \dot{m}_4 (TET - T_2)$
Fixed turbine	$W_{56} = W_{12} / \eta_{MECH}$
	$\Delta T_{56} = W_{56} / (\dot{m}_5 C_{p5})$
	$\Delta T_{56}' = \Delta T_{56} / \eta_{it}$
	$r_{P65} = (T_5 / (T_5 - T_{56}'))^{k/(k-1)}$
	$P_6 = P_4 / r_{P65}$
	$T_6 = T_5 - T_{56}$
Mixing	$\dot{m}_7 = \dot{m}_1 + \dot{m}_f$
	$T_7 = T_6 - (1 - X_{C95}) \dot{m}_9 / \dot{m}_5 (T_6 - T_2)$
Power turbine	$P_8 = P_{AMB} + P_{LOSS}$
	$r_{P87} = P_6 / P_8$
	$r_{T87}' = r_{P87}^{(k-1)/k}$
	$T_8' = T_7 / r_{T87}'$
	$\Delta T_{87}' = T_7 - T_8'$
	$\Delta T_{87} = \Delta T_{87}' / \eta_{it}$
	$T_8 = T_7 - \Delta T_{87}$
	$W_{78} = \dot{m}_7 C_{p7} \Delta T_{87}$

3.4.3 Design Point Performance

A design point model using the above equations was developed on a spreadsheet so that each variable could be varied at will and trends produced. The spreadsheet is included in appendix A. The base hypothetical engine chosen had the following characteristics;

Ambient	ISO conditions	101.325 kPa	(P_{AMB})
		15.0 °C	(T_{AMB})
Inlet	Mass flow	150 kg/s	($\dot{m}_{in}$)
	Pressure loss	1.0 %	(X_{P01})
Compressor	Pressure ratio	12	(r_{P12})
	Isentropic efficiency	86 %	(η_{1c})
Bleed	Cooling and sealing	5.0 %	(X_{C29})
Combustor	Pressure loss	5.0 %	(X_{P34})
	Fuel LCV	42000 kJ/kg	
	Combustion efficiency	100 %	(η_{comb})
	TET	1400 K	(TET)
Turbines	Isentropic efficiency	88 %	(η_{it})
	Cooling flow split	2/3 before first turbine 1/3 after first turbine	(X_{C95})
Shaft	Mechanical efficiency	99.5 %	(η_{MECH})

These values give the following results

Thermal efficiency	34.0 %
Power output	45.6 MW
Fuel flow	3.19 kg/s
Exhaust temperature	832 K (559 °C)

This engine is not intended to be a real situation, but rather a base

Figure 3.11 Design Point - Efficiency vs Pressure Ratio

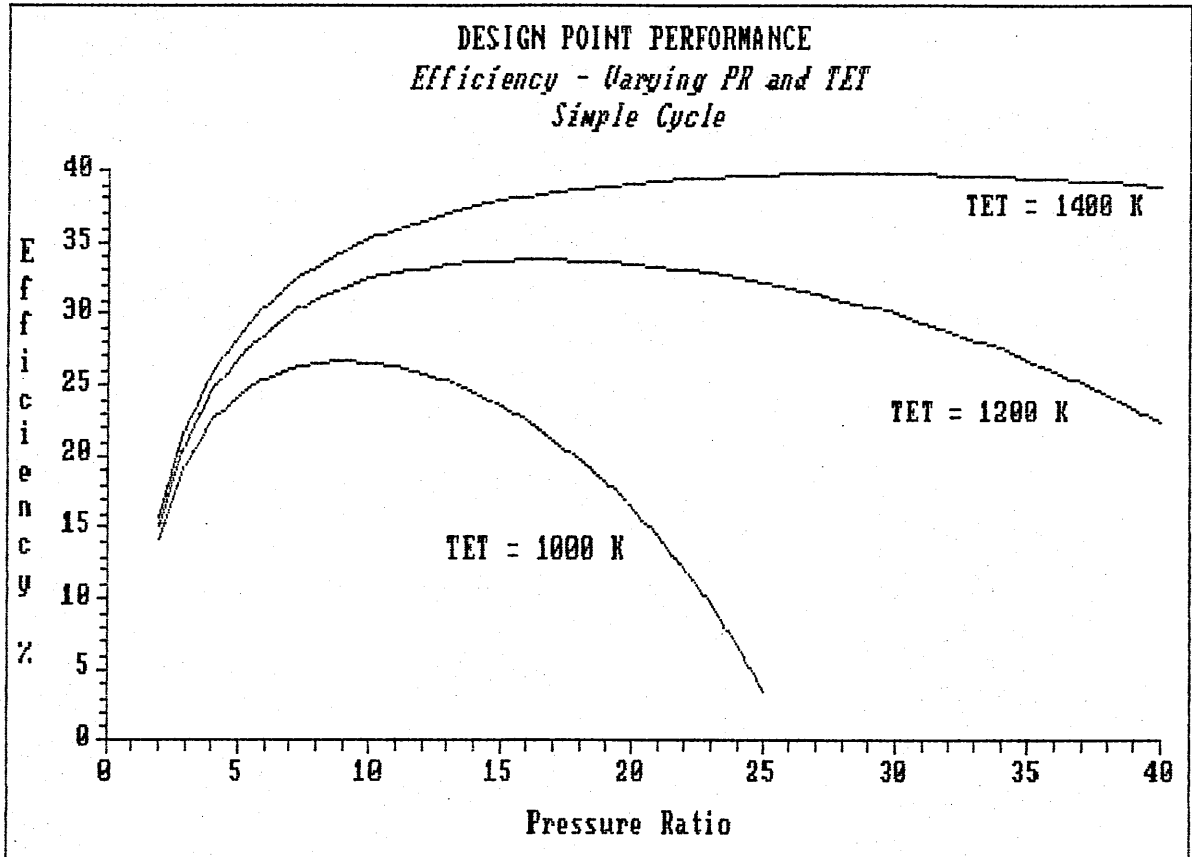


Figure 3.12 Design Point - Power vs Pressure Ratio

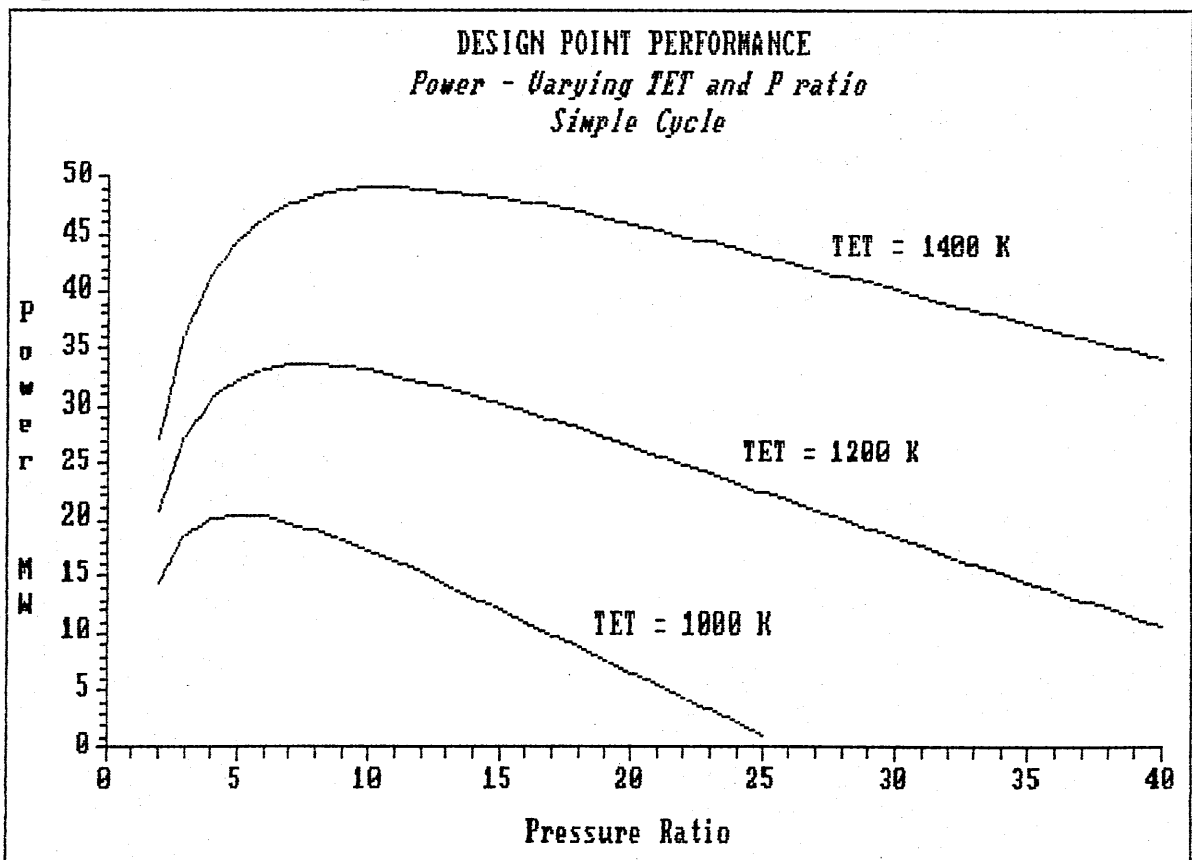


Figure 3.13 Design Point - Efficiency vs TET

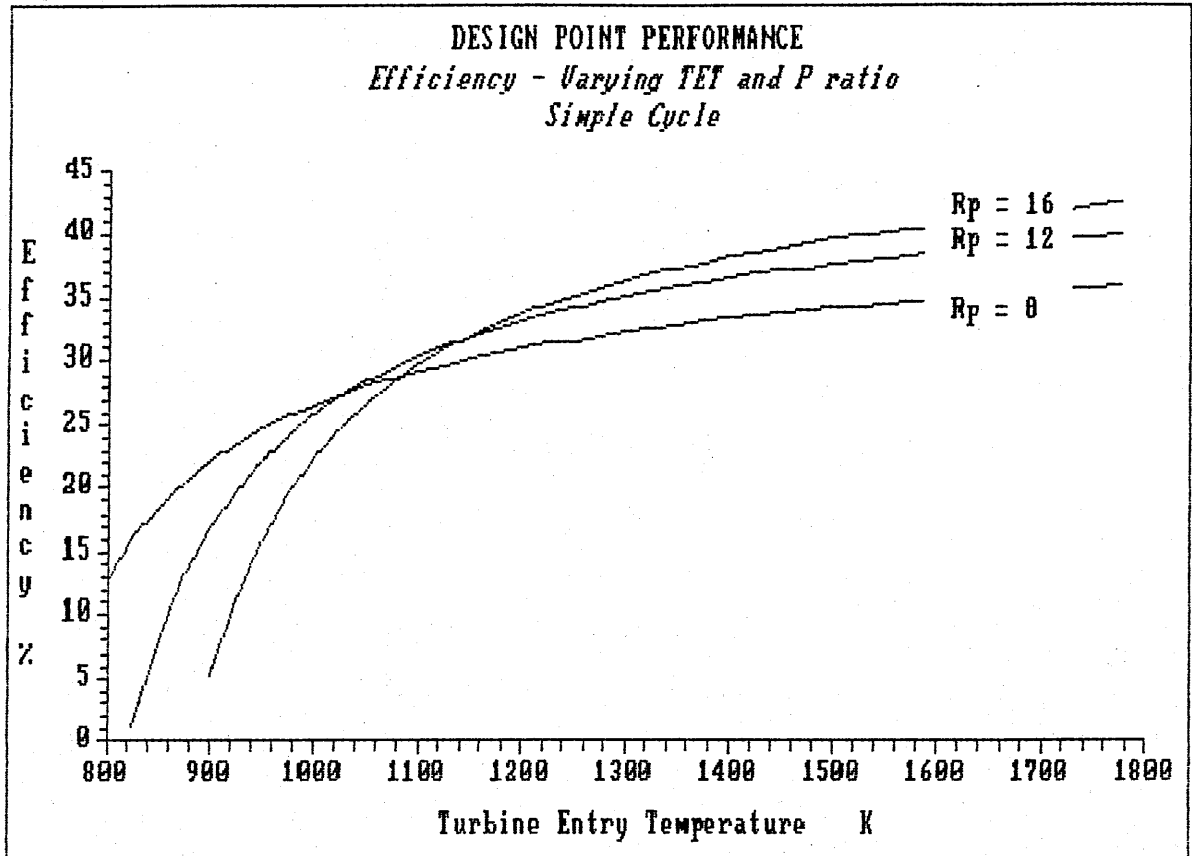
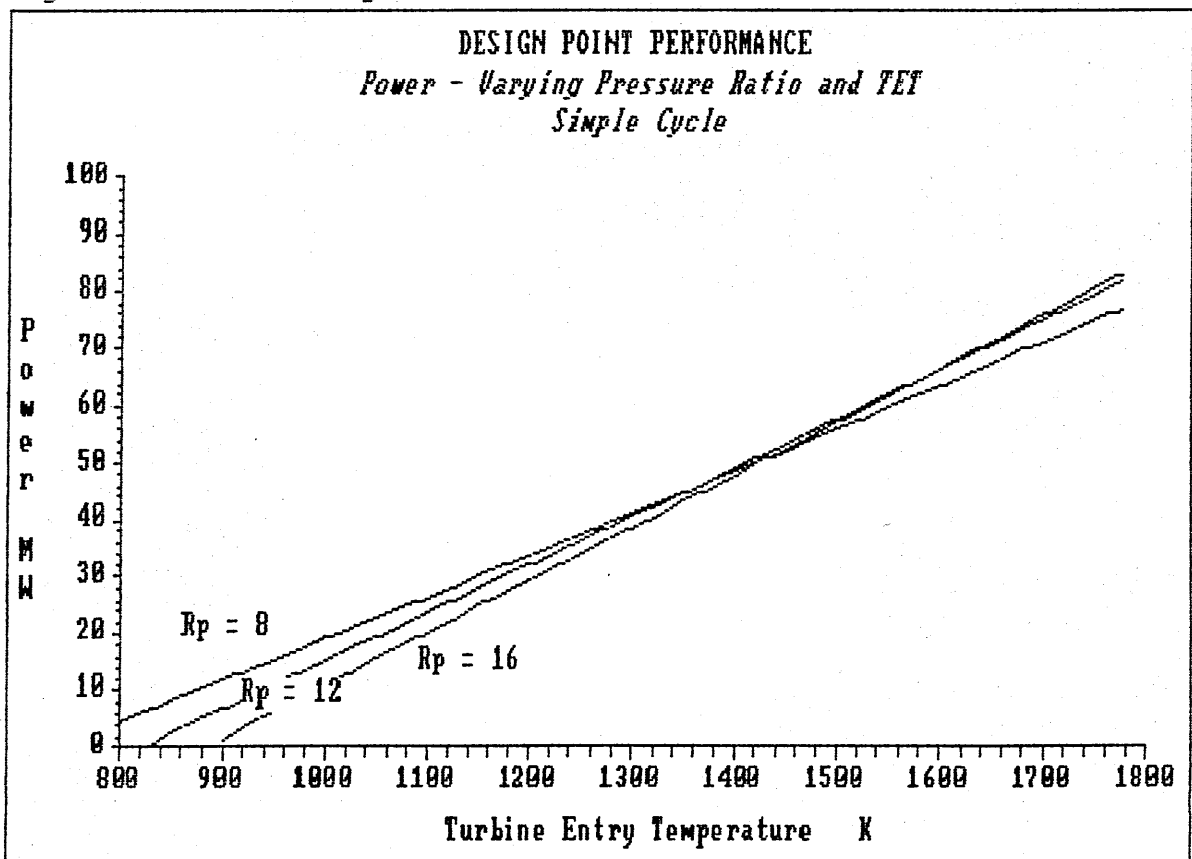


Figure 3.14 Design Point - Power vs TET



for varying parameters to study their effects. Section 8 contains plots attempting to model a real engine. The most important parameters are the pressure ratio and the TET.

The pressure ratio and TET were varied while the other variables were held constant. The power is the power per 150 kg/s air mass flow and is thus a measure of specific work. The power plots are thus also specific work plots. Cycle variation can be seen in the cycle diagrams of figures 3.8 and 3.9. The results are plotted in figures 3.11 and 3.12. It can be seen the efficiency and power are both dependant on pressure ratio but do not peak at the same value. Also the peak values are different at different TET's. For both power and efficiency the higher the TET the better. The same information is plotted in figures 3.13 and 3.14 and show a near linear increase in power with TET but diminishing returns for efficiency. These graphs are sometimes combined into a single plot as in figure 3.15, which uses the value 'heat rate' instead of the efficiency. The heat rate is essentially the inverse of the efficiency. It can be seen at the current TET's available that further increases in TET will effect specific power more than the efficiency. An additional graph is included to show the exhaust temperatures in figure 3.16 for later reference.

Variables other than the TET and r_p can also be varied, such as compressor efficiency, but these are looked at more closely in section 8. Changes in the inlet and exhaust pressure losses are more correctly an off design performance problem, and would give false results varied in the design point mathematics. These are included in section 3.4.5. It will be seen in the combined cycle section that none of the losses due to inefficiencies result in an improved bottoming cycle performance sufficient to compensate for the loss in gas turbine performance.

In reality isentropic efficiency η_{ic} decreases and η_{it} increases as design r_p increases. This is because compression is in a series of small steps, each with a small amount of preheat due to the losses in the previous stages. The effect is detrimental to compression, but in the turbine the heating is partially recovered. This is why the polytropic efficiency definition is often used as it is unaffected by pressure ratio. Polytropic efficiencies are used in the model of section 8.

In real engine design the cooling flow will need to be increased for increasing TET given that the cooling methods and materials remain the same. Raising the TET raises the efficiency and specific work, but the surfaces of the exposed components must be maintained below a certain safe temperature limit consistent with mechanical stability and corrosion resistance. As the TET increases more intensive cooling is required and a larger proportion of the turbine requires it. Thus the compressor bleed flow must be increased and eventually a TET is reached beyond which the cycle efficiency starts to fall for further increases in TET and cooling flow. Current aero-derivatives use about 20% and industrials 10% of the compressor flow for cooling. The peak in specific power is usually at a higher TET than the maximum efficiency TET. The previous theoretical results are for the assumption that increased TET is a response to increased material properties or cooling techniques rather than use of more cooling air.

In a cycle using exhaust heat recovery the apparent increase in efficiency with pressure ratio is misleading. The efficiency increases

Figure 3.15 Design Point - Heat Rate vs Specific Power

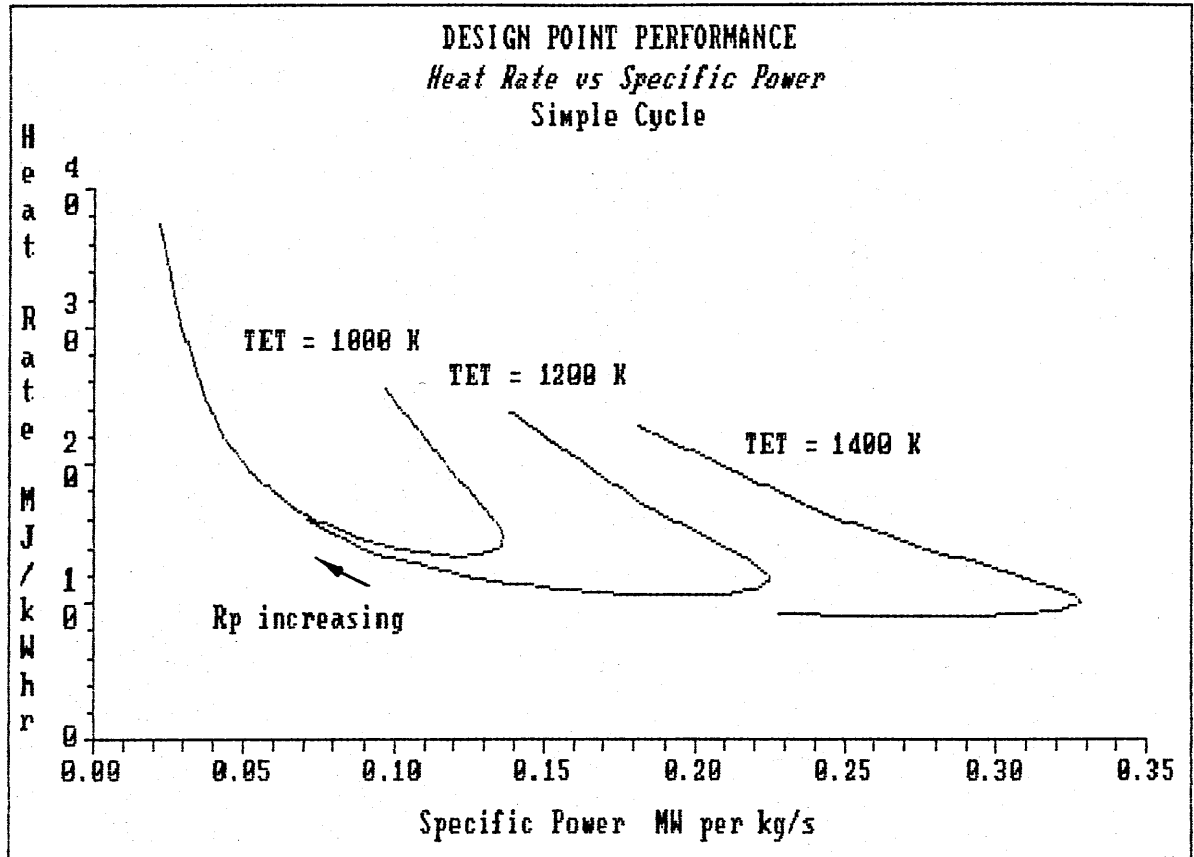
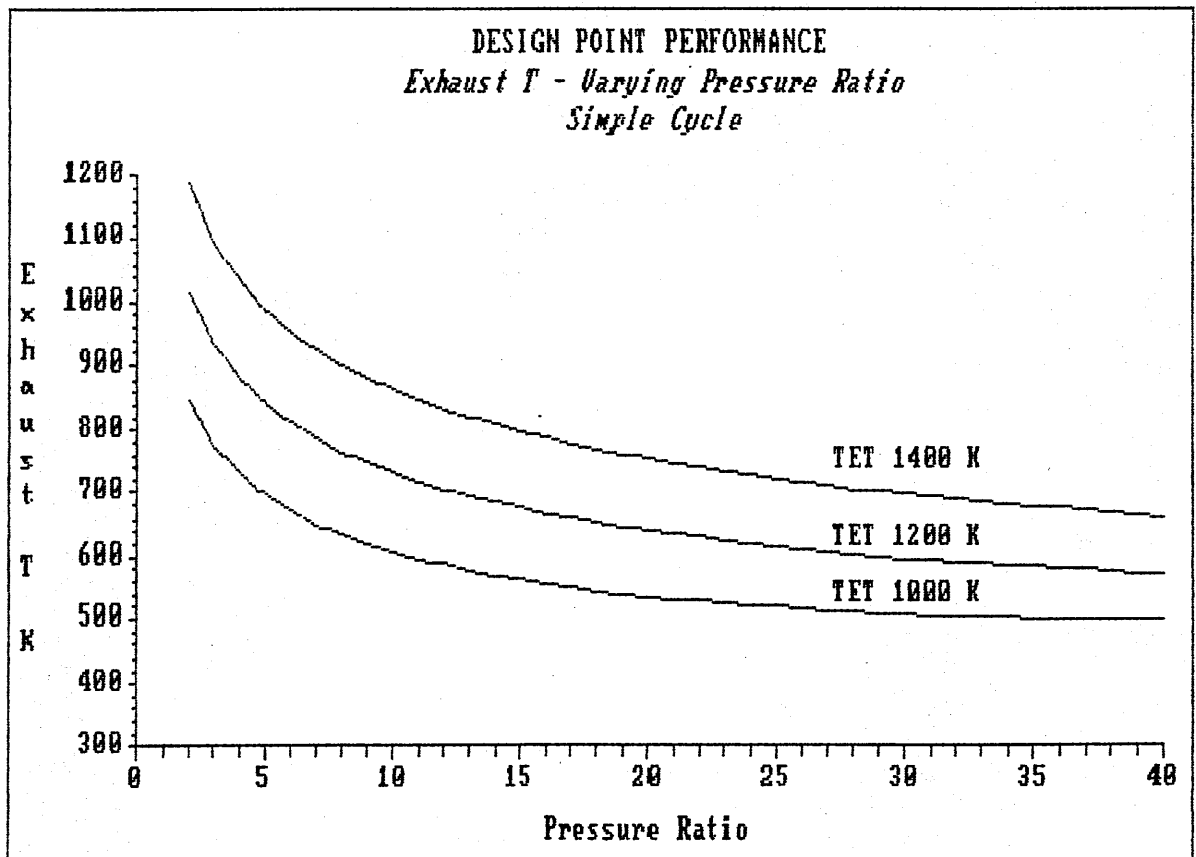


Figure 3.16 Design Point - Exhaust Temperature vs Pressure Ratio



with increased pressure ratio as the work increases, but the dissipated work becomes increasingly unrecoverable with declining temperatures. Thus a high pressure ratio may mean a smaller fraction of lost work (inefficiencies) and rejected heat is recovered as the low exhaust temperature reduces exhaust recovery effectiveness.

3.4.4 Real Engine Data

The theory can be compared to reality by use of manufacturers published data. A data base has been compiled in appendix D of engine data. The data represents electrical generation at ISO conditions burning natural gas, with the efficiency calculated at the bus bar. Plots can be seen in figures 3.17, 3.18, 3.19 and 3.20. No effort was made to adjust each plot to factor out the effect of other variables. Comments on the accuracy of quoted engine performance data is given in sections 3.6, 8.4.1 and reference [49].

It can be seen from figure 3.17 that as expected there is a strong relationship of efficiency to pressure ratio, and that this is similar to figure 3.11. The effect of TET on efficiency is shown in figure 3.18. The TET effect is not so pronounced as it is eclipsed by the pressure ratio effect and is also a compromise with cooling flow.

The size and type effects can be seen in figure 3.19. Here the aero-derivative group are clearly seen. Most of the size effects are masked by the fact that the pressure ratio is also a function of size, as can be seen in figure 3.20. It can be clearly seen that the industrial heavy duty engine is generally of a lower pressure ratio and a lower efficiency. The industrial group increases in efficiency at close to 3% per 100 MW.

Real TET's are a compromise between the cooling flow, the material cost and the temperature. This means the TET is a poor comparison between engines without taking into consideration the other parameters. A table of parameters is included in table 3.1. These figures give an indication of the main industrial engines, but is from a different and older source than the other plots thus gives slightly lower performance.

3.4.5 Off Design Performance

Once an engine is built its geometry is fixed. When this engine is then operated under different loads and speeds from the design point it is said to be running off design.

The compressor has a characteristic performance map, which shows how it reacts to various flows and speeds. Similarly the turbine has its own performance map. Typical maps are shown in figure 3.21 and 3.22. When the gas turbine is at equilibrium, neither accelerating nor decelerating the work levels and map characteristics must match. The locus of equilibrium operating points is called the "running line", and varies with the engine configuration.

The off-design characteristics are dependant on the configuration of the engine. A single shaft engine has the compressor synchronised to the generator load. As the load varies the speed is controlled constant, and the equilibrium line corresponds to a constant speed line. As a large proportion of the turbine load is removed for starting the running line

Table 3.1 Comparison of Gas Turbine Performance Parameters

ISO PEAK-LOAD		Installed Performance										
		LM5000	FT50	LM2500	FT9	RB211	W501	FT4E	PG7981	FT4C	GT55	PG5341
Output	MW	36.4	110.8	18.2	23.8	18.8	94.8	32.9	77.0	31.3	75.0	25.6
Efficiency LCV	%	36.4	35.8	34.8	33.8	33.2	31.9	32.4	32.1	31.4	30.5	27.9
TET	K	1561	1550	1478	1430	1366	1389	1358	1414	1350	1394	1255
Av. Metal T	K	1089	1089	1089	1047	1089	1089	1055	1144	1055	1153	1033
Pressure Ratio	-	31.5	17.9	17.0	18.5	20.0	12.1	14.7	11.5	13.8	10.5	10.2
Cooling Flow	%	21.0	21.0	20.0	20.0	18.0	10.0	10.0	13.0	11.0	-	10.0
Exhaust Flow	kg/s	135.6	368.8	65.3	101.6	83.9	326.1	138.3	255.8	137.4	278.1	118.4
Exhaust T	K	714	778	789	711	-	839	739	844	737	839	755
Length	m	6.42	8.99	5.50	6.29	2.46	7.13	6.63	24.61	7.00	20.54	12.84
Width	m	1.80	2.70	1.80	2.44	1.31	3.85	2.65	7.45	2.65	10.27	2.78
Weight	kg	9.07	112.49	4.67	9.66	-	122.47	13.88	128.59	13.88	113.40	84.37
ISO BASE-LOAD		LM5000	FT50	LM2500	FT9	RB211	W501	FT4E	PG7981	FT4C	GT55	PG5341
Output	MW	32.1	104.4	17.0	23.8	18.8	92.6	29.1	71.6	27.6	69.3	23.8
Efficiency LCV	%	36.2	35.7	33.8	33.8	33.2	32.1	32.0	32.0	30.6	30.3	27.3
TET	K	1450	1522	1444	1430	1366	1411	1302	1358	1294	1339	1216
Av. Metal T	K	1072	1019	1072	1047	1089	1072	1022	1128	1022	1100	1005
Pressure Ratio	-	31.0	17.3	16.5	18.5	20.0	12.2	13.8	11.3	13.0	10.3	10.0
Cooling Flow	%	21.0	21.0	20.0	20.0	18.0	10.0	10.0	13.0	11.0	-	10.0
Exhaust Flow	kg/s	131.5	361.1	61.2	101.6	83.9	326.1	132.4	255.8	131.5	278.1	118.4
Exhaust T	K	700	763	772	711	-	839	714	811	714	808	758

up to full speed is low on the compressor map relative to the free gas generator running line. With a free turbine and a gas generator the pressure ratio is controlled by choking in the power turbine. The load on the gas generator is thus determined by the swallowing capacity of the power turbine in contrast to the single shaft where the running line depends on load. The running line of the gas generator compressor is higher on the compressor map. A two spool engine has different characteristics again, but is more closely related to the single spool free turbine case.

The off design behaviour of gas turbines was modelled mathematically using the Cranfield program TURBOMATCH. The same hypothetical engine as in the design point case was modelled, firstly as a single shaft engine, then with a twin shaft and free turbine and finally a two spool and twin shaft engine. The Turbomatch programs used are included in appendix C.

Part load

The first off design case is when less load is required, and the fuel flow is reduced to reduce the power. In the first study it is done with a single shaft, at constant speed such as in power generation. Figure 3.25 shows how the performance varies with load. It can be seen the efficiency rapidly reduces giving poor part load performance. This is especially so in the case of a constant speed engine, which can be best explained using figure 3.25. It can be seen that at constant speed the mass flow and pressure ratio remain high at reduced load, thus requiring

Figure 3.17 Real Data - Efficiency vs Pressure Ratio

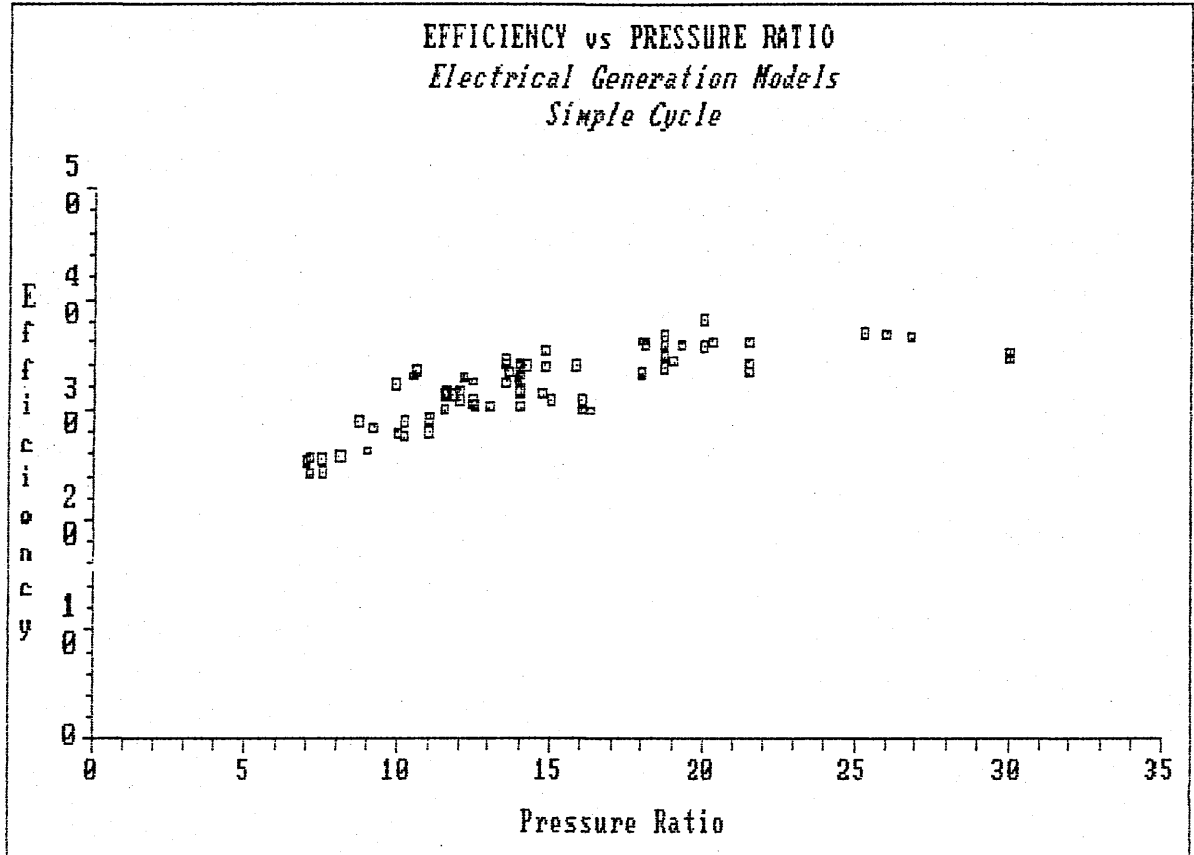


Figure 3.18 Real Data - Efficiency vs TET

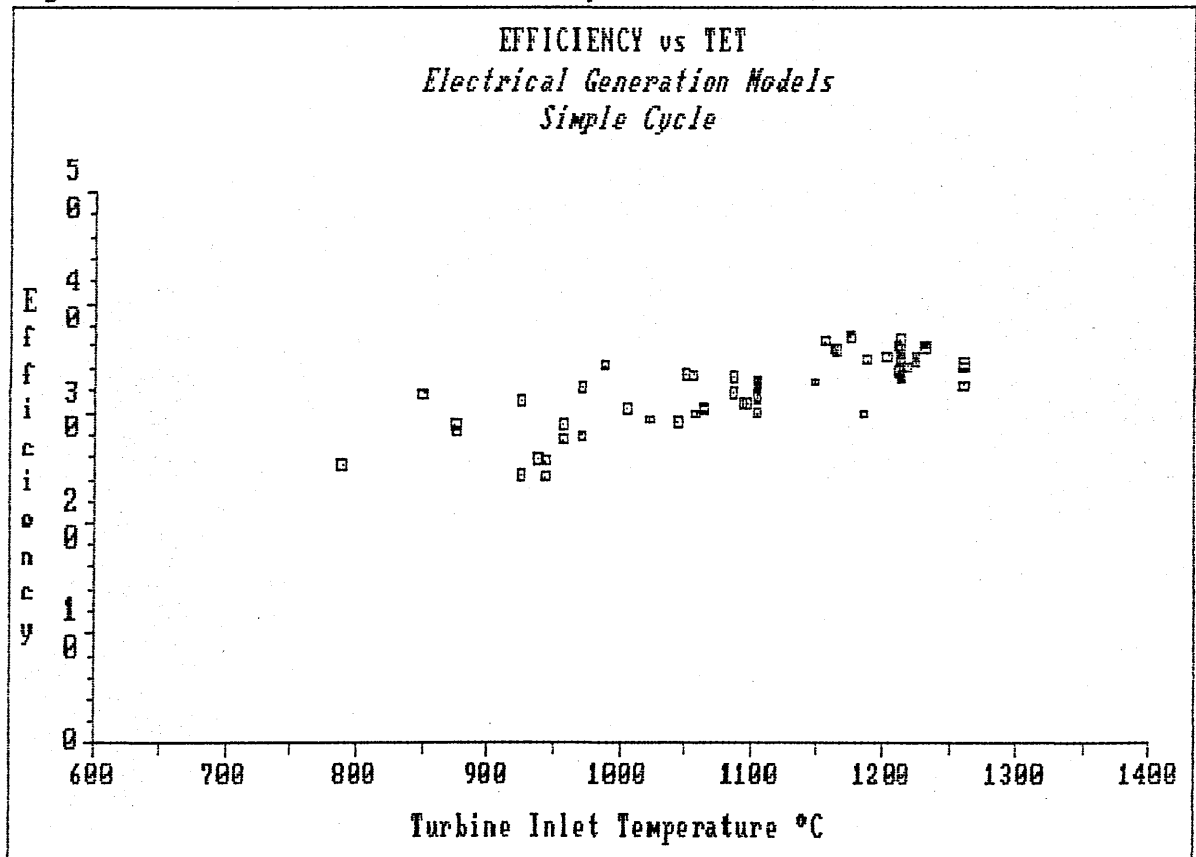


Figure 3.19 Real Data - Efficiency vs Size

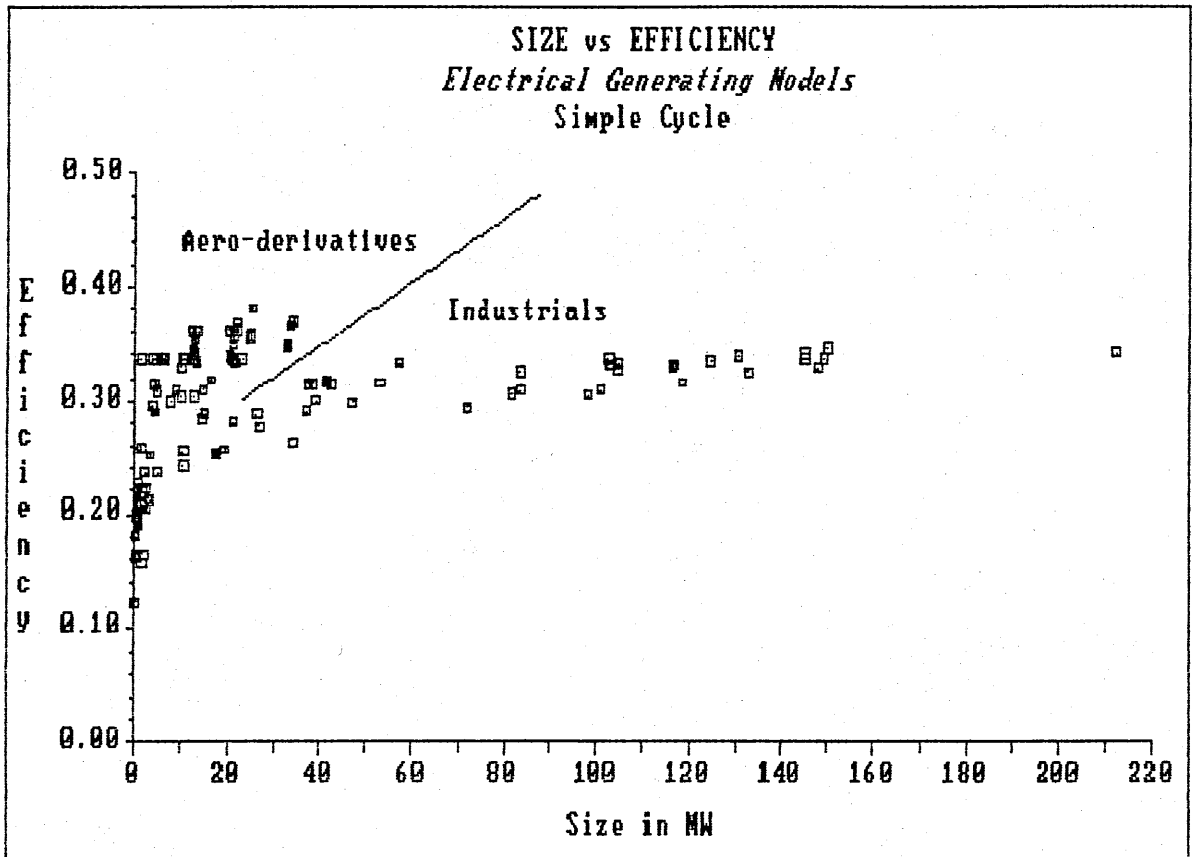


Figure 3.20 Real Data - Pressure Ratio vs Size

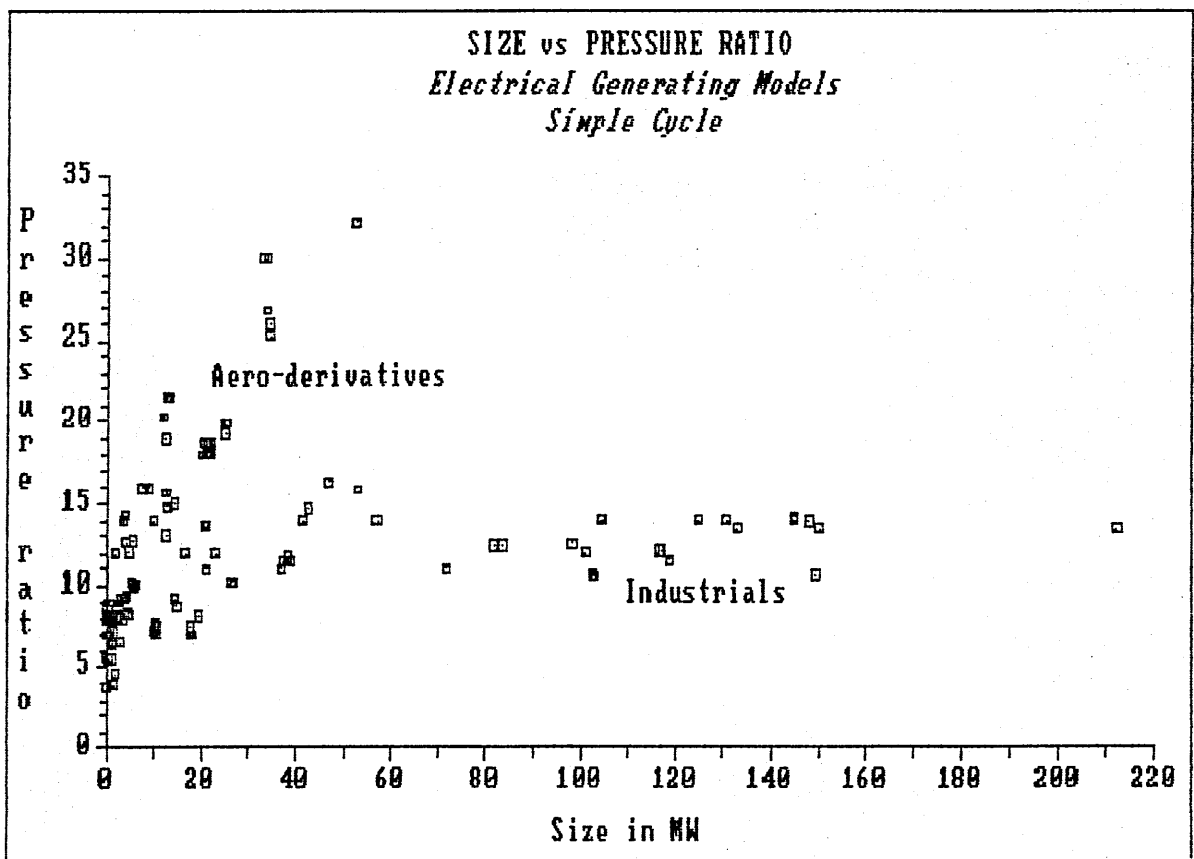


Figure 3.21 Compressor Map

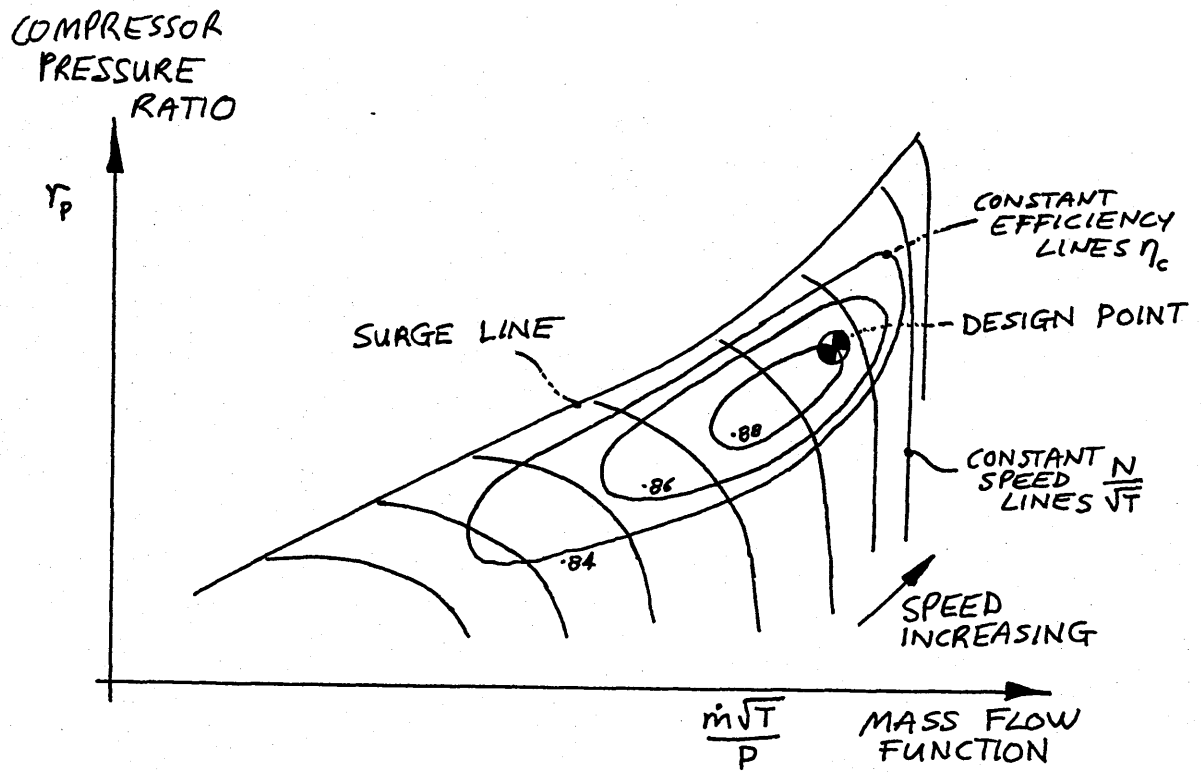


Figure 3.22 Turbine Map

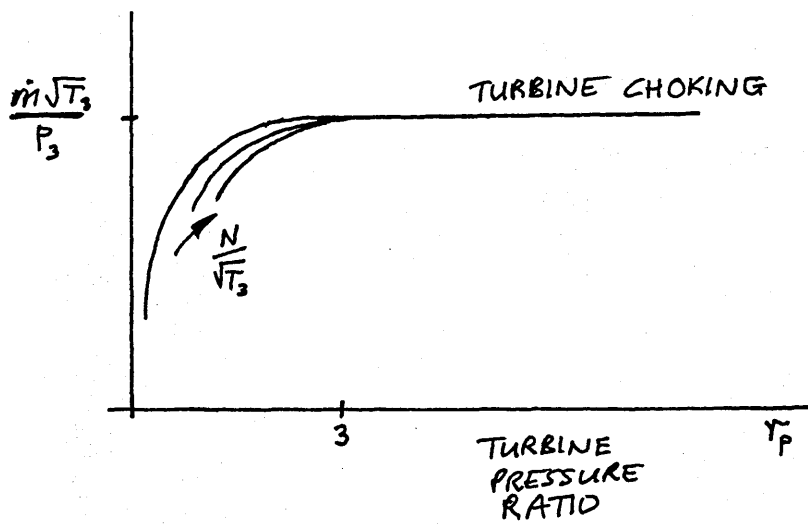
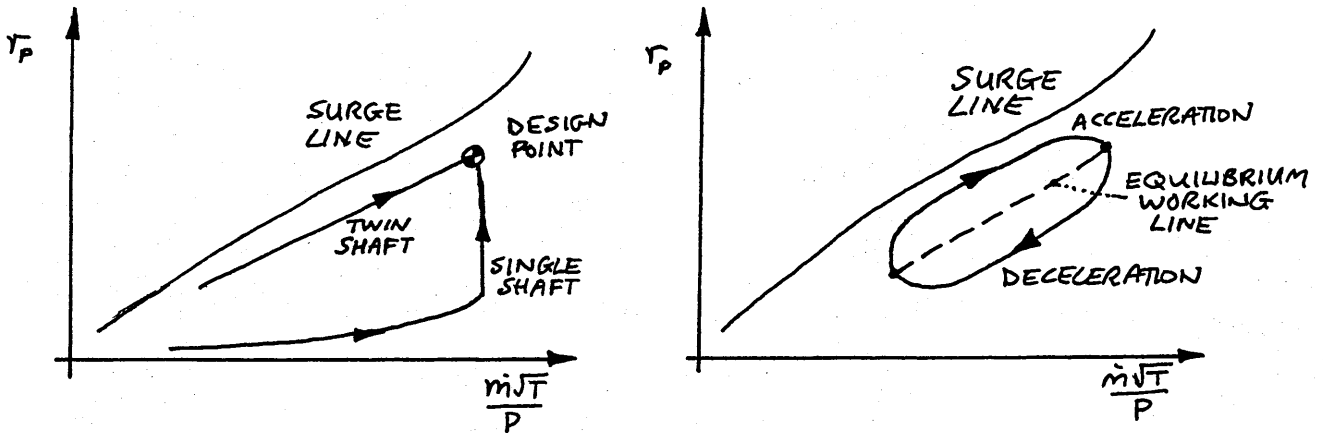


Figure 3.23 Compressor Working Lines Figure 3.24 Compressor Transients



a high level of power for compression when the engine is a low net power output. This increases the ratio of compressor work to turbine work (back work ratio) and amplifies the effect of the inefficiencies. Also a single speed engine moves down a constant speed characteristic on the compressor map (figure 3.23) and thus moves through lower compressor efficiencies more rapidly than a two shaft engine, further amplifying this effect.

Twin shaft engines with single and twin spools were modelled and the results are plotted in figures 3.26 and 3.27. It can be seen that the twin shaft engine reduces mass flow and pressure ratio much more quickly than a single shaft engine, and from figure 3.23 the gas generator compressor efficiencies are maintained higher. This holds up the thermal efficiency for longer. The twin spool three shaft engine has even faster fall off of mass flow and better efficiencies at part load. The efficiencies are compared directly in figure 3.28, and it can be seen a part load efficiency improvement of 5% can be expected at 50% load and up to 15% at 25% load. This is for simple cycles but it will be shown the effect is similar and amplified in the combined cycle, section 7.10. The comparison of exhaust temperature and mass flow between engine configurations is shown in figures 3.29 and 3.30.

Some engines use variable compressor inlet guide vanes and variable turbine guide vanes to control mass flow. For example the MS5002B uses a variable geometry turbine inlet guide vane to control mass flow and hence keeps up the temperatures at part load. This is especially important in combined cycles as will be discussed later in section 7.10.

Ambient conditions

The second off design case is when the ambient air conditions change. Figures 3.31 and 3.32 show the effect of the ambient temperature varying at fixed ambient pressure, for the two cases of holding the TET constant and holding the power constant. The performance is deteriorated at higher inlet temperatures as the lower air density reduces the mass flow, and the compression is at a higher temperature thus requiring more

work.

It can be seen in figure 3.32 that holding the power constant requires an increase in TET at high ambient temperatures, which would cause a great loss in blade life in normal conditions. The TET would need to be increased about 5% (70 K) on a 30°C day. If the 5% increase in TET corresponds to a 5% increase in metal temperature then the life of the engine may suffer unacceptably. On the other hand holding the TET constant would mean the power varying which may also be unacceptable in certain situations. To combat this the manufacturers may supply some de-rated constant power with an equivalent average life.

The inlet temperature of the turbine can be lowered by an inlet cooler to increase power. A typical method would be an evaporative cooler, which is effective in hot and low humidity climates.

Other limits to performance are the maximum pressure ratio dictated by case strength and the maximum speed dictated by blade and disk stress levels. A typical published performance graph is shown in figure 3.33 and shows the various limits and the "flat rating" for equivalent life. Efficiency is not effected by the inlet temperature as much as power is, but still has significant variation. Rather than produce two graphs for each of the other variables the rest of the off design studies are for constant TET.

In combined cycles the steam turbine is sized to suit the range of ambient temperatures expected. If the temperature drops too low the gas turbine output will need to be reduced to maintain steam flow within the steam turbine capacity, and thus efficiency falls off for both high and low ambient temperatures.

The effects of varying ambient pressure are shown in figure 3.34. It can be seen the power performance is adversely effected by low pressure as the engine swallows less mass flow. This loss of power not only occurs with weather but also if the engine is sited at high altitude. An engine sited at altitude is usually de-rated appropriately. It should be noted the efficiency is unaltered for the case of constant TET and inlet temperature, as the lower inlet density at the same temperature only effects the mass flow and not the pressure ratio or work per kilogram. Table 3.2 is included for derating factors.

Table 3.2 Altitude Derating Factors

Altitude ft	Power Factor
0	1.000
500	0.982
1000	0.964
1500	0.947
2000	0.929

Figure 3.25 Part Load - Single Shaft, Constant Speed

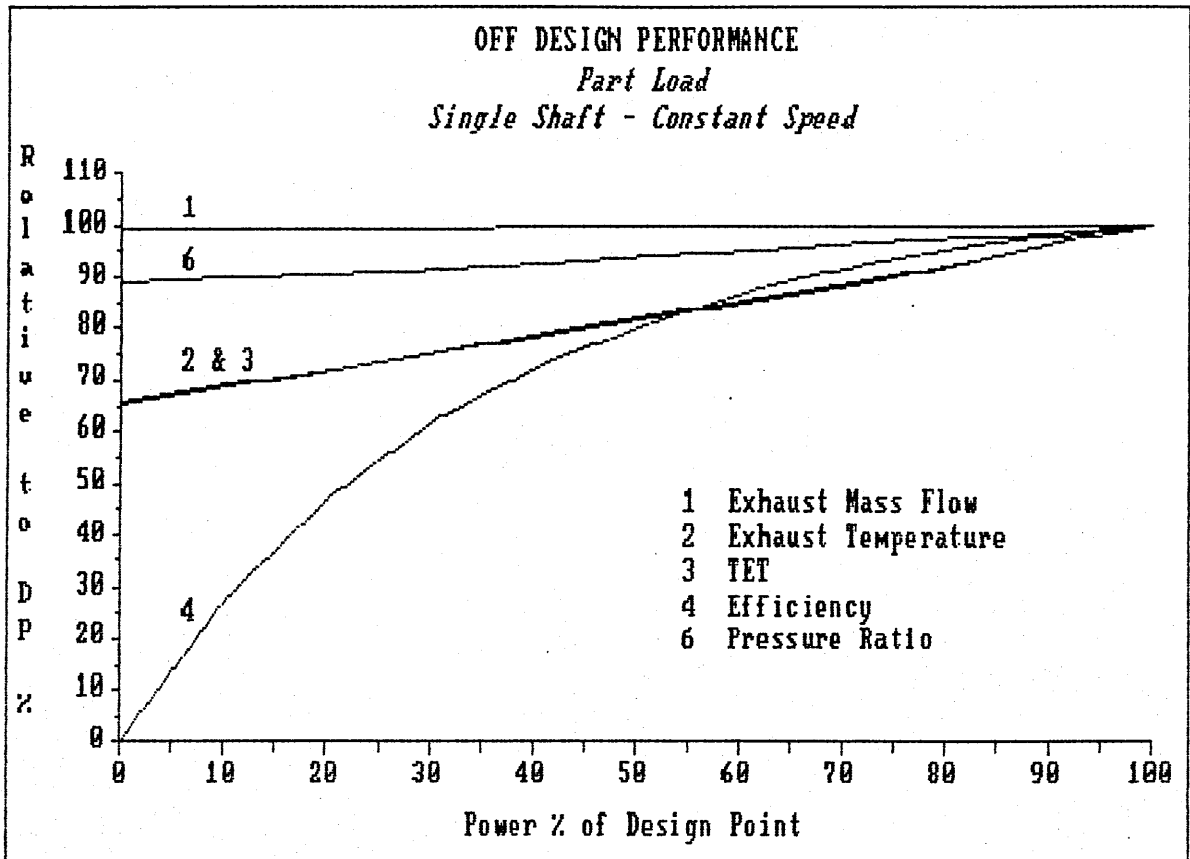


Figure 3.26 Part Load - Twin Shaft, Single Spool

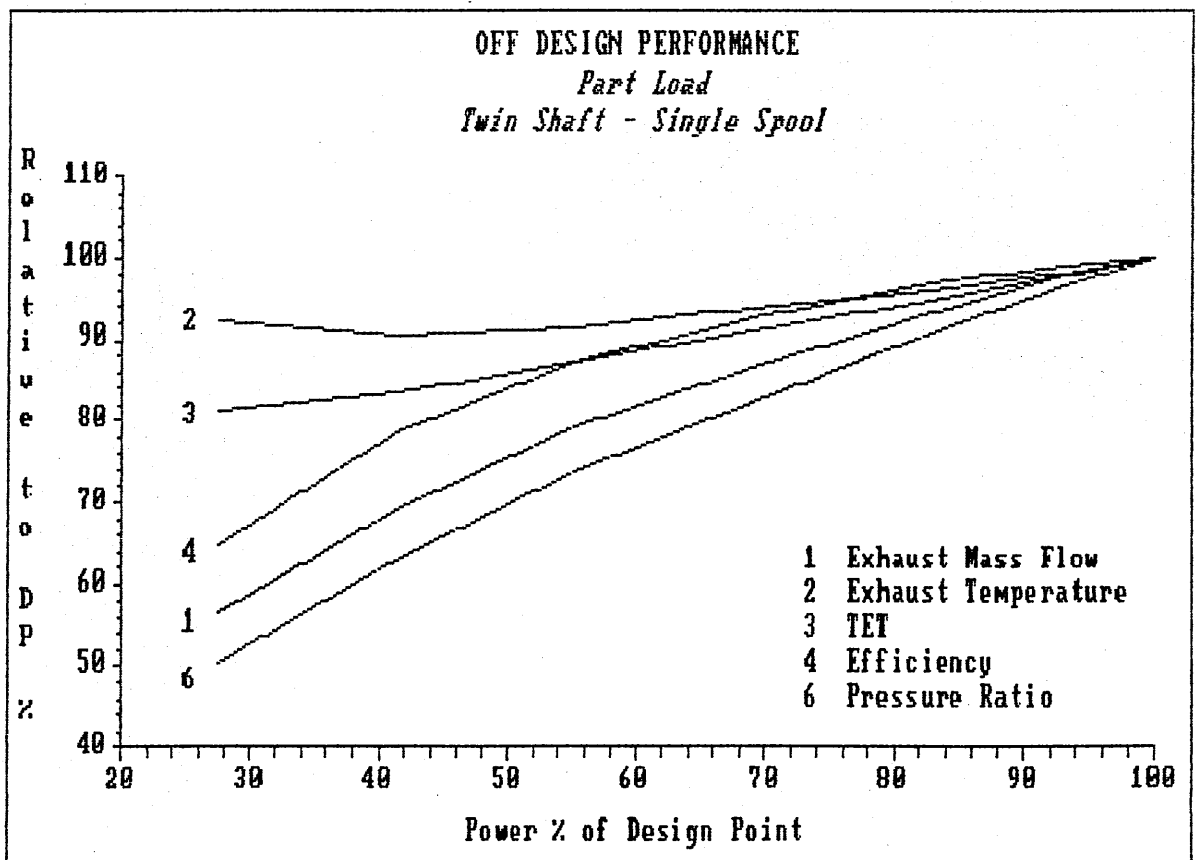


Figure 3.27 Part Load - Three Shaft, Two Spool

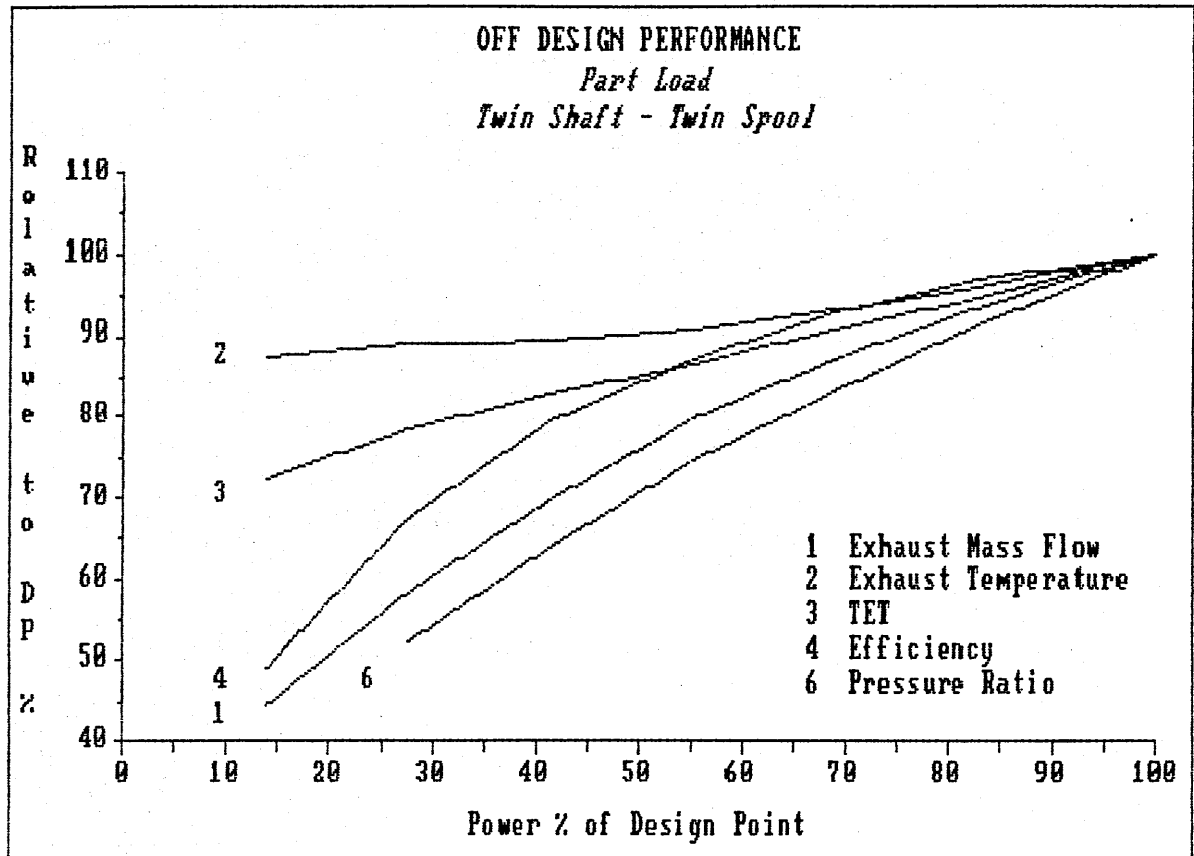


Figure 3.28 Efficiency At Part Load

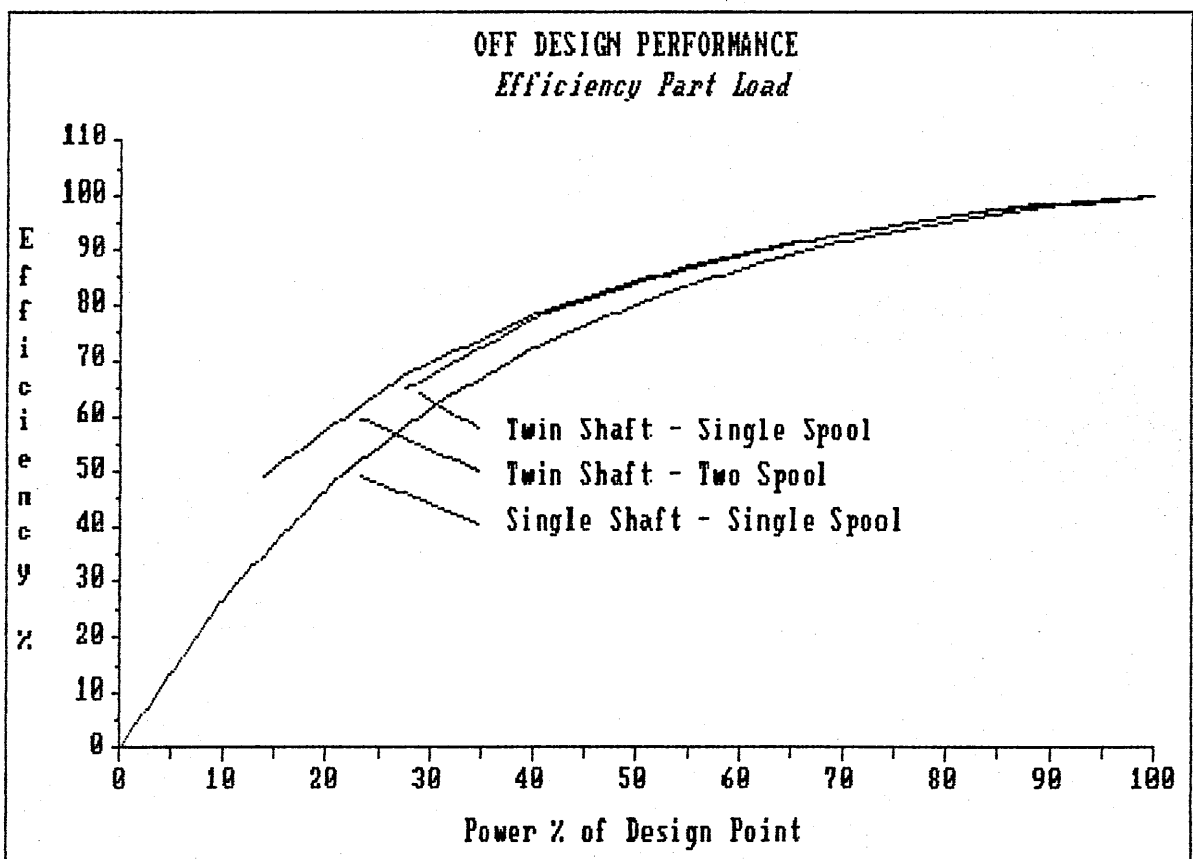


Figure 3.29 Exhaust Temperature At Part Load

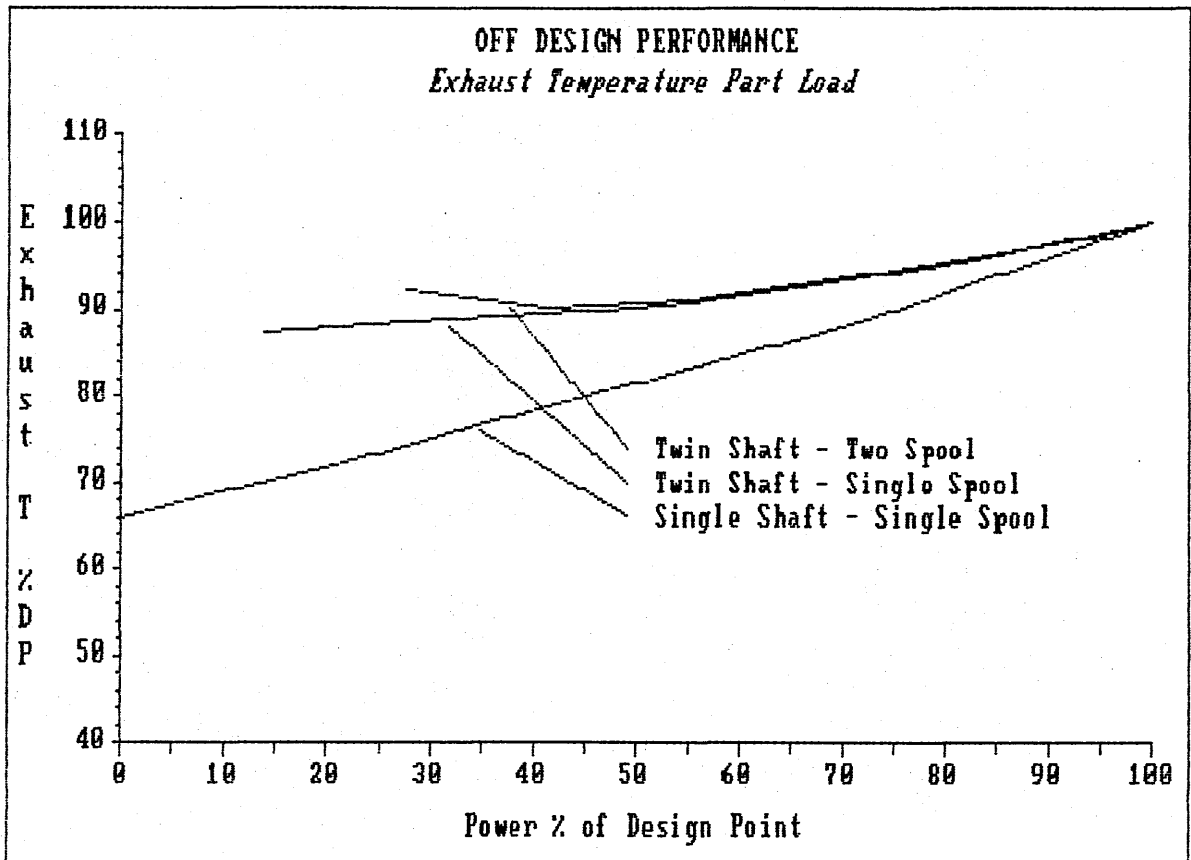


Figure 3.30 Exhaust Flow At Part Load

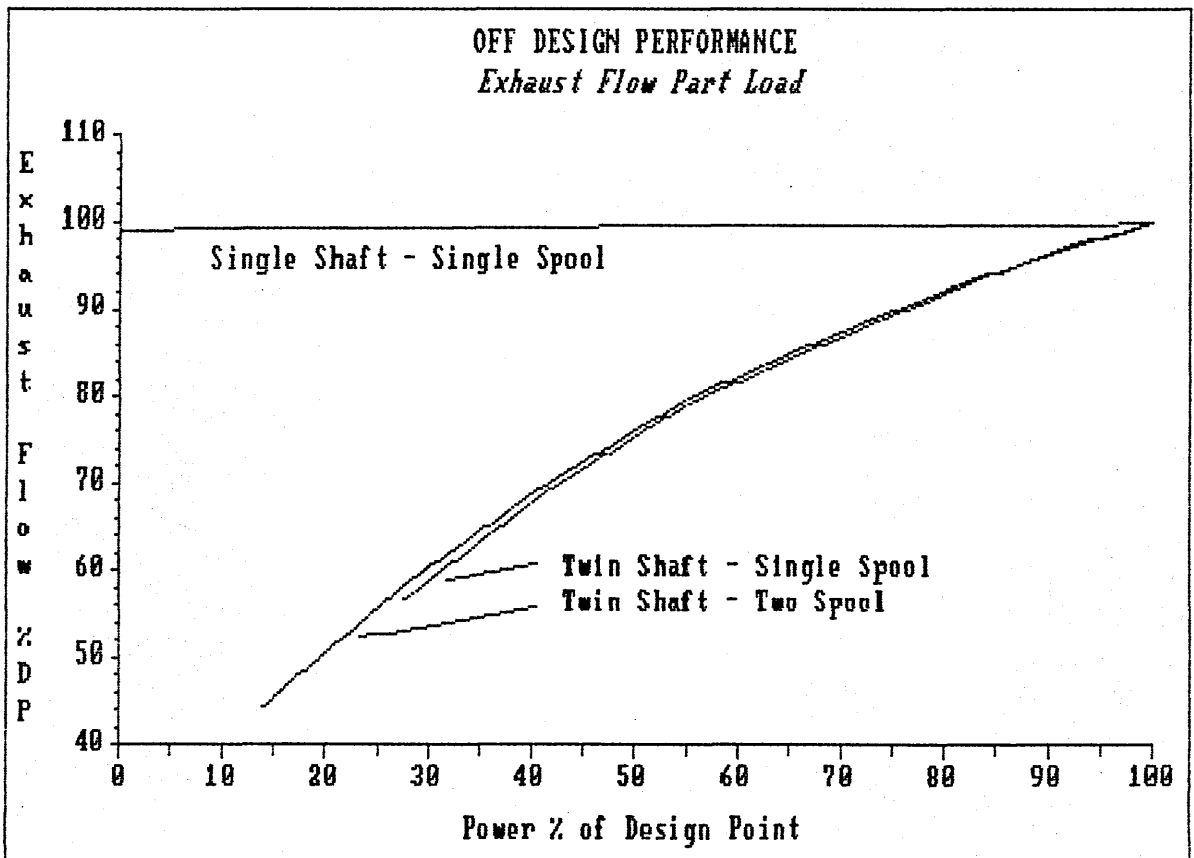


Figure 3.31 Off Design - Varying Ambient Temperature, Constant TET

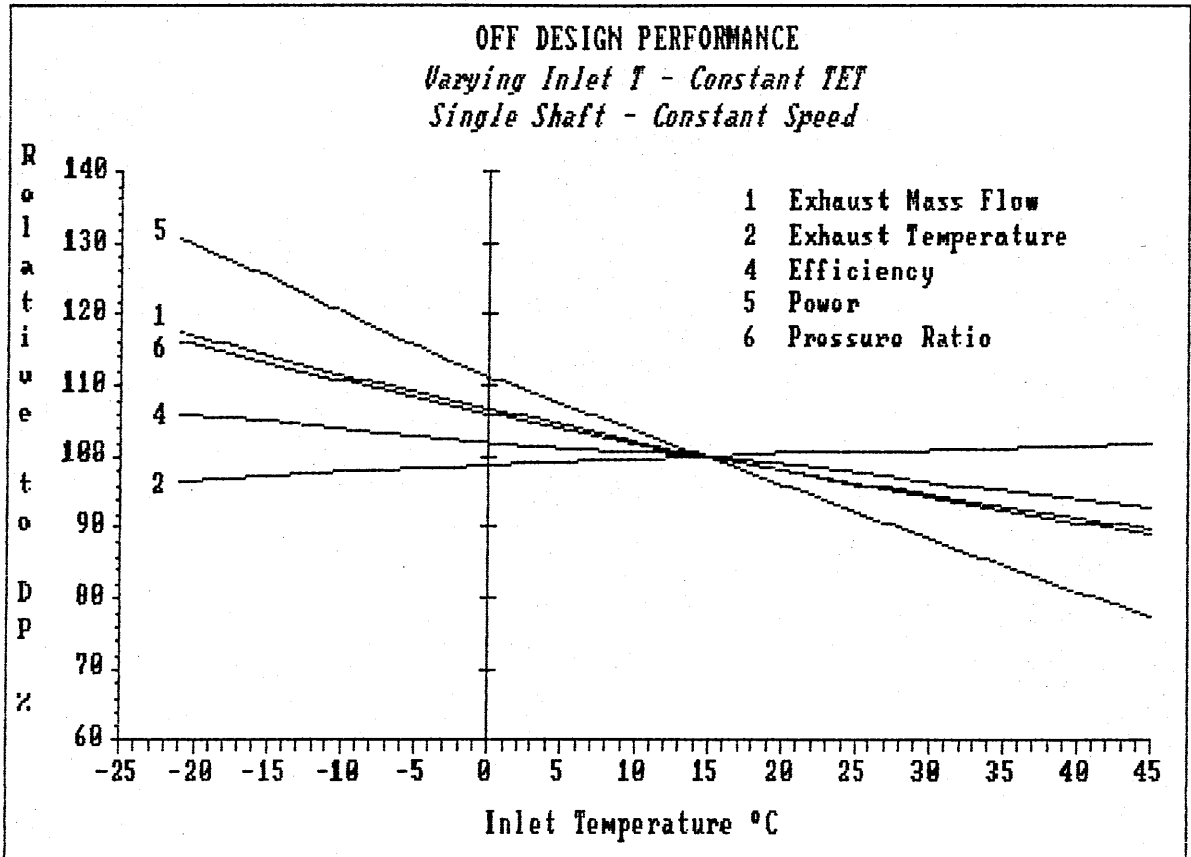


Figure 3.32 Off Design - Varying Ambient Temperature, Constant Power

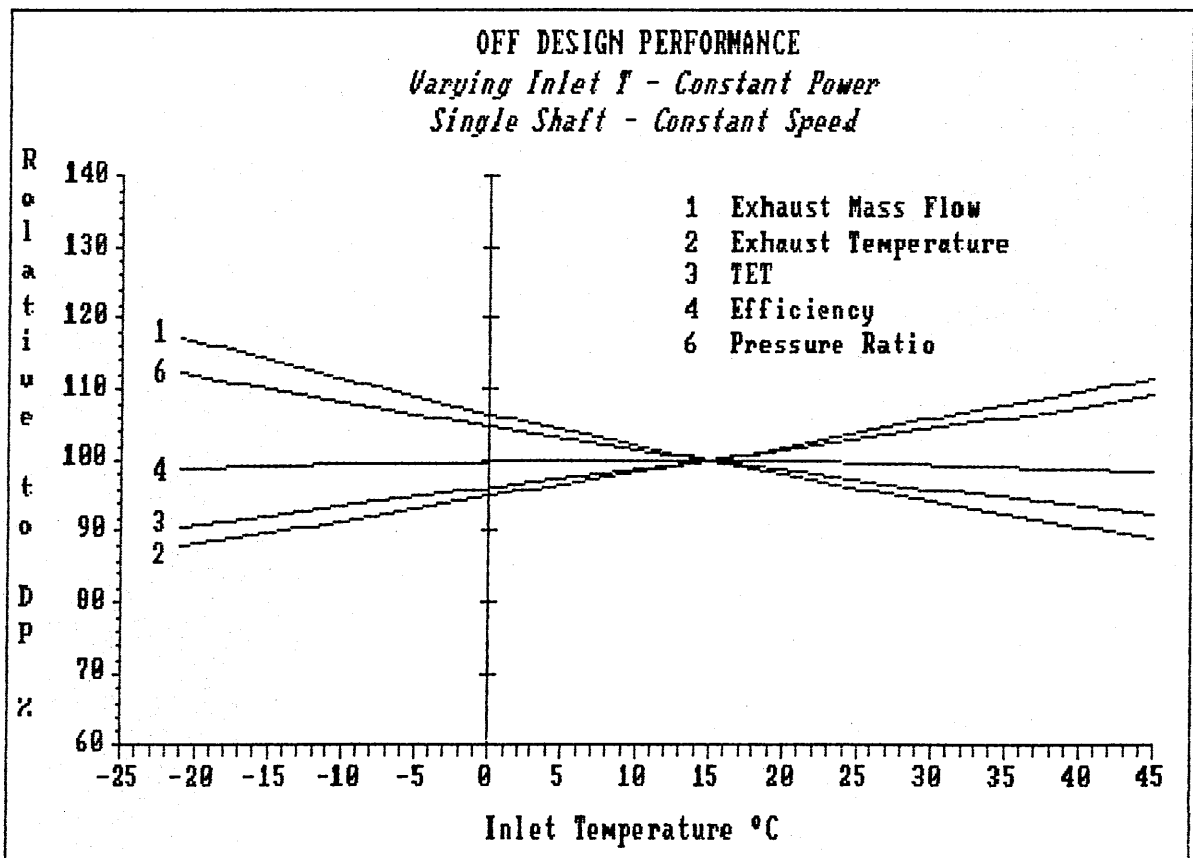
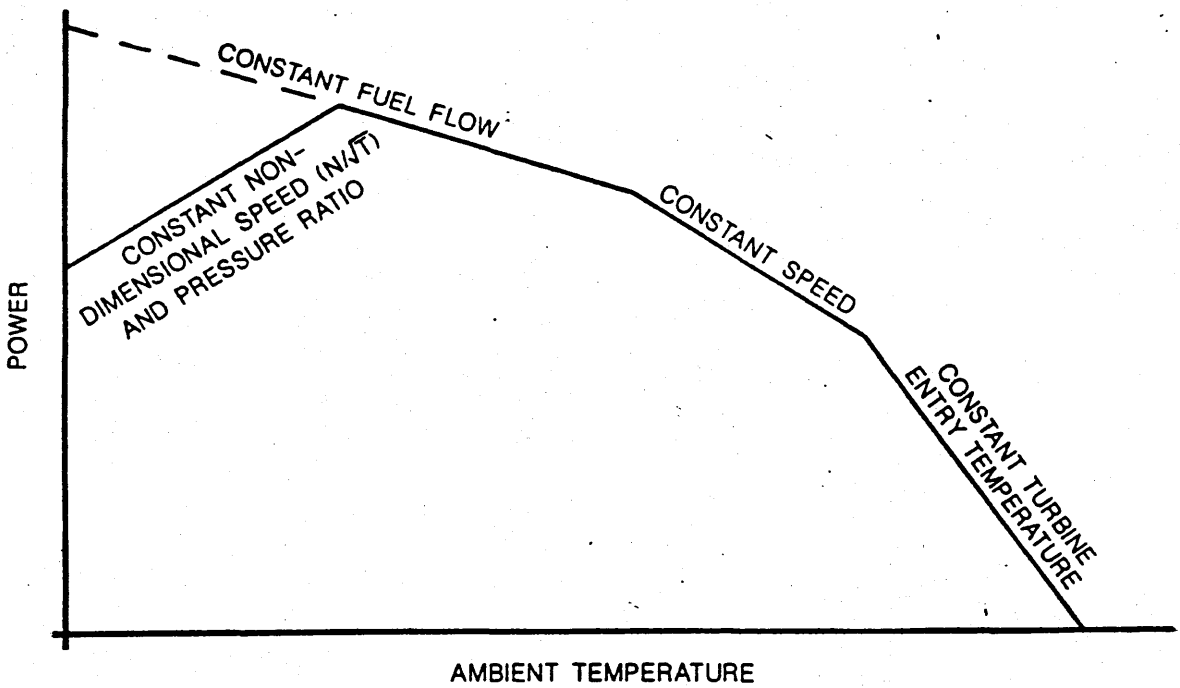


Figure 3.33 Gas Turbine Power Rating



S192/14

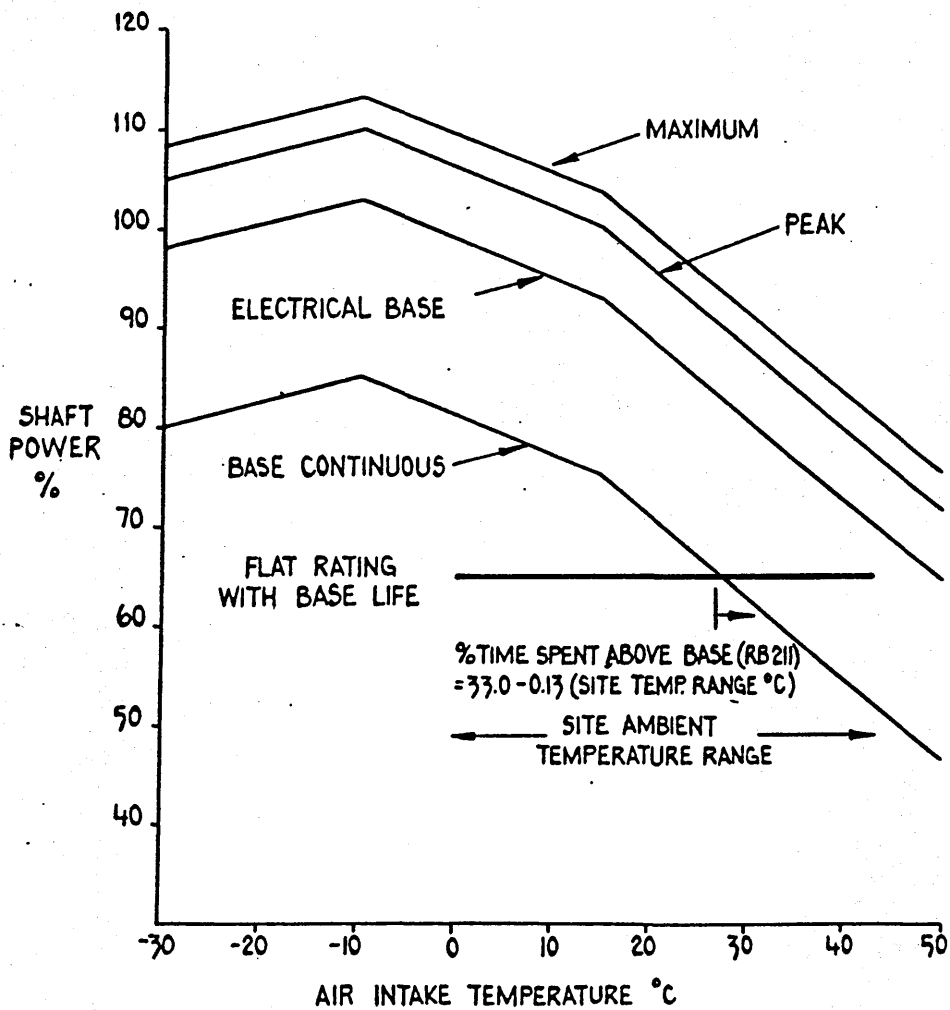


Figure 3.34 Off Design - Varying Ambient Pressure, Constant TET

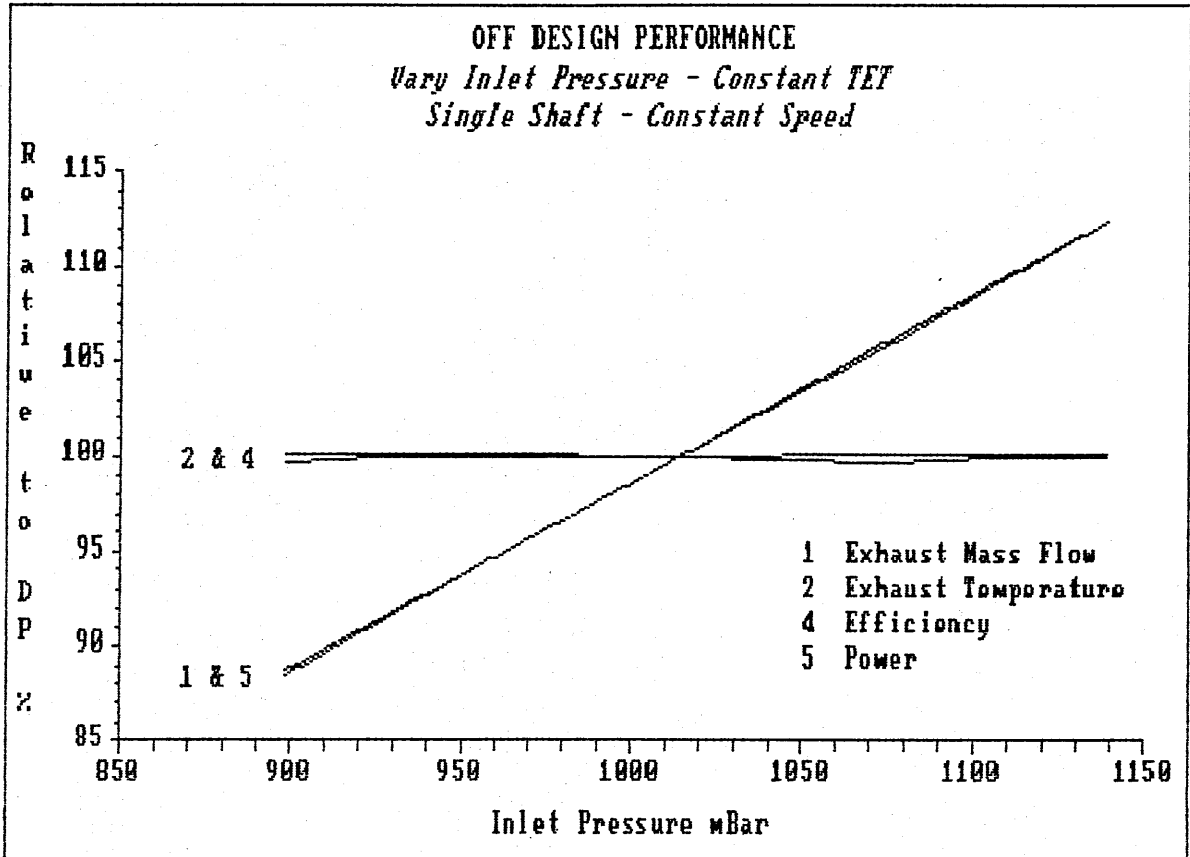


Figure 3.35 Off Design - Varying Inlet Pressure Loss

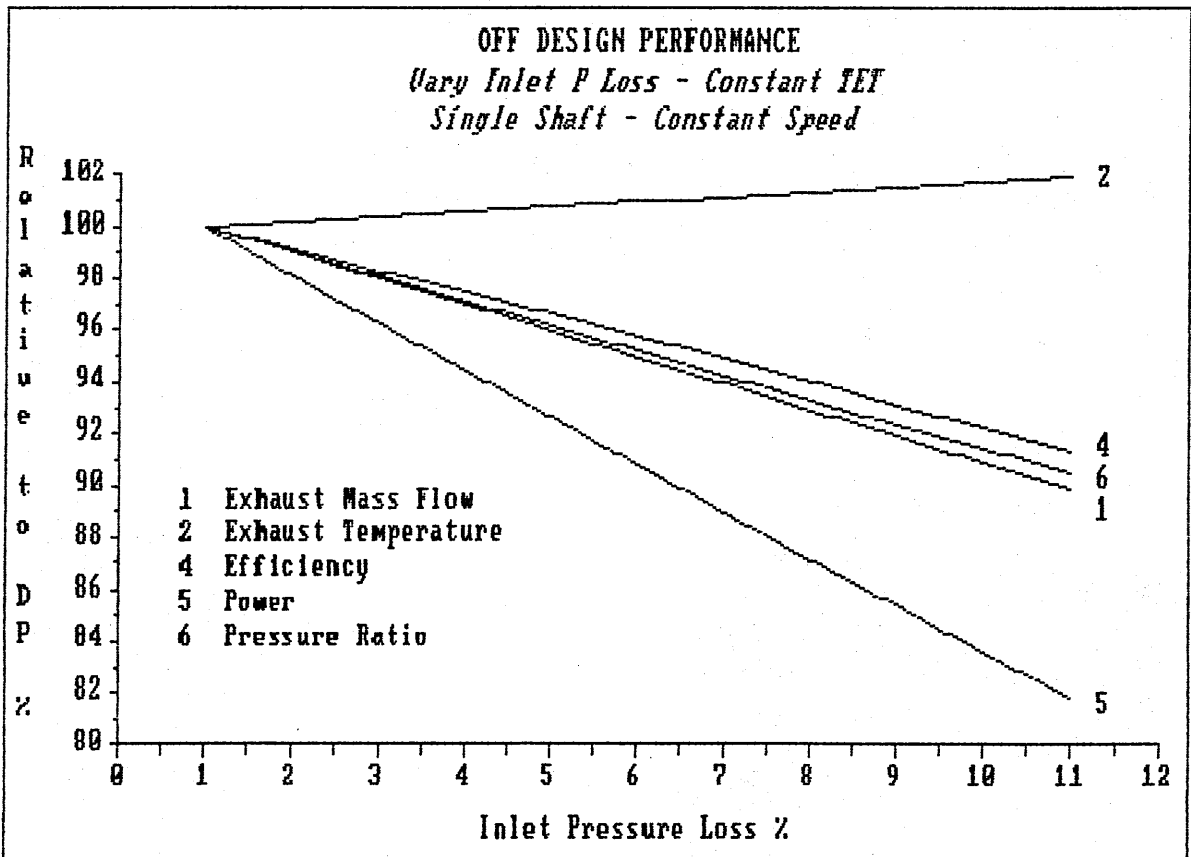


Figure 3.36 Off Design - Varying Compressor Fouling

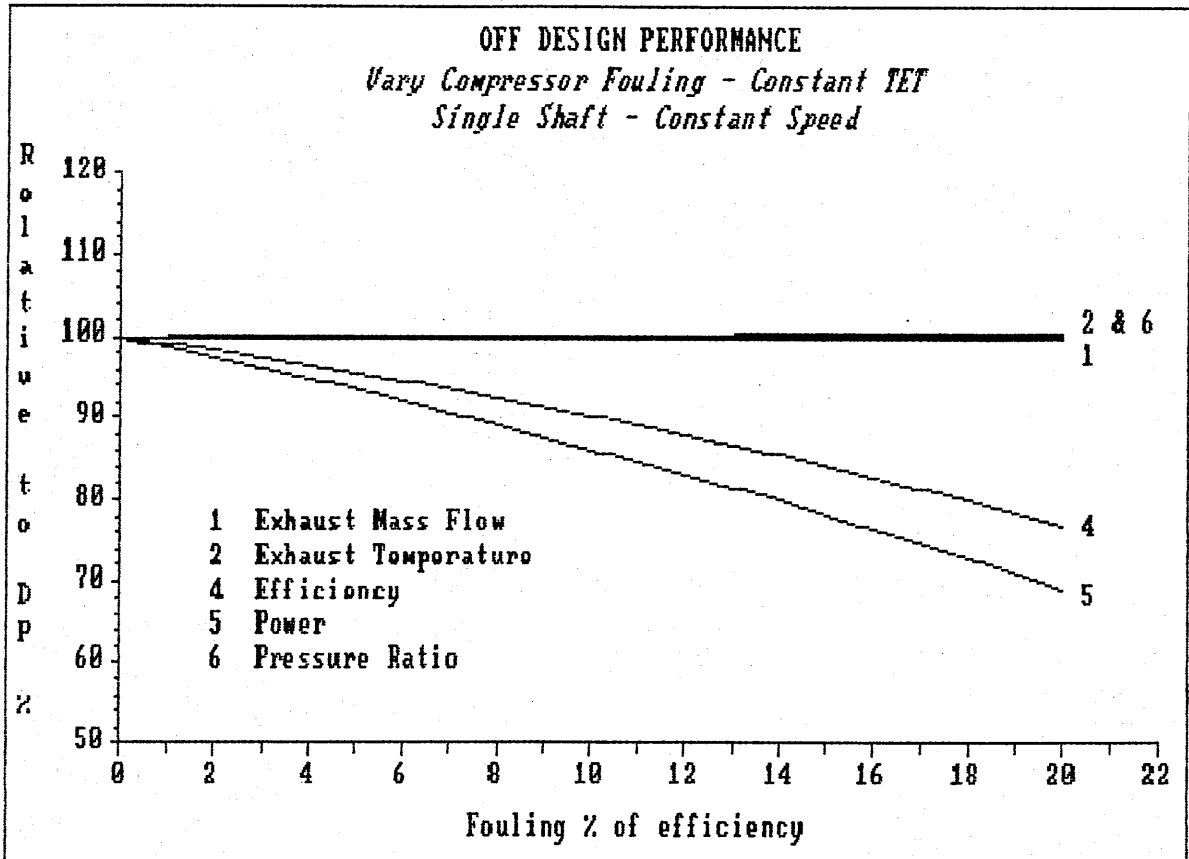
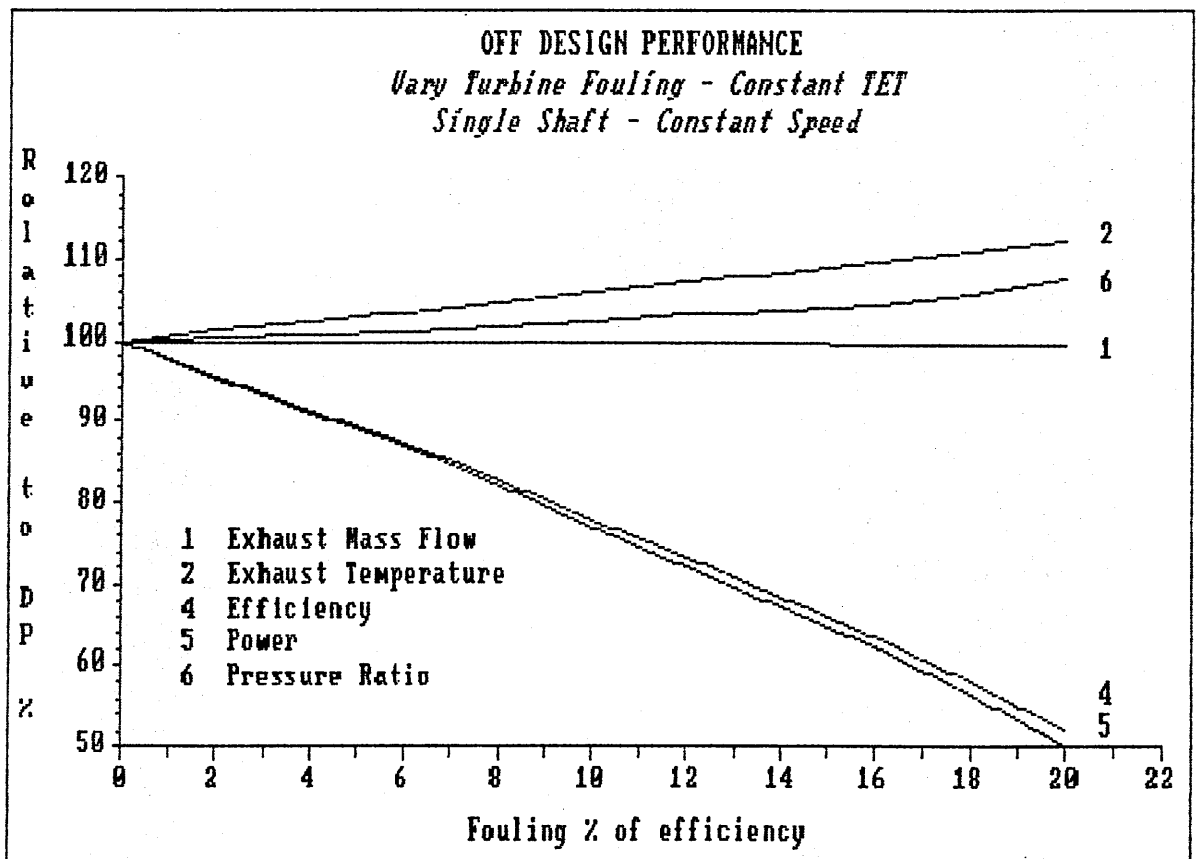


Figure 3.37 Off Design - Varying Turbine Fouling



Inlet and Exhaust Losses

Figure 3.35 shows the effect of inlet pressure loss on performance, such as inlet ducting, filter blockage or of fitting finer filters with a higher pressure drop. Normally the filter design pressure drop is about 0.5 to 1 kPa and increases by say 0.5 kPa before change-out, thus the power would drop say 1%. The effect is nearly 2% power per kPa, and the efficiency drop is close to a half the power loss, or 1% per kPa. The actual correction depend on the engine design, but the common rule of thumb for inlet and outlet losses are as follows;

Inlet losses,	Power	-2 %/kPa	Efficiency	-1 %/kPa
Exhaust losses,	Power	-1 %/kPa	Efficiency	-0.5 %/kPa

This is percentage of performance, thus 1 kPa inlet loss means 30% efficiency goes to 29.7% not 29%.

Compressor and Turbine Fouling

The components of the gas turbine are susceptible to dirt deposits, erosion and corrosion. This deterioration is discussed more fully in section 5.2 and it's effect is to reduce the components efficiency. This was modelled on TURBOMATCH using a reduced isentropic efficiency and the results are shown on figures 3.36 and 3.37. This is not the best method of modelling the deterioration as in practice it occurs more strongly in the first stages and involves a loss of mass flow as well as efficiency. But this analysis does give an indication of the effects. From the figures it can be seen the effects of fouling on the compressor and the turbine are different. This difference allows differentiation between the forms of deterioration and forms the basis of some on-condition monitoring and maintenance techniques that will be discussed in section 5.7. The performance of a fouled compressor or turbine can be regained partly by cleaning but gradual permanent loss can be expected. Fouling and cleaning are discussed more fully in section 5.

3.4.6 Instabilities and Transients

Stall

Gas turbines inevitably have continuous flow nonpositive displacement compressors. Once the turbine is built the geometry, and thus the annulus area ratios, are fixed for the design point condition. At low speeds the first stage tends to stall, which is a major problem that gets worse with higher pressure ratios.

At low speeds the middle to rear compressor stages do not achieve their design compression ratios and the overall pressure ratio falls. This effect can be seen in figures 3.26 and 3.27. The lower pressure ratio means a lower density at exit and therefore the ratio of axial velocity to blade velocity increases from front to back. The result of this is to unload the rear stages until the last stage chokes. This puts back pressure on the first stages and pushes them into stall. Stall causes loss of performance and unwanted vibration.

Stall can be combated by three principal means; compressor bleeds, multiple compressors and variable geometry vanes.

Bleeding off some gas part way through the compressor and venting to atmosphere increases the mass flow in the first part of the compressor relieving stall in the first stages and choking in the last stages. This gives poor efficiency as compressed air is lost, and is thus used only during starting.

Multiple compressors are used in series with compressor and turbine pairs on different shafts. The high pressure compressor is driven by the high pressure turbine with the shafts usually concentric to each other. They are called multi-spool engines and have been developed only in aero-engines. The first or LP compressor will tend to run at a lower speed and the ratio of the HP to LP spool speed decreases as the machine load increases. Also the HP compressor takes an increasingly higher proportion of the total pressure ratio for reducing load which tends to ease the rear stage choking effect.

The use of variable geometry to reduce stall is achieved by adjustable stagger angle on the stator vanes. Multiple stage variable stator vanes were first used in aero-engines and include typically the first 4 to 6 stages. Many industrial engines of pressure ratio less than about 12 use a simple method of only the first stage inlet guide vane variable combined with a mid-compressor bleed at low speed. More modern industrial designs of pressure ratio about 15 and above are using multi-stage adjustment. The variable stators are partially closed at low speed which allows the first stages to operate at reduced flow without stalling, by changing the inlet angle. Also reducing the flow area restricts the flow and counteracts the unloading of the last stages at low speed. The stator vane is controlled from closed at start-up to design point position at full speed according to a speed related schedule. With heat recovery boilers, as in combined cycles, the inlet guide vanes are also used to reduce flow at part load and this raises the exhaust temperature to increase the overall efficiency.

Surge

At high pressure ratios the flow may reverse suddenly and cause large forces and possible catastrophic damage. This is predictable and is called surge. It is indicated on the compressor map, figure 3.21, as the surge line. Single shaft electrical generation engines have an advantage in that the external load is removed for acceleration during starting, moving the running line away from surge. Also the constant compressor speed at part load results in the working line on the compressor map running away from the surge line, figure 3.23. Twin shaft engines with a free turbine have a running line that follows somewhat more parallel to the surge line as shown in figure 3.23. These engines may surge at low load if no corrective actions were taken. Actions are the same as for stall and include variable vanes, multi-spoils or the use of bleed valves. Also fouling of the compressor may reduce the margin for surge and cause unexpected problems.

Transients

When a load change is required on a gas turbine, the fuel flow is varied and thus the TET. The fluid and thermodynamic changes are much faster than the mechanical changes and on load increase may result in surge or over heating. Generally increasing the TET increases the turbine power and the engine accelerates. The problem is that it also increases

pressure ratio and moves the compressor toward surge. The degree of overfueling possible will depend on the temperature capability of the turbine and the proximity of the surge line. On load shedding the opposite occurs and the compressor moves away from surge. The problem of proximity to surge is more severe on free spool gas generators. Here the compressor running line is closer and more parallel to the surge line. On twin spool engines the effects are similar for the HP spool, but opposite on the LP spool. During a deceleration the LP spool moves towards surge, but on both acceleration and deceleration the LP spool follows the running line more closely than the HP spool.

Aircraft engines are usually higher pressure ratio and are required to operate over a large range with rapid accelerations. This means aero-engines require more anti stall and surge features than industrialised engines.

The second major problem in load change is thermal transients on components. This can lead to overheating and thermal gradients, causing additional stress and low cycle fatigue. Starting is the worst thermal loading and the speed of loading can severely effect the engines life. All starts require some sort of over temperature to achieve full load and frequently started engines consume their life more quickly. An engine can usually be started faster than normal in an emergency, but at the expense of additional life usage.

On a deceleration the reduced fuel flow can cause a flame out due to a weakened mixture, and this may limit deceleration. Deceleration may also be limited in normal operation to minimise thermally induced strains and fatigue. But for an emergency unloading the fuel might be stopped to speed the process and then require a reignition. A sudden loss of load, such as with an electrical system fault isolating the generator or an overspeed trip, would tend to cause the turbine to rapidly accelerate. This is especially the case for a free turbine engine where the compressor load is not attached. The twin shaft control system should be able to react to cut off the fuel supply in less than about 0.1s to prevent overspeed [130].

The controller on an electrical generation gas turbine typically has a dead band of less than 0.1% of the rated speed, and this should be even less on larger engines. The speed governing and control system are usually considered stable if the sustained oscillation in speed is less than $\pm 0.12\%$ of rated speed, however again larger engines can be expected to have lower limits [130].

3.5 STARTING AND SHUTDOWN

Starting a gas turbine requires two main systems;

- * Provision to rotate the compressor and turbine up to speed
- * Provision for ignition of the fuel/air mixture.

The starter system is also usually used for motoring the rotor for compressor washing and maintenance. Separate from the starting system is sometimes a barring system to rotate the rotor shaft at low speed while it is cooling to prevent uneven cooling and bowing.

The starter motor has to supply sufficient torque to accelerate the

rotor smoothly to the 'self sustaining speed'. The main starting sequences are as follows:

- 1) Start of auxiliaries and possible diesel warm up.
- 2) Acceleration by starter for purging to give several complete air changes of the entire exhaust and stack volume to remove any leaked fuel vapours. Liquid fuel systems may require other special precautions to purge the system.
- 3) Acceleration or deceleration from purge speed to a speed optimum for ignition. This is sufficient flow for stable combustion and minimal hot spots, but low enough for minimal thermal fatigue damage.
- 3) Igniters are energised and then the fuel flow started.
- 4) Acceleration, possibly after a warm up period, is continued under the joint power of the starter and turbine to just above the self sustaining speed which is where the turbine is producing enough power for it's own acceleration.
- 5) The starter motor is dropped out.
- 6) Acceleration is continued under the engines own power to idle speed or to synchronisation with the grid. Periods of idle or scheduled acceleration rates may be necessary to allow the engine temperatures to even out and minimise thermal transient stresses.
- 7) The generator is synchronised in frequency and phase, then the circuit breaker closed.
- 8) The load is increased to the set point or for frequency control.

The start sequences are usually controlled by a computer. It may be initiated by one button push, and the computer carries out the actions and checks. If appropriate times and measurements are not adhered to the controller will abort the start and initiate shutdown.

Problems can occur and must be detected by the start controller. The most critical problem is probably failure to 'light-off', which is failure to get ignition. The fuel can build up in the turbine and exhaust duct with fire and explosion risks. After a failure to light-off the gas turbine may need to be motored without fuel for purging, to produce a specified number of air changes in the exhaust system. A failure to 'light round' is when a multi combustor engine fails to light all the combustors through the connecting 'crossfire' tubes. Combustion problems are detected by the multiple thermocouples that are mounted in the exhaust. If the starting period is excessive or fails to reach self sustaining speed it is called a 'hung start'. If increased fuel flow fails to accelerate the engine and it is overheated then it is called a hot start. These may be caused by insufficient starter torque, a starter dropping out too soon, failure to light round or poor fuel control. A hung start can be caused by stall induced by fouling or FOD. An engine with inadequate stall margin can cause a hung start when started cold and a hot start when started hot. Starters must be adequately sized for the coldest ambient start when the power required is the greatest.

Many industrial engines have heavy shafts and journal bearings. It may be possible to lift these shafts by a 'jacking' oil system prior to starting to reduce the stresses and the torque required. The starting requirement varies a great deal with the engine design and configuration. A single shaft heavy duty industrial engine may need 3% of it's rated output, for instance 3 MW for a 100 MW engine. The 200 MW GE frame 9F has a turbine, compressor and generator rotor weight of 150 tonnes and requires 90 kNm starter torque. A free turbine engine will

require much less as only the gas generator needs turning. Likewise a two spool engine requires even less power as only the high pressure spool needs turning.

Starting times are possible from 10 seconds in emergency generators of critical processes, to 20 minutes on large heavy industrial plant. The engine life is significantly effected by starting times and often both slow and fast starts are possible on the same engine. An example of a 20 MW aero-derivative is the aero-derivative LM2500 which takes 1.5 minutes to start to full speed at no load and 5.5 minutes to full power. This can be reduced to 2 minutes total for an emergency start. An example of a heavy engine is the 200 MW GE 9E which takes 8 minutes to full no load speed and 20 minutes to full power. This can be reduced to 10 minutes in an emergency. Starting times include a purge period, which varies from less than 1 minute up to 10 minutes with a large volume heat recovery system. In complex plant the gas turbine starting requirements might be relatively simple and the other plant then dictates the starting procedure and times.

Shutdown typically consists of the following sequences;

- 1) Controlled unloading to zero output at synchronised speed.
- 2) Opening of the circuit breaker.
- 3) Reduction to idle speed and a period of cooling if applicable.
- 4) Fuel cut off and coast down, including shut down of many auxiliaries.
- 5) Barring period if required for uniform cooling.
- 6) Remaining auxiliaries shut down.

A slow shut down may be used in normal operation to reduce thermal strains and fatigue. For example the 200 MW frame 9F normally has a 37 minute shutdown sequence, but in an emergency this is reduced to a few minutes. The cooling period may be from say 5 hours on small engines to 24 hours on larger units.

An emergency shutdown may be initiated automatically by protective devices. An overspeed sensor typically activates at 105% speed. Other devices detect flame failure, vibration and bearing temperature. On a sudden load loss the fuel valve should react sufficiently quickly to prevent overspeed, and the GT should be designed to be capable of subsequent normal operation without special inspection. Special cycles with stored heat or air, such as a regenerative cycle with a heat exchanger, may need vents or brakes to prevent excessive acceleration on a load trip.

3.6 RATING

The gas turbine can only cope with a certain maximum TET if it is to achieve a certain life. The effects of variable design point TET have been shown in figures 3.13 and 3.14. Figures 3.31 and 3.32 show how the weather can also effect the power or TET. The other major effect on life is the number of starts as starting puts the engine into over temperature and also large thermal loadings. The resulting maximum power is established for the turbine by the manufacturer. Standard classes are used for rating such as in ISO 3977:1978(E) and BS 3863:1979. The power should be quoted at ISO ambient conditions of 101.325 kPa, 15.0 °C and 60% humidity. Performance is assessed on the LCV of the fuel, for liquids using 42000 kJ/kg and for gaseous fuel, 100% methane and 50000

kJ/kg. The fuel LCV includes sensible heat of the fuel from 15 °C.

Rating come in standard classes and ranges as follows [130];

- Class A up to and including 500 hr/yr (reserve peak)
- Class B up to and including 2000 hr/yr (peak-load)
- Class C up to and including 6000 hr/yr (semi-base-load or mid-range)
- Class D up to and including 8760 hr/yr (base-load)
- Range I over 500 starts per year
- Range II up to 500
- Range III up to 100
- Range IV up to 25
- Range V continuous without planned shutdown for inspection or maintenance for a specified period

An example would be class B range II called ISO peak-load, giving 2000 hr/yr and 500 starts/yr on average.

The site rating is the generating plant net electrical power at the generator terminals with adjustments for auxiliaries, related to specified ambient and pressure losses. This is necessary as the average conditions would be rarely the ISO standard. Other ratings include type of starting, ranging from manual to automatic (one signal) and subsequent loading, manual or automatic.

The main confusion in rating is what is included in the output. Engines are quoted with or without inlet/exhaust losses, auxiliary power consumption, generator efficiency and using lower or higher heating value. They are also quoted on different fuels, with natural gas giving about 2% more power than a distillate due to higher specific heat in the combustion products resulting from the higher water vapour content. Also two shaft engines may quote the power as the isentropic gas power, which assumes a 100% polytropic efficiency on the power turbine. Some rules of thumb are included as follows;

* Conversion from Distillate to Natural Gas

- Power

Aero-derivatives power $\approx$ Constant

Heavy Industrial Natural Gas power $\approx$ Distillate power $\times$ 1.02

- Efficiency

Aero-derivatives

Natural Gas efficiency $\approx$ Distillate efficiency $\times$ 1.007

Heavy Industrial

Natural Gas efficiency $\approx$ Distillate efficiency $\times$ 1.01

* Actual shaft power $\approx$ Isentropic gas power $\times$ 0.87

* Efficiency LCV $\approx$ Efficiency HCV / 1.06

* Power with inlet and exhaust losses $\approx$ Power at 0 losses / 1.03

* Efficiency

with inlet and exhaust losses $\approx$ Efficiency at 0 losses / 1.02

* Efficiency = 3412 / Heat Rate in Btu/kWhr

3.7 SPECIFIC ENGINES. CASE STUDY

This section is included to give typical engine descriptions. The engines chosen include the older and successful aero-derivative the GE LM2500, and the new industrial engine the ABB Type 8.

3.7.1 General Electric LM2500

The LM2500 is an aero-derived industrial engine based on the CF6 family of aircraft engines (figure 3.38). It is derived from the military TF39 on the C5 transport and the commercial CF6-6 in the DC10-10. The engine was first developed in the 1960's, and is a second generation aero-engine using semimodular design and on-condition monitoring. Technology advancement since the first version resulted in the LM2500PE uprated to about 22 MW. As at January 1988 there were over 1000 aero engines in active service, and over 500 industrial and marine versions. The gas turbine is usually supplied as part of an overall system driving a unit such as a power generator or through reduction gears to a pump. The specific rating depends on the application and model. LM2500 electrical models are usually rated at 20-21 MW and are quoted at 33-36% thermal efficiency. The rated power is guaranteed and includes a 3% margin for unit to unit variation, thus actual power could be higher. Rapid starting can be achieved in 2.0 minutes to full base-load, but normal start time is 5.5 minutes. Of this time only 1.5 minutes is to start the engine to no load full speed, while the remainder is loading time.

The engine is single spooled with a free turbine. It consists of a gas generator, power turbine, a flexible coupling shaft, and inlet and exhaust components. Typically the weight of the gas turbine is about 3355 kg (from inlet to exhaust components) and dimensions of 6.7 m long and 2.2 m high. Including the base and enclosure the size is 16700 kg and 8.6 m long by 2.8 m wide. The gas generator speed is 5000 rpm at idle and 9500 rpm at maximum power. The free turbine can be run at 3000 or 3600 rpm thus it does not need to be geared for power generation.

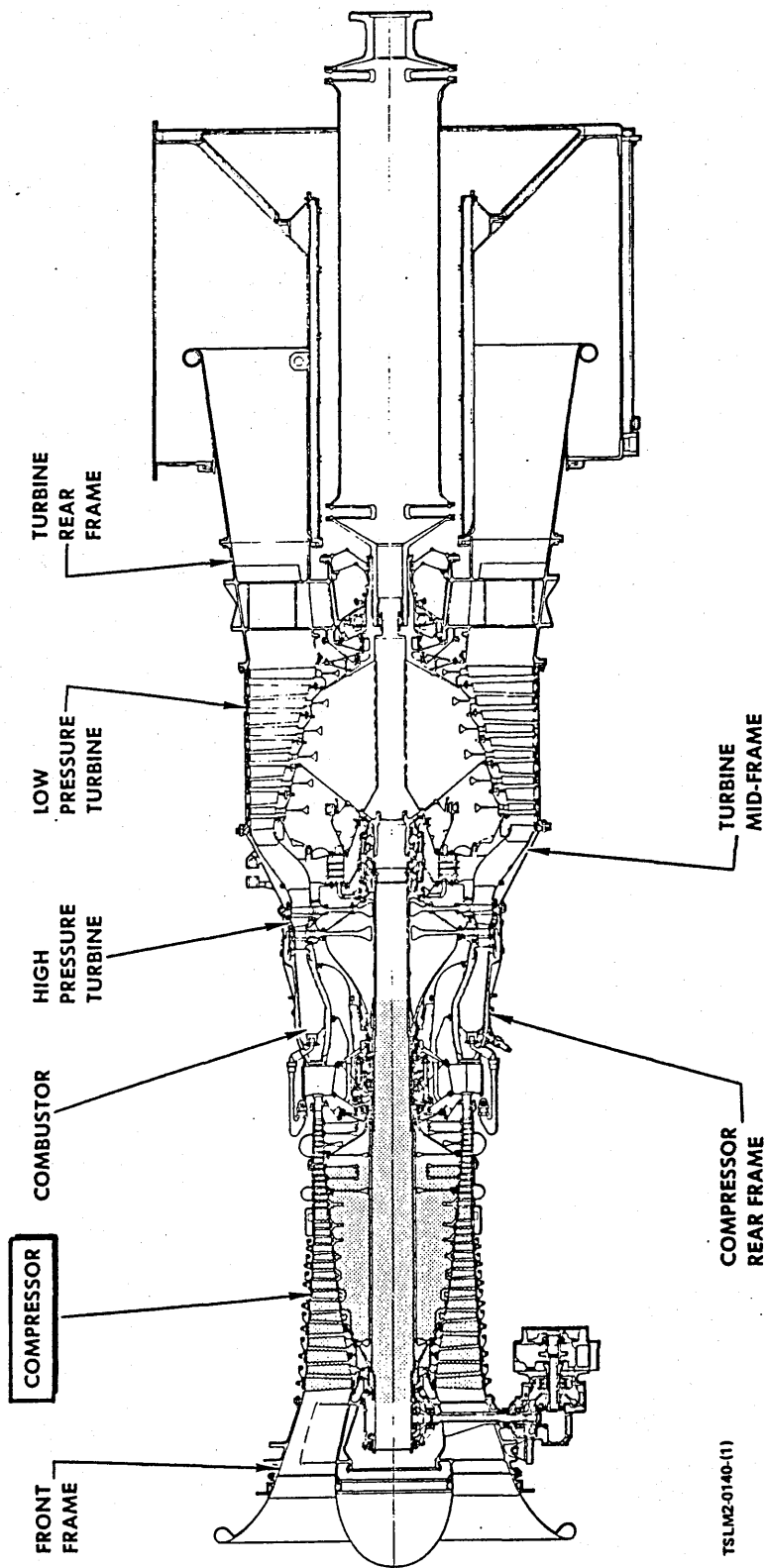
There are six ball and roller bearings, two for each of the compressor, high pressure turbine and power turbine. The compressor has 16 stages giving a nominal pressure ratio of 18. To prevent surge and improve part load efficiency the compressor has variable angle inlet guide vanes and the first 6 stators variable. The combustor is annular with the TRIT ranging up to 1485 K. As an aero-derivative it is less suitable to heavy fuels. There are two turbine stages on the gas generator and 6 stages on the free turbine. About 20% of the compressor flow is used for cooling. The exhaust flow is 67.0 kg/s and about 800 K thus suitable for combined cycles. To have an ungeared steam turbine at only 3000 or 3600 rpm multiple engines would be needed for each steam turbine.

The gas turbine is typically water washed at a power loss of 2-5% with off-line washes occurring at somewhere in the order of 2-5 weeks. Water washing is typically a caustic cleaner and consists of a solvent added to water. The water is added while cold and cranking at a speed of about 2100 rpm and is followed by one or two plain water rinses. An average nonrecoverable performance loss of 4-5% is expected in about 3 years. The hot section repair interval is normally 25000 hours or 3 years at ISO conditions, assuming base-load of 8600 hr/yr and natural gas.

For constant power the engine uses a controller to adjust the TET as ambient temperature and pressure vary. In this case the engine is de-rated to give an equivalent life. The inlet guide vanes are controlled on ambient temperature and compressor speed. Compressor bleeds for cooling are at the 8th, 9th, 13th and 16th stages, with a customer bleed available from the 16th (compressor discharge pressure).

Figure 3.38

LM2500 Side View



TSLM2-0140-11

Lubrication pumps and essential auxiliaries are driven by a radial shaft coupled directly to the main shaft. The casing includes 40 borescope inspection access ports, and the engine consists of modules.

A module is essentially a subassembly of certain major engine maintenance units. An engine maintenance unit is a component assembly that is interchangeable. The GT consists of the following modules:

- Compressor front frame
- Gearbox module
- Core engine
- Power turbine

There are 11 engine maintenance units ranging up to 227 kg. They are designed to be replaceable individually on-site using on-condition maintenance. It is claimed that the gas turbine can be removed and replaced in 9 hours and the power turbine in 12 hours with a crew of 5. Hot section removal and replacement is claimed as 19 hours.

The LM2500 is delivered prepackaged in typically 15 assemblies ranging from the turbine, accessory and control compartments to water wash and lubrication oil skids. Water or steam can be used to reduce NO_x emissions, although this reduces the performance and increases CO and unburnt hydrocarbon emissions. The LM2500PE on natural gas has NO_x emissions of 175 ppm, which corresponds to 66.7 kg/hr of NO_2 and 1.4 kg/hr CO. With water injection the NO_x emissions reduce to 42 ppm and with steam to 25 ppm. CO and UHC emissions are not quoted for water injection. These are guaranteed values and are increased for distillate fuel operation. The water to fuel ratio is 0.6:1.0 or the steam to fuel ratio is 1.5:1.0. Water for injection is required to be distilled/deionized. Gaseous fuel delivery pressure is nominally 22 bar. Oil consumption is about 95 cm^3/hr average.

The LM2500PH is a modified engine for when steam injection is used for power augmentation, with a slightly increased turbine inlet area. The maximum injection rate is 5 kg/s or 7.5% of inlet air flow. Steam injection is discussed in section 7.12.

References for the LM2500 include [140], [141], [142] and [143].

3.7.2 ASEA Brown Boveri GT8

The type 8 is a recent small industrial engine (figure 3.40), with the first order placed in 1984. Only 13 had been ordered by late 88. The engine is a single shaft and single spool gas turbine generating 45 MW. The efficiency is quoted at 31.6% for base-load or 32.0% at peak-load. As the speed is 6300 rpm, the gas turbine is geared to the generator. In this size range the speed needs to be higher than 3000/3600 rpm to achieve reasonable efficiency and size. It passes 180 kg/s of exhaust at 523°C. This is 800 K and suitable for a combined cycle. The TET for this exhaust temperature and pressure ratio must be about 1470 to 1480 K.

The turbine runs on two journal and one thrust bearing, and uses a solid shaft. The shaft comprises individual discs welded together. The compressor has 12 stages and uses variable inlet guide vanes, giving a pressure ratio of 16.3. The turbine has 3 stages. In ABB tradition the combustor is one very large vertical chamber and has long resident times

for use with poor fuels including crudes and residuals. For compressor or turbine access the cast case is horizontally split allowing access without removing the shaft. All maintenance is intended to be done on site

As a heavy industrial gas turbine the engine is very much heavier than an equivalent aero-derivative. The modules sizes are as follows;

gas turbine	83.2 tonnes	8.4 x 3.2 x 3.8 m
combustor	17.4 tonnes	6.2 x 2.5 x 3.0 m
gear box	34.0 tonnes	7.0 x 3.1 x 3.1 m
generator	97.0 tonnes	8.7 x 3.2 x 3.5 m
control module	28.0 tonnes	5.9 x 2.8 x 3 m

The plants total mass is over 260 tonnes, and gives a minimum footprint of approximately 38 x 12 m with a typical stack height of 17 m.

Gaseous fuel delivery pressure is normally about 21 bar, and pressurised fluidised coal combustion is offered as an alternative, although not as yet built. The standard generator package has a clutch for operation as a synchronous condenser.

References for the type 8 include [251], [144] and [145].

Figure 3.40 Section Through the Type 8 Gas Turbine

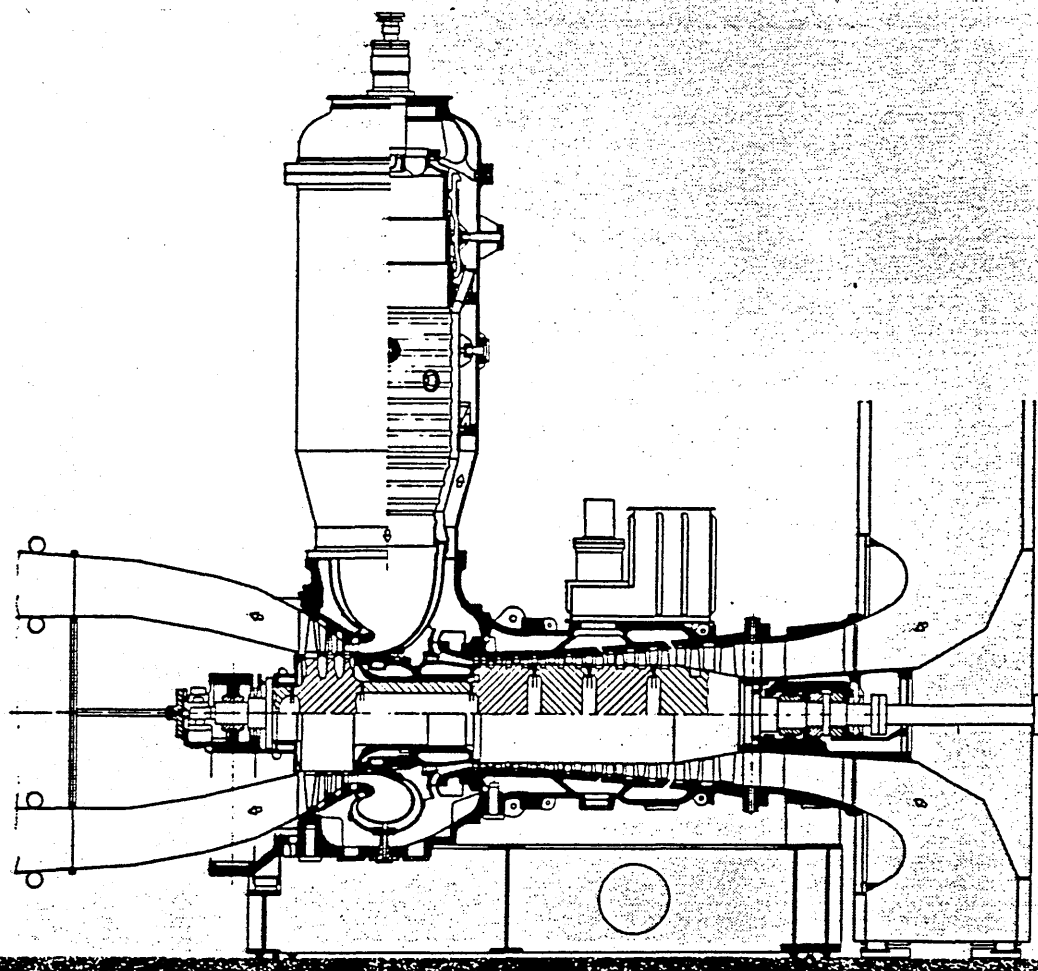


Figure 3.41 Plant Layout for Twin Type 8 Installation

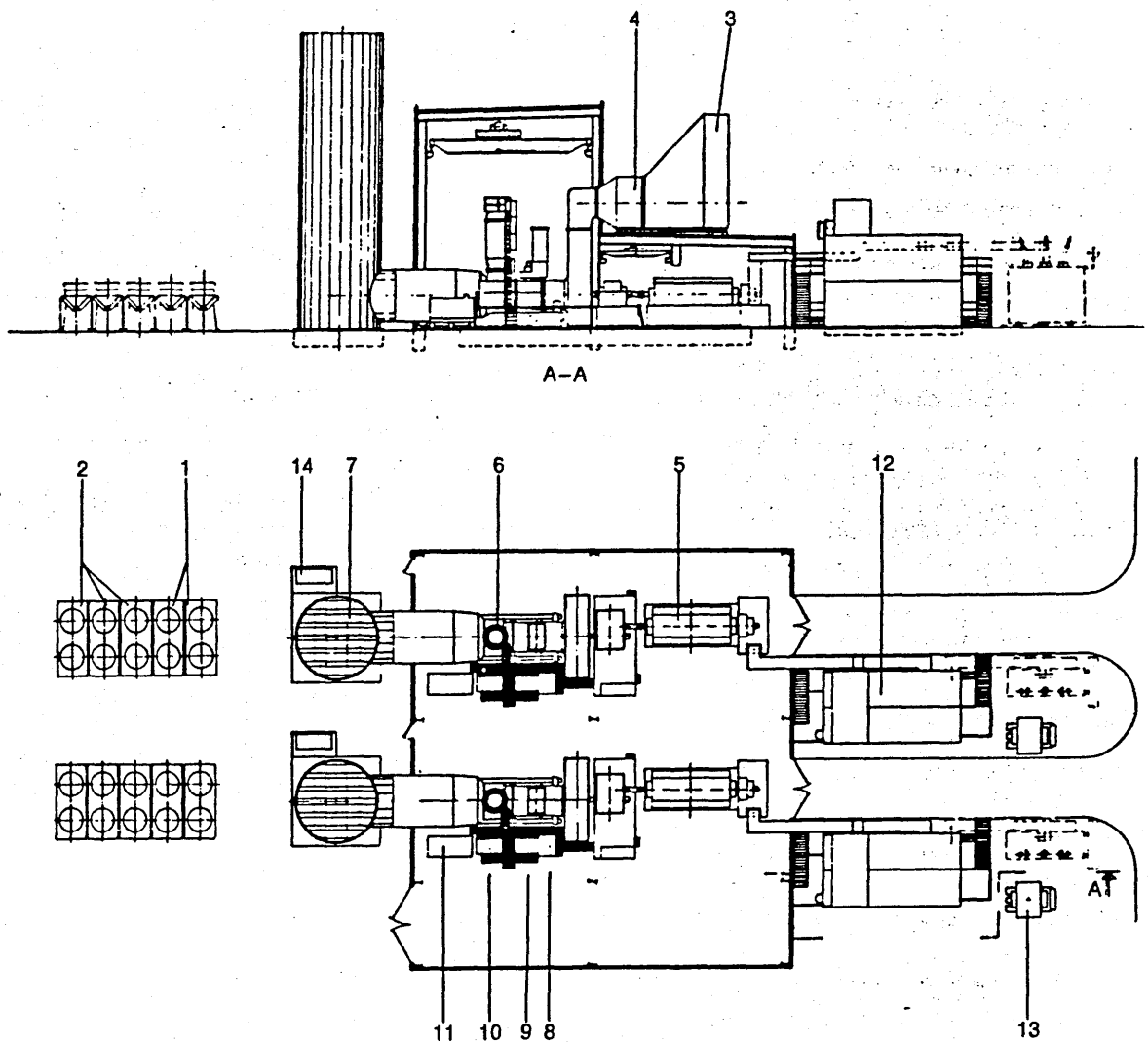


Fig. — Plant layout for a twin indoor installation

- 1 = Generator cooler
- 2 = Oil cooler
- 3 = Air-inlet system
- 4 = Sound absorber
- 5 = Generator
- 6 = Combustion chamber

- 7 = Exhaust-gas stack
- 8 = Auxiliary module
- 9 = Control module
- 10 = Gas control module
- 11 = Fuel pump module
- 12 = Control and switchgear modules
- 13 = Auxiliary and start-up transformer
- 14 = Air cooler

4 GAS TURBINE FUELS

4.1 INTRODUCTION

Closed cycle gas turbines can conceivably use any type of heat source, but in an open cycle mainly gaseous or liquid fuels have been used. Much work has been completed on coal burning with commercial plant now available and being built. The two main fuels, gas and liquid, have widely differing properties and can effect the turbine operations. Generally natural gas and the lighter fuels have higher efficiency and lower maintenance cost but these fuels may be more expensive.

By far the most common fuel in America, Europe, Japan and Australasia for stationary GT's is natural gas, because it is clean burning, available and currently inexpensive. In the Middle East there are large numbers of turbines using crude oil and the some burning of crudes or residuals by the rest of the world. Most of the remaining engines are using distillates with some process gasses. Natural gas supplies are limited and it's price is expected to increase in the future, thus interest is increasing in the use of the cheap residuals and coal.

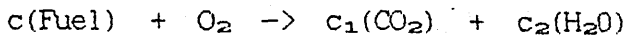
Aviation turbine fuels are almost exclusively kerosine of various specifications. Traditionally stationary aero-derivative engines have lacked the ability to burn heavy fuels but some are now able to burn these although with tendency for smoke formation. Experiments have been conducted on using liquified hydrogen or liquified methane in aircraft applications but are not of particular interest to industrial gas turbines.

The combustion process involves atomisation and evaporation of the fuel, if it is liquid, and near stoichiometric combustion at high temperatures. This occurs in the 'primary' combustion zone. The combustion is then cooled in an intermediate zone by some air dilution to complete combustion and reduce dissociated products. The combustion products are then diluted again with the remaining air to the turbine entry temperature in the dilution zone. This is shown in figure 4.1.

4.2 FUELS BACKGROUND

Almost without exception fuels that are burnt are hydrocarbons. The compounds of hydrogen and carbon number approximately two million, but each can be classified into any of many groupings. Carbon combines with itself in single bonds (saturated), paraffins or naphthalenes, and multiple bonds (unsaturated), olefins, acetylenes and aromatics. Also any particular number of atoms in a molecule can form various arrangements called isomers. The number of possible isomers increases very rapidly with number of carbon atoms ranging from 1 for a three carbon atom molecule to 10,000 for 16 carbon atoms per molecules. These variations lead to a large number of properties. Generally fuels are mixtures of many compounds and have properties due to the combined effect of the constituents. Higher numbers of carbon atoms per molecule tends to increase the fuel density and lower it's boiling point. Density is thus used extensively to measure the fuel properties against.

The combustion of hydrocarbons can be generalised by the following equation;



It can be shown that given otherwise identical specifications there is a difference in theoretical efficiency with different fuels and that thermal efficiency of the engine increases as the parameter c_1+c_2 is increased [62]. Coal (C) has a parameter of 1 whereas hydrogen (H_2) and methyl alcohol (CH_3OH) have a value of 2, with the parameter increasing with hydrogen content. The decrease in parameter from 2 to 1 decreases the theoretical thermal efficiency by about 2% even before any de-rating of the turbine for other factors. This is mainly due to the combustion products having lower specific heat.

Other than the fundamental efficiency changes, the use of heavy fuels may require the turbine to be de-rated due to hot combustion components and fuel contaminants. Generally higher carbon content increases radiation effects and leads to a hotter combustion chamber. A low hydrogen fuel has a lower calorific value and also tends to carbon deposition and smoke. Contaminants are to do with the source and processing of the fuel but modern refinery practices generally means heavier fuels are more contaminated. High carbon content is thus generally undesirable.

Natural gas is removed from the crude oil at the well head, and usually contains largely methane (CH_4). The crude can be burnt directly after cleaning but is usually refined. Refining consists of distillation (physical separation), cracking (chemical conversion) and treatment. The process makes use of the decreasing boiling points of hydrocarbons with carbon content. The crude is heated in a controlled temperature gradient in columns, and various groups (fractions) of compounds are separated. Steam is used to assist separation by depressing boiling points and partial pressures of volatile components. This distillation process is where the terms 'distillates' and 'residuals' come from. The straight distillation normally produces proportions of products that do not match demand and further cracking is used to modify the compounds. Cracking is done under heat and pressure to dissociate the carbon bonds in large molecules using catalysts. Reforming is partial cracking to give controlled conversion and combination.

Various tests are available to specify the required fuel properties. These are related to the handling, combustion, and products of combustion of the fuel. Tests and specifications for liquid fuels include;

Density;	Usually at 15°C, directly effects spray droplet size
Calorific Value;	Determines the energy density
Distillation;	Gives a broad indication of fuel blend and effects volatility for combustion and starting
Vapour Pressure;	Only necessary on very light fuels, effects storage losses and safety
Flash point;	Temperature of weak combustion limit above fuel
Spontaneous Ignition;	Temperature at which fuel burns without an ignition source (Cetane Number)
Viscosity;	Dynamic indicates settling rates of contaminants, and kinematic the pumping power needed
Freezing point;	Point at which wax crystals just begin to form or disappear

Figure 4.1 Conventional Combustor

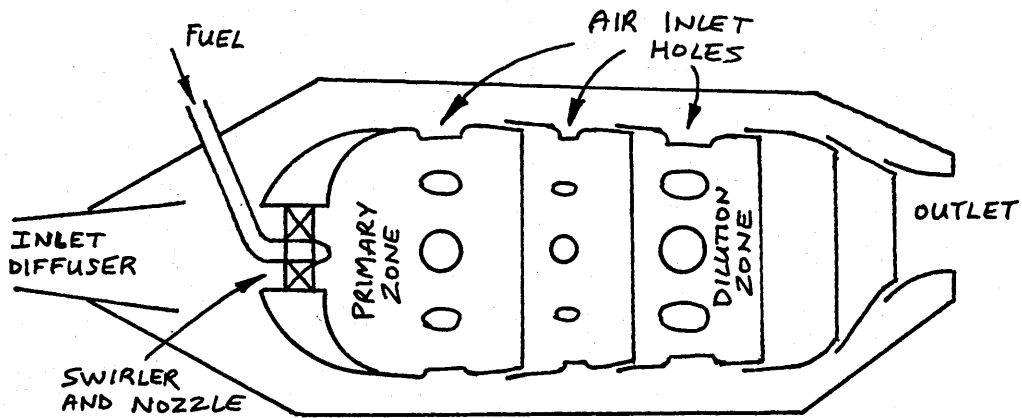


Figure 4.2 Combustion Temperature vs Emissions

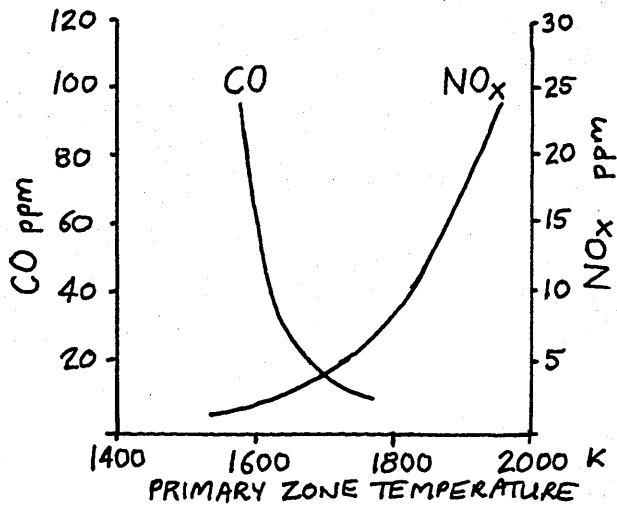


Figure 4.3 Power vs Emissions

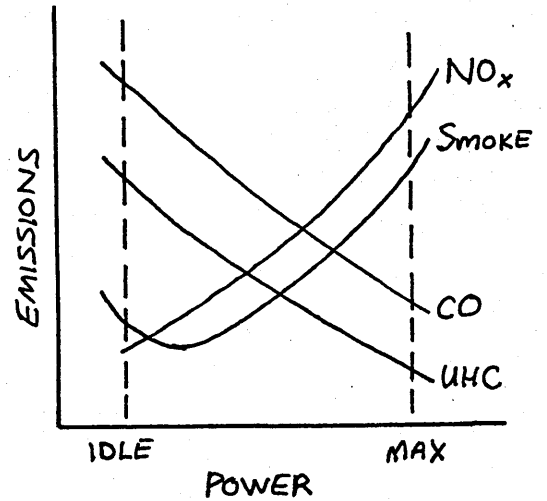
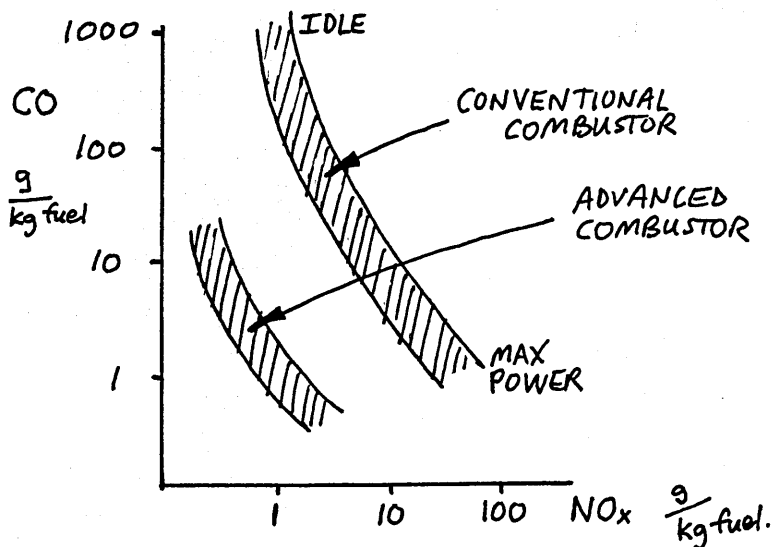


Figure 4.4 Combustor Characteristics



Other tests include Aniline Point, Bromine number, Olefins & Aromatics, Asphaltenes, Char value, Carbon residue, Smoke point, Existent gum, Sediment, Ash content, Water content, Sulphur content, Mercaptan content, TEL content and others.

Standard specifications are used to classify fuels, such as kerosine, fuel oil or heavy oil. Specifications vary with the application and include various aviation kerosines and vehicle petrols and diesels. For industrial gas turbines standards include ASTM (D-2880-78), DIN, SIS and others giving fuels like 0-GT, 1-GT, up to 4-GT which are described as follows;

- 1-GT Pure and light distillate with low carbon formation
- 2-GT Medium distillate which may require some fuel heating
- 3-GT Heavy distillate, blend of residual and distillate or residual meeting low ash requirements. Requires preheating.

The relative cost of the fuel depends on the tightness of the specifications, the quality of the crude oil and the demand for each fraction of oil. If demand for a particular fraction is too high other fractions will have to be altered to increase the yield. A current trend is to use improved technology to extract more of the lighter oils which lowers the quality of the residuals.

4.3 GASEOUS FUELS

The most common gaseous fuel is natural gas, but certain blast furnace gases, gas products from oil refineries, mine gases and rubbish land-fill gases have been used. Synthetic gasses can be derived from coal.

Natural gas is considered the best fuel because it maximises efficiency, minimises maintenance and is currently cheap and requires little handling or customer treatment. It has essentially no ash and consequently leads to little deposition or high-temperature corrosion. This means the blade life of the turbine can be near it's oxidation life of up to 100,000 hours. The high hydrogen content means higher theoretical efficiency and burning characteristics mean maximum rating of the turbine.

Natural gas consists of mainly methane but usually has less significant components of heavier gases in rapidly smaller proportions such as ethane, propane, butane, etc. Usually a significant content of carbon dioxide and nitrogen are present. The components have a characteristic dew point and heavy fractions such as the C_5 's and C_6 's may condense out at low temperatures. Low temperatures are often present due to pressure reduction at the gas receiving station, thus the fuel system must usually be designed to remove these liquid components to prevent the potential damage due to localised heating.

Sour gases are gases containing sulphur such as sulphur dioxide (SO_2) and hydrogen sulphide (H_2S). If these components exceed approximately two percent then they need scrubbing or other treatment. Sour gases cause a mild form of sulfidation and can reduce blade life although modern alloys have reduced the problem. Sodium can aggravate sulfidation problems excessively, such as air borne salt, and therefore filtering is important.

Process gasses include blast furnace, refinery, coke oven and other by-product gases and are generally low calorific value. This can be a major drawback if the gas has to be compressed before use or can be an advantage if it arrives at pressure. For example a prepressurized gas containing 60% CO₂ could increase the power of the turbine by 8% by increased turbine mass flow. The low energy density also calls for larger pipes and fittings, and special combustor design. Blast furnace gas generally arrives at low pressure, therefore requiring compression work, has erosive and corrosive contaminants needing removal, and may require a pilot flame from a separate fuel. Refinery gas may be largely H₂ which requires special handling to prevent leakage and explosion. These gasses have been successfully used in several locations.

Synthetic or manufactured gasses cover a wide range of components, usually carbon monoxide, hydrogen or methane. These are generally made from coal gasification and are usually of either low or medium heating value. This is discussed in section 4.7.

Table 4.1 Typical Properties of Gaseous Fuels

	Natural Gas	Coal Gas Low CV	Coal Gas Medium CV	Blast Furnace
CV MJ/m ³	35-43	4.1-6.1	19-26	3.4-3.7
Spec. Gravity	0.58-.72	0.8-.92	0.41-.48	0.95-1.05
Composition %				
CH ₄	75-99	0.5-4.5	20-35	0
C ₂ 's+	2-20	0	2-4	0
H ₂	0	12-16	40-55	1-4
CO	0	2-32	5-15	25-30
N ₂	1-16	30-55	4-11	55-60
CO ₂	0-10	0.5-10	2-4	8-16

4.4 LIQUID FUELS

The chemical compositions of liquid hydrocarbons are often very complex, involving many compounds and isomers. These compounds usually also contain other elements like sulphur (S), nitrogen (N) and vanadium (V). The fuel properties depend on their composition which in turn depends on the source of the crude oil and the refining processes. As mentioned earlier the density varies with molecule size and carbon content, and is used as a base to measure other properties. Generally as the density increases the strong patterns of table 4.2 emerge.

Table 4.2 Trends in Hydrocarbon Fuels with Density

Increasing	Decreasing
Density	Spontaneous ignition temperature
Freezing point	Vapour pressure
Flash point	Energy per unit mass
Viscosity	Volatility
Distillation temperature	
Energy per unit volume	

Properties of liquid fuels can be divided into two main categories, physical and chemical. Physical properties dictate fuel handling and include viscosity, pour point, wax separation, flash point and others. Chemical properties include composition of the fuel and effect combustion characteristics, hence the combustor, and the products of combustion, corrosion, deposits and pollutants. Chemical properties can also effect handling via fuel stability such as gum forming or cracking. High temperature corrosion and deposits are a problem when contaminants form ash with a lower melting point than the gas temperature.

Naphtha and kerosene make excellent gas turbine fuels. Both of these fuels have low viscosities and require special lubricating measures to prevent pump seizures. They also dissolve fats and degrease their fuel systems which can cause rust problems in storage and rust particles carried into the fuel system. Naphtha also needs special handling due to it's flammability. These problems are readily solved. The fuels are primarily used in aero-derivative gas turbines.

Number 2 fuel oil is ideal for industrial gas turbines. It is used in modified aero-derivatives but their small combustors can create some problems. Precipitation of waxes can cause blockages, which is solved by using heating in colder climates. Ash products are usually not a problem, although where the fuel quality cannot be guaranteed it is necessary to consider fuel additives.

Heavy distillates are denser than number 2 fuel oil and will solidify at about 25 to 50°C. This means that heavy distillates require heated storage and handling, and the fuel will also need further preheating before atomisation. These fuels are suitable for heavy duty industrial gas turbines, and can be supplied with a low sulphur content at a price.

Crudes and residuals require heating for storage and handling and usually contain damaging ash forming contaminants. The contaminants and protective measures are discussed in the following section.

4.5 CONTAMINANTS

Contaminants can be divided into three major types, particulate, water soluble and oil soluble. Effects of these contaminants are mainly due to the ash produced or the erosion effects. The main elements of concern are vanadium, lead, nickel, iron, sulphur, sodium, potassium, and calcium.

Water soluble contaminants are usually salts of sodium, potassium and calcium. They are dissolved in emulsified water and often enter the fuel in transport, for instance as sea water contamination. This means even clean distillates may be contaminated by the time they are delivered. These contaminants are removed by water washing. Clean water is added to the fuel and mixed to form an emulsion. The water then dilutes the salts and is removed by either centrifuge or electrostatic precipitation. Both systems have their advantages and disadvantages. Centrifuging is generally favoured on small installations and increases the effect of the density difference. Electrostatic methods are favoured in large plants and use the charge to coalesce the droplets into bigger sizes to speed gravitational separation. Generally distillate emulsions break easier than heavies, which may require chemical aids. When the specific gravity of the heavy fuel is near 1.0 then washing may be

impossible. Fuel washing is an established technology and removes about 90% of the salts in each pass down to about 1 ppm. Often two or more passes are needed and then the contaminated effluent water must be treated before disposal.

Particulates such as sand, rust or other debris are initially filtered out. Secondary removal is by the same means as water removal although centrifuges are better than electrostatic precipitators. In some installations dry centrifuges are used purely for removing particles. Particles usually do not effect corrosion or deposits but do cause erosion.

Oil soluble contaminants are generally vanadium, lead, nickel, iron and sulphur. Most crude oils contain vanadium, iron and nickel, and due to their high weight and low vapour pressure these are left in higher concentrations in the residual fuels. Lead is not generally naturally occurring in crude but finds it's way into fuel for instance when the sludge from the gasoline storage tanks is disposed of by adding it to the residuals. Nickel is usually present in only small fractions. Sulphur is always present in all grades of fuel and it's concentration is a significant cost factor. The most important contaminant for corrosion is vanadium which is almost always present in crude oil in concentrations from 1 to several 100 ppm (gram/tonne).

High temperature corrosion and deposits are the effects of contaminant elements. These can reduce the material life and performance dramatically in unacceptably short periods. The main problem is vanadium, which with sodium forms low melting point corrosive ashes. Examples of the reactions are as follows;

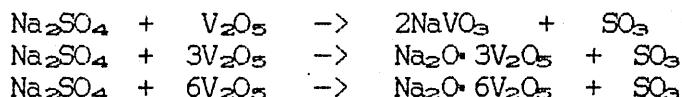


Table 4.3 Melting Points of Gas Turbine Ash

Compound	Melting point °C	Compound	Melting point °C
Na_2SO_4	883	K_2SO_4	1076
$\text{Al}_2(\text{SO}_4)_3$	770	CaSO_4	1450
MgSO_4	1123	NiSO_4	840
$\text{Na}_2\text{S}_2\text{O}_7$	400	ZnSO_4	740
V_2O_4	1970	V_2O_5	675
FeCl_3	282	NaCl	800
KCl	776	CaCl_2	765
$\text{Na}_2\text{O} \cdot \text{V}_2\text{O}_5$	630	$\text{Na}_2\text{O} \cdot 3\text{V}_2\text{O}_5$	668
$\text{Na}_2\text{O} \cdot 6\text{V}_2\text{O}_5$	702	$2\text{Na}_2\text{O} \cdot \text{V}_2\text{O}_5$	640
$2\text{Na}_2\text{O} \cdot 3\text{V}_2\text{O}_5$	565	$3\text{Na}_2\text{O} \cdot \text{V}_2\text{O}_5$	850
$\text{Na}_2\text{O} \cdot \text{V}_2\text{O}_4 \cdot 5\text{V}_2\text{O}_5$	625	$5\text{Na}_2\text{O} \cdot \text{V}_2\text{O}_4 \cdot 11\text{V}_2\text{O}_5$	535

The melting characteristic of ash has four distinct stages;

- 1) Sintering - Shrinkage and mild adhesion, low deposit forming
- 2) Initial melting - Greater adhesion and deposit forming begins
- 3) Melting point
- 4) Flux point

Problems with deposits occur when the metal surface or gas temperature is sufficient to cause adhesion. Melting points are given in table 4.3. Approaches used are to remove the contaminant before combustion, use an additive that reacts to produce a less aggressive compound or to use additives to dilute the ash and reduce its effect. Corrosion and deposits are reduced if additives produce high melting, dry combustion products that are thus relatively nonadhesive. Unfortunately additives for corrosion suppression can increase deposits, and require more additives to combat these effects. Initial use of heavy fuels encountered few problems due to the lower temperatures used, and deposits were the main concern. When temperatures increased above about 730°C corrosion became a severe problem and deposits became secondary, and many additive development programs were started.

Additives;

The additives for reducing vanadium ash problems are silicon, calcium and magnesium. Additives that have been used are Kaolin (Aluminium silicate, $\text{Al}_2\text{O}_3 \cdot 2\text{SiO}_2 \cdot 2\text{H}_2\text{O}$), Talc, magnesium silicate and organo-silicate. Talc and magnesium silicate proved to be less effective in preventing fouling than kaolin due to the formation of sulphates, but kaolin caused some erosion. Silicon prevents sulphate formation and renders vanadium compounds harmless by absorption, but is less effective than magnesium at higher temperatures over approximately 700°C. Calcium showed promise in inhibiting corrosion but the deposits formed are hard and difficult to remove, thus magnesium is better as a whole. One problem with magnesium is that vanadium prefers to react with sodium so to stop sodium ruining the effect the fuel must be water washed in conjunction with the additive. Magnesium solved the corrosion problem but did not resolve the deposit problem. To prevent build-ups of deposits the fuel must not be over treated and the turbine will require regular cleaning.

High temperature turbines over approximately 930°C tend to get MgO rather than MgSO_4 , and the deposit tends to be MgSO_4 on the cooled metal surface with an over-layer of MgO, which is harder to remove and potentially can plug cooling holes. Addition of silicon oxide as a further antifouling agent has reduced deposits by up to 90%, thus this combination can give satisfactory service.

Current Practice;

Typical current practice is to water wash to remove the sodium content to less than 1 ppm. Then the fuel is dosed with an anti-corrosive additive of magnesium sulphate at a concentration of about three times the vanadium content. If the dosage is low then a more expensive oil soluble magnesium additive might be used. A further additive of silicon dioxide is used to reduce the fouling. Periodic on-line solids cleaning can extend the time between water washes to say 1000 hours, although solids cleaning is not always possible.

Sulphur;

Sulphur content causes formation of SO_x , which apart from environmental problems with acid rain, can form sulphuric acid in the exhaust if cooled sufficiently. This is a problem for heat recovery boilers which have much cooler exhaust temperatures. Sulphur tends to increase with fuel density, with less than 0.5% in distillates and up to 4% in some residuals. The major product is SO_2 which is a pollutant but causes few

problems to equipment. SO_3 is the problem product and corrosion increases rapidly when catalysed by iron oxide. A coating of MgO negates this effect and addition of Mg not only reduces the catalytic formation of SO_3 but tends to neutralize any H_2SO_4 formed. Usually Mg added to treat vanadium is sufficient for SO_3 protection as well.

4.6 EMISSIONS

The main emissions of a gas turbine are smoke, CO_x , NO_x , SO_x , and unburnt hydrocarbons (UHC). As evidence mounts up about the greenhouse effect there is increasing concern about CO_2 emissions. However as CO_2 is an unavoidable product of liberating heat it will suffice to say less CO_2 is produced by fuels of higher hydrogen content per unit of energy output. Natural gas produces about 50%, and oils about 75%, of the CO_2 produced in burning coal.

NO_x , UHC, SO_x and CO are harmful to the environment. CO and UHC represents partially burnt fuel and lower combustion efficiency. The production of CO and UHC is reduced by increasing the local temperatures of combustion. NO_x is the reaction of air or fuel bound nitrogen. Air borne nitrogen reaction to NO decreases with decreased local combustion temperatures and decreased combustion residence times. In liquids the fuel bound nitrogen generally reacts fully to produce NO , thus the higher nitrogen content of heavy fuels like residuals and crude oils add to NO production. NO_2 is produced when NO is cooled in the combustion products. All sulphur in the fuel is burnt to form SO_x . The only effective control for sulphur is to treat the fuel at source. This means formation of SO_x is largely to do with fuel type and specifications.

Smoke is soot formed from fuel rich pockets of combustion in the combustor. It is due to air and fuel mixing processes and is not chemically driven. Smoking tendency increases with reduced hydrogen content, as fuel atomisation produces larger less volatile droplets. Smoke is always produced in the primary zone of combustion but is largely burnt again in the later zones. This means short aero-derivative combustors have more tendency to produce smoke than large long residence time industrial combustors and aero-derivatives may not be able to burn heavy fuels. Smoke can be reduced with refined combustor design, dual orifice fuel nozzles or additives. It should be noted that metallic additives used to reduce smoke may be more toxic than the smoke itself and tend to form deposits on the turbine.

The requirements for reducing NO_x are opposite to requirements of reduction of CO and UHC. Low CO and UHC requires high temperatures while low NO_x requires low temperatures, thus there must be a compromise. Figure 4.2 shows this trade off. The emissions from the gas turbine at various loads are shown in figure 4.3. At low loads the lower temperatures increase CO and UHC, while at full load the higher temperatures increase NO_x . Emission control techniques adjust the effective temperature level and often just shift the compromise to a different point. For any particular combustor the performance is shown more effectively in figure 4.4, which discriminates between mere trade off between CO and NO_x and true advancement in technology, which would shift the line. Real improvements include lean or rich combustion techniques and improved precombustion mixing. These techniques change the combustion temperature and reduce local hot or cold spots. The future combustor may include features such as variable geometry, twin

wall cooling, premix/prevapourisation fuel injection and/or two stage combustion.

For a given combustor improvement of the NO_x emissions at the expense of CO and UHC emissions can be achieved by water or steam injection, or exhaust re-injection. These processes dilute the combustion with an inert compound and reduce temperatures. Water injection is more effective if well atomised and injected into the primary combustion zone of the combustor rather than into the full air flow, and the effect of water injection is less on heavy fuels with fuel bound nitrogen. The process requires a continuous supply of demineralised water in ratio to fuel of 0.5 to 2.0, with appropriate water handling and storage facilities. Steam injection requires even higher flow rates and is only practical where steam is otherwise available. Water and steam injection do have other side effects as follows;

- The increased turbine mass flow increases the compressor pressure ratio and reduces surge margin
- Increased water content of combustion products raises heat transfer coefficients and raises metal temperatures.

4.7 COAL AS A FUEL

Coal can be utilised in a gas turbine in several ways, with combustion integrated in the cycle, or used to produce a different fuel such as a gas. Coal gasification development was stimulated by the oil shock of the mid 1970's, but currently it is a proven technology without commercial application due to the low prices and availability of alternate fuels such as natural gas.

Direct combustion of pulverised coal is possible in a gas turbine combustor. Coal can be pulverized and cleaned to produce an ash content as low as 0.5%. The cleaned coal can then be burnt in a gas turbine slagging combustor. However the ash content is still too high and erosion, corrosion and pollution problems are unsatisfactory. The complex cleaning process used to make coal water slurry also makes it very expensive and currently uneconomical. Direct combustion of pulverized coal is still at an early stage of development.

Coal can be gasified to produce gases of low, medium or high calorific value. If an air and steam process is used the gas is of a low heating value due to the large amount of nitrogen. When oxygen and steam are used the gas produced is of medium heating value. Low heating value gas can be burnt in gas turbines but there are several side effects as follows;

- increased turbine mass flow increases the compressor pressure ratio and reduces surge margin,
- increased fuel volume flow, made worse as coal gasses are usually delivered hot, requires larger pipes and valves,
- wet gases increase the combustion product heat transfer coefficients and raises metal temperatures,
- low CV gases require more air for combustion and high TET engines may not be able to burn them.

A number of gasification plant exist in the US and Europe. The best known integrated gasifier combined cycle (IGCC) is probably the Cool Water plant using a Texaco oxygen-blown entrained bed gasifier. This is

a demonstration plant, but future applications will be determined by economic grounds rather than technical ones. Cool Water operates at a thermal efficiency of about 31%, but as it is experimental any future commercial plants are expected to operate at much higher efficiencies of about 38%. A full description can be found in reference EPRI AP-3486.

The German VEW plant has run at 42% efficiency and is aiming at a full sized 800 MW plant at 47% efficiency [237]. Other plant are the KRW gasifier in Pennsylvania and the British Gas Lurgi gasifier at Westfield. Due to the low natural gas prices there is little prospect of large scale application until the end of the century. IGCC plant are more complex than conventional coal fired steam plant but are expected to generate at an efficiency of 3-4 percent higher and possibly up to 47% thermal efficiency, without as many environmental problems. They are also expected to be cheaper than conventional steam plant when including flue gas clean-up.

One concept is to use an integrated pressurised fluidised bed as part of a combined cycle. A fluidised bed is a bed of solids which is kept in a fluid state by a rising flow of gas. This can be used in coal combustion, and the coal can be mixed with a sorbent such as limestone or dolomite for desulphurization. The steam generator is immersed in the bed for heat transfer and bed cooling, giving a combustion temperature of about 750 to 950 °C. A set of cyclone separators reduce the solids from the flue gas. The sulphur reacts with the sorbent to form CaSO_4 which is removed with the ash and waste. A fluidized bed is characterised by low combustion temperature, low NO_x emissions, no slag formed, dry waste products, small tube wall surfaces due to high heat transfer coefficients, small plant, flexibility in coal type and shorter construction time than gasification schemes. A pressurized fluidizes bed (PFBC) has an even smaller size, and lower emissions.

Brown Boveri are marketing a PFBC combined cycle as shown in figure 4.5. It uses a bed at 850 °C, and uses either a high temperature process of 850 °C TET or a low temperature process of cooling to 400 °C TET. The higher TET has a net output of the GT of 20% of the total while the lower TET means the GT is just a pressure generator and the steam turbine produces 100% of the output. The preferred cycle is the low temperature process as this allows a simpler plant, less emissions and proven technology. The low temperature plant is 2% less efficient but 20% less expensive.

4.9 FUTURE FUELS

Environmental considerations and shortage of fuels will influence the selection of fuels in the future. There is an adequate supply of petroleum and natural gas for only 40 to 60 years, at increasing prices. Further oil reserves to be tapped are the oil shales and tar sands. Coal reserves are large enough for several centuries and it is expected that coal fuelled gas turbines will be used extensively. The expected fuels in the medium future will be;

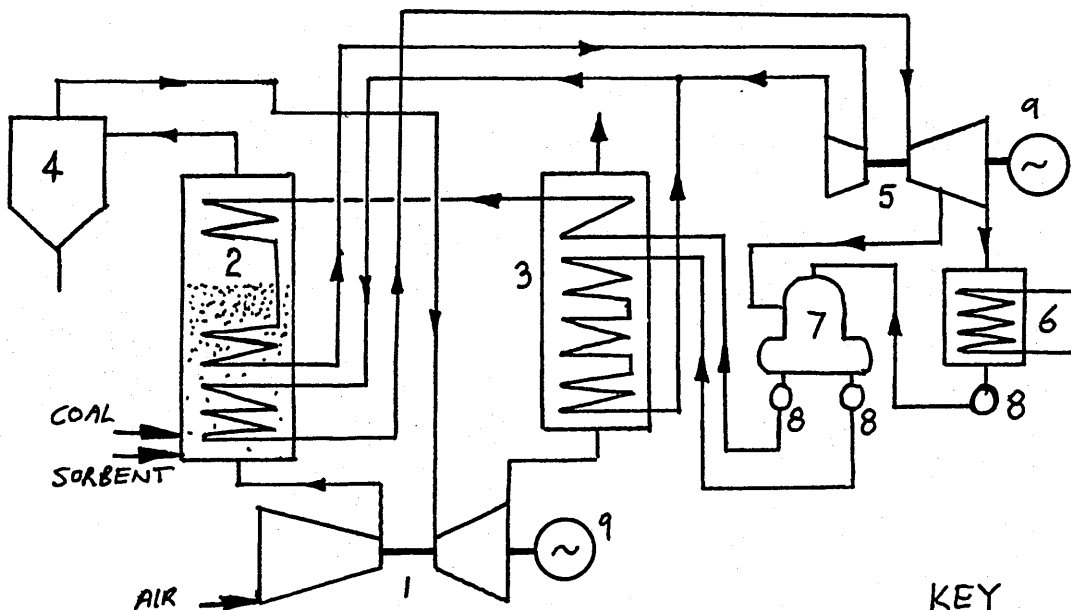
Solid	Fluidized bed with direct sulphur retention
Liquid	Petroleum products as at present
	Liquids from oil shales and tar sands
	Coal derived liquids
	By-products of chemical process plants

	Methanol
Gases	Natural gases
	High calorific gas from coal
	Medium and low calorific value gases from coal
	Process and refuse gases
Nuclear	Closed cycle gas turbines
Solar	Possibly closed cycle solar turbines.

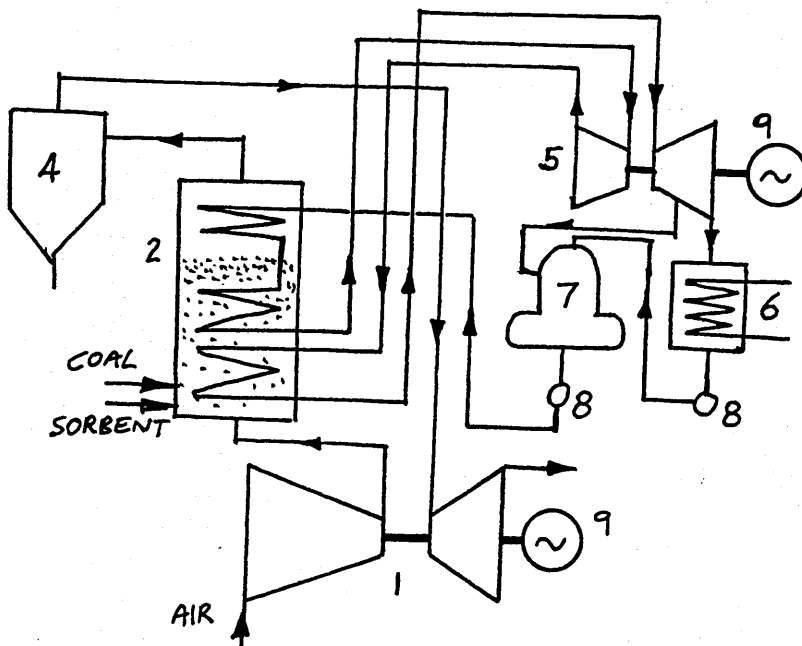
Many of the problems associated with future fuels have been identified and serious problems are not expected.- Obviously as new fuels become commercial they will have to be tested and adjustments made to combustors, nozzles and other equipment.

Figure 4.5 ABB Integrated PFBC Combined Cycle

4.5.1 HIGH TEMPERATURE CYCLE



4.5.2 LOW TEMPERATURE CYCLE



KEY

- 1 GAS TURBINE
- 2 PFBC
- 3 HRSG
- 4 DUST SEPERATOR
- 5 STEAM TURBINE
- 6 CONDENSER
- 7 DEAERATOR
- 8 PUMP
- 9 GENERATOR

5 OPERATION AND MAINTENANCE

5.1 INTRODUCTION

"Objectives in maintenance vary considerably from company to company, or even with time within a company. To define objectives or even a planned method will find disagreement." [108]. This quote indicates the wide variety of different approaches to maintenance that are taken. But common to all is that a prime consideration in maintenance is to minimise downtime, because unavailability costs can be high relative to the actual maintenance costs. Unavailability costs are directly related to the spare plant capacity needed and plant unit size. Typically annual expenditure for routine maintenance on a gas turbine may average from 10 to 20% of the turbines initial price.

"Few combustion turbines in service achieve the full design life potential of the turbine blading and nozzle vanes" [108]. Limited life components are usually designed for a life of 100,000 hours, based on creep or low cycle thermal fatigue. This is not achieved due to foreign object damage (FOD), domestic object damage (DOD), fouling, erosion, cooling hole blockage, imbalance, locking of blade roots, corrosion and others. Many of these are related to air quality.

The aim of maintenance is to minimise the operating cost by firstly reducing downtime by maximising availability and reliability. Secondly the aim is to maintain performance and life expectancy, and lastly to incorporate product improvements as necessary. Increasing availability and reliability is a function of unexpected plant trips, and the response to these. Even the planned maintenance time can be minimised through proper planning and training. Issues are often to do with maintainability of the engine and has lead to the term RAM (Reliability, Availability and Maintainability).

Fuel type has a major impact on maintenance. Approximate factors on maintenance costs are listed as follows;

Fuel type	% Hydrogen by weight	Factor
Residuals	less than 10%	over 3
Heavy Distillates	10 to 13%	2 to 3
Light Distillates	13 to 17%	1 to 2
Natural Gas	over 17%	1

PROBLEMS

5.2 PROBLEMS ENCOUNTERED

Compressor fouling

Approximately 70% of performance loss is due to compressor blade fouling [108]. Generally fouling is attributed to particles from sub-micron to about 10 micron size. This ranges from sea salt to industrial emissions to pollens. Also oil leakage on the inlet bearing can cause problems. The deposits inevitably reduce compressor efficiency, pressure ratio and mass flow, and either reduces output or increases TET to maintain output. The result is increased fuel consumption and reduced component lives. Fouling tends to increase the stage loading on the early stages and reduces the surge margin. The compressor can be cleaned by water,

water/solvent mixture or solid abrasive. However too frequent washing can reduce availability and possibly cause erosion, while too little washing can cost significantly in fuel and engine life and become more difficult to clean with just a water wash. Cleaning is not 100% effective and is discussed in section 5.3.

In practice the 1st stages are most fouled due to higher relative humidity at inlet when the gas pressure drops as it accelerates into the compressor. These conditions favour precipitation of dust. As temperature rises over the successive stages the deposit rate reduces. We can define a factor z_k such that;

$$\Delta Q / Q = z_k (\Delta \eta_{1c} / \eta_{1c}) \quad \dots 5.1)$$

where Q is volume flow rate of the machine and η_{1c} is compressor efficiency. Here z_k is highly dependant on the nature of the deposits and their distribution on surfaces [82]. z_{av} (average) varies depending on the location of fouling. It is less than unity if fouling is worst on early stages and greater than unity if fouling is worst on later stages. z_{av} changes with cleaning and can indicate if cleaning only effects the first stages or even redeposits the dirt on later compressor stages.

It is possible that the effect of fouling is a function of stage loading, and that high performance aero-derivatives may be more susceptible to fouling than their industrial counterparts. This is a finding of reference [83] but needs further experimental investigation.

Compressor erosion

Erosion is generally attributed to particles larger than 5 microns. Thinning of the trailing edge may initially increase the efficiency but the wear changes the blade cross section and decreases the fatigue life. Erosion resistant coatings are not fully satisfactory and the best solution is adequate filtering.

Cooling passage blockage

The build up of deposits is a function of the passage size, temperature and type of contaminant. The process can be undetected for long periods and lead to accelerated thermal fatigue, followed by cracking. The problem is greatly influenced by filtering and the location of cooling air off-take.

Rotor imbalance

Even a well balanced rotor can go out of balance after a partially successful cleaning, or where loosened dust mass moves in the rotor. There is the possibility that non-uniform deposits can lead to non-uniform cooling and thermal bowing.

Locking of Turbine Blade Roots

Some blade designs such as 'fir tree' roots require some freedom to ensure an even stress distribution. Trapped dust or corrosion has the effect of locking the roots, raising the stresses and changing the blade natural frequency.

Hot corrosion

The most common form of hot corrosion in gas turbines is due to deposition of alkali sulphates on the metal surface. Sulfidation occurs from sulphur in the fuel and sodium in the air, and it is worst in nickel based and some cobalt superalloys. It is related to the temperature of the blade and the gas stream. Hot corrosion is relatively insensitive to fuel borne sulphur levels but very sensitive to airborne sodium. To achieve a full life the sodium levels in the air would need to be less than about 15 ppb. Other solutions are to reduce fuel sulphur level, reduce metal temperatures, use improved metals or use corrosion resistant coatings. Corrosion is covered more fully in section 4.5.

5.3 CORRECTIVE ACTIONS

Air Quality and Filtration

The ambient air conditions and turbine filtration requirements vary considerably. It is important to filter correctly to reduce erosion, corrosion and fouling thus correct filtration has a large influence on maintenance and availability. It is best to test and sample air at the site over an extended period before construction and then use the experiences of the manufacturer and filter supplier to give the best solution.

Erosion problems are largely solved by inertial type filters which remove particles down to about 10 microns with a pressure loss of about 0.25 kPa. Corrosion and fouling problems require a high efficiency filter. It is typical to filter air to less than 0.05 ppm alkali metals. The high efficiency filter comes in two types, surface loading which builds up a cake of dust, and the deep bed which uses a torturous path of air passages. Surface loading filters can be of any virtually unlimited efficiency but at the trade-off in size and cost. Surface filters are sometimes cleanable but deep filters are generally not regenerable. Often high efficiency filter life is extended by use of medium efficiency prefilters, giving a combined pressure loss of about 0.5 kPa when clean and about 1 kPa when dirty. A two stage system of an inertial filter followed by a high efficiency filter is common. Filters in dirty environments, such as when susceptible to sand storms can use self cleaning filters which give a constant pressure loss of about 0.75 kPa.

The inertial filter needs checking regularly (say daily) and cleaning by wire brush and steam at least annually. Self cleaning filters need to be serviced annually and elements may give a 2 year life. High efficiency filters are changed on a pressure differential basis. Filters need bypass implosion doors incase of plugging, such as by ice.

Generally it is better to err in favour of a more expensive filter system rather than in an insufficient system, as the consequences in maintenance might be considerable

Compressor cleaning

The major portion of performance loss is due to compressor fouling. The problem can be detected by trending and when satisfied that the losses are from fouling then a visual examination through the front of the

turbine might be possible. The type of cleaning depends on the deposits. Hard deposits may be cleaned by insertion of abrasive solids, such as walnut shells, but which may cause problems in some engines so must be used to manufacturers recommendations. Water washing can be done on-line at reduced load or off line, and solvents are sometimes added to improve the effect. It is important to measure performance before cleaning to determine if cleaning was effective. To prevent deposits redepositing on the turbine it is usually washed after the compressor wash. Prior to some compressor cleaning methods it may be necessary to blank off instrument lines and air bleed holes.

Water washing is the most effective method of regaining most of performance degradation. Washing can be separated into two groups, on-line at reduced load and off line. Off-line water washing requires a cooling down period, wash, soak, and rinse followed by drying run, but it is more effective. Intervals vary greatly, but may be say 250 hours for an on-line wash and 1000 hours for an off-line soak wash.

Anti-corrosion Coatings

Nickel and cobalt based superalloys in hot sections of all types of gas turbines can be coated to enhance their resistance to hot corrosion. The most widely used coatings are those based on compounds of aluminium, formed by interaction of the superalloy surfaces to aluminium. Other compounds for coating have been used, including compounds of Co, Cr, Al, Y, Pt and Rh. There should be no need to schedule turbine overhaul based on corrosion life, as appropriate coatings have proved successful.

The four main methods of applying coatings are as follows;

- Pack cementation
- Physical vapour deposition
- Plasma spray
- Sheet cladding

Pack cementation is a diffusion bonded coat of aluminium, forming NiAl or CoAl. The component is heated in contact with a powder mixture of aluminium and an inert agent. Additives of chromium, platinum and silicon can be used to modify the coat. It is characterised by ease of application and bond integrity but has a limited life. A reference is [72].

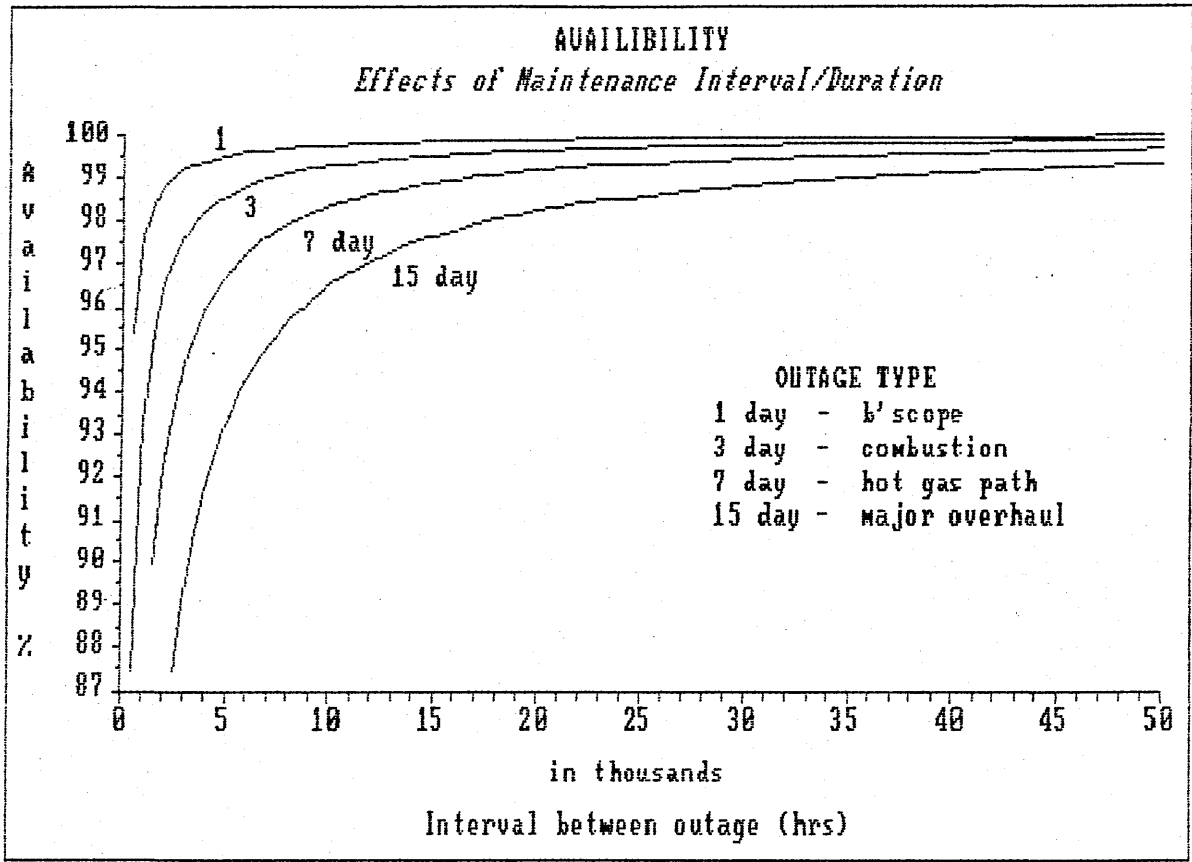
Physical vapour deposition occurs by vaporisation of a bar stock by an electron beam. It can deposit a wide variety of materials and to greater thickness than pack cementation, but it is more expensive.

Plasma spray uses a high temperature and high velocity plasma spray into which is injected a metal powder. Not only can protective coatings be applied but restoration of parent surfaces. It is cheaper than physical vapour deposition.

Sheet cladding uses a diffusion bonded layer of metal using hot isostatic pressing.

Evaluations of various coatings by EPRI have found that none of the coatings are totally resistant, and the coat degrades in localised areas [70].

Figure 5.1 Availability vs Inspection Duration and Interval



5.4 AVAILABILITY AND RELIABILITY

Availability can be defined in many ways. The definitions may or may not include waiting for spare parts, crew and other time consuming actions. High figures are possible when looking at a period when no major maintenance has been carried out, and availability is also effected by the usage patterns, fuel, filters and the environment. Therefore "reliability and availability statistics should not be seen in terms of absolute values but rather as indicators of trends and guides to problem areas" [108].

The various factors are defined here using the common definitions.

$$\begin{aligned} \text{Starting reliability} &= \frac{\text{Number of successful starts}}{\text{Number of attempted starts}} \\ \text{Operating reliability} &= \frac{\text{Period hours} - \text{forced outage hours}}{\text{Period hours}} \\ \text{Availability} &= \frac{\text{Period hours} - \text{all downtime hours}}{\text{Period hours}} \end{aligned}$$

$$\text{Load or use factor} = \frac{\text{Turbine fired hours}}{\text{Period hours}}$$

A successful start is a start from initiation to synchronisation without exceeding limits. All downtime includes all forced and all planned outage hours. It is vital that the operation be logged and reasons for outages or failed starts be recorded. The various factors can also give insights into the others. For instance a low load factor may mean no scheduled maintenance during a period of high demand and give an improved availability when needed.

5.5 MAINTENANCE INSPECTIONS

The interval between and duration of downtime effects the availability of the engine, but increasing the interval beyond a certain figure does not increase the availability significantly. This is seen in figure 5.1 where availability is plotted to inspection interval and duration. From this figure a simple rule of thumb can be made that if an outage lasts x days then it will impact availability greatly if it occurs more frequently than x thousand hours, but obviously an outage will be minimised regardless of frequency. Aero-derivatives and heavy industrial engines have different maintenance characteristics so will be treated differently.

For heavy engines maintenance generally falls into one of various categories. These are listed here with typical duration and timing between events.

- Running inspections and data recording
4 hr, twice per shift
- Borescope inspections
4000 to 8000 hr lasting 1 day
- Combustion inspections
8000 to 16000 hr lasting 3 days or more
- Hot gas path inspections
15000 to 25000 hr lasting 7 days or more
- Major inspections
30000 to 45000 hr lasting 15 days or more

The maintenance interval and duration varies depending on the operating conditions, fuel type, load cycle and who is giving the figures, their experiences and bias. Figures claimed vary even within the manufacturing company. The above figures are from reference [108]. The figures from GE for the MS7001 [275] are;

Combustion inspection	interval	Peaking 3000 hr or 400 starts Base-load 8000 hr
	duration	64 hr (6 men)
Hot gas path	interval	Peaking 12-16000 hr or 1200 starts Base-load 20-24000 hr
	duration	160 hr (7 men)
Major	interval	Peaking 24-32000 hr or 2400 starts Base-load 40-48000 hr
	duration	320 hr (8 men)

These times assume replacement only of parts (no repair) and all tools

and parts are available. It does not include the generator and accessories. The labour is direct labour only and is for continuous 24 hr/day work in three shifts, thus three times this number of tradesmen are needed and also supervision. Other assumptions are that the crew is of average trade skills and adequate planning is made. Times can be effected largely by tooling as typically 25% of hot section inspection downtime is consumed in removing and replacing bolting.

For aero-derivatives the maintenance practice is different. Aero-derivatives like the RB211 and LM2500 have modular or semimodular construction and allow replacement of faulty modules with reconditioned or hired modules. Besides the visual inspections, the LM2500 for instance has no predetermined schedule for overhaul, the hot section repair is based on condition monitoring. It is estimated the hot section repair interval will be about 25000 hours for natural gas base-load.

Typically the on-site maintenance for the aero-derivative consists of external preventative actions and partial disassembly, with replacement of modules and possibly blades/vanes. Maintenance such as complete disassembly, inspection, refurbishment and repair is typically off-site. While major maintenance actions are occurring the engine can be replaced by a refurbished trade-in, a spare or a leased engine. Replacement of an aero-derivative component can be achieved in very short times, thus giving an availability of over 98%. For instance for the LM2500 it is claimed that replacement times are as follows;

Replaced Item	Hours	Men
Gas Turbine	6.8	5
Gas Generator	10.0	5
Power Turbine	9.5	6
Hot Section Components	19.0	6

These are ideal times for a trained and prepared crew to replace a part of the engine with a replacement component.

Unplanned outages may result from an engine showing signs of distress or measurements exceeding allowed levels. Distress includes items such as;

- * high vibration or a shift in vibration
- * high exhaust temperature differentials
- * high lube oil consumption
- * ingestion of objects
- * sudden changes in performance
- * sufficiently decreased fuel efficiency

The duration of outage for a damaged engine may be significantly longer than those claimed for routine maintenance. Figures for heavy engines from an insurance company, Huppmann, Allianz, Germany (1979) are as follows;

Turbine rotor and stator blades	80+ days
Compressor rotor and stator blades	60+ days
Combustor and hot ducts	2-30 days
Bearings	5-30 days
Exhaust Duct	30 days
others	1-10 days

Huppmann also provides a breakdown on the causes of failures which are summarised as follows;

	1970-75	1975-79
Production faults	70%	73.1%
Operation faults	26%	10.6%
External influences	4%	16.3%

Availability and reliability performance is much improved by improvement to fuel and air quality and by minimising repair time by proper planning. Planning a major overhaul would typically start 12 months before the date to ensure all things run smoothly. Reducing undesirable trips can be achieved by good quality instrumentation. Redundant instrumentation and having accessories maintainable on-line can also improve availability.

Inspections are done by borescope or by opening the casing. More major inspections may include removal of all nozzles and blades for testing, as it is not really possible to tell the condition of the component until it is removed and cleaned. Some major work may be quicker in a vendor overhaul facility, and some aero repair facilities also do industrial work. Inspection criteria is a balance between running a component to full life and risking a failure. The consequences of a failure must be measured as this can far outweigh any benefits of additional life. Failure may render the component unrepairable and damage further components. Often a removal while still in reasonable condition can allow refurbishment to near or as new condition by a repair agent. Inspection methods include

- visual - borescope, direct and magnified
- Dimensional - wear, distortion and location
- Crack - die penetrant, magnetic particle, ultra sonic and radiographic
- Analysis - oil testing, mechanical tests on sample blades

5.6 MAINTENANCE ACTIONS

Repair

Repair is generally possible with modern repair procedures, and statements that components are 'not repairable' require more qualification by the manufacturer. For small operators it is recommended [108] that consultant specialists be used. Repairs that can achieve major savings are combustors, transition pieces, rotor blades and NGV assemblies. Typically the cost of repair to as new or near new condition may be 30 to 60% of replacement cost. Repair also reduces lead time for ordering some parts, reduces spares inventory, and reduces world consumption of scarce metals used in superalloys. Parts purchase and repair must be from vendors with the appropriate equipment, not just for repair but also for verifying the repairs integrity. The operators must establish proper specifications and insist on full material recertification. Vendors need to give warranties and have liability for their work.

Repair techniques include welding to replace sections of a damaged component with a section of another component, hot isostatic pressing (HIPing) to close internal voids and restore fatigue life, and plasma

spraying to restore surfaces lost to erosion, corrosion or wear. Various other techniques are also available including heat treatment, grinding, fusion welding, diffusion bonding, sintering, electron beam welding and straightening.

Vibration

Vibration analysis is measuring the motion of the machine. Measurements are often amplitude, such as mm peak to peak, or acceleration, in g's. Fourier transforms can be done by specialist equipment to break the complex motion into a sum of simple sine waves. The frequency spectrum obtained is a signature or fingerprint of the machine's vibration. Frequency spectrum can be measured and compared over time to give an indication of changes in level for individual frequencies. Problem frequencies can be compared to known excitation frequencies to attempt to discover their source. Examples are unbalance, misalignment, oil whip and for aero derivatives ball bearing rotation and passing frequencies. Frequency trends can give a prior warning to problems.

Balancing

Industrial gas turbines often have the facility to balance the rotor on site. Typically there are 5 balance planes, and as most overhaul shops are unable to balance at full speed further trimming is needed in the field. But some balance points may be inaccessible after the casing is replaced and thus field trimming may be limited. Measurements are by vibration pick-ups, proximity probes and key phasors.

Oil

"Oil is the lifeblood of the turbine and, as such, must be monitored and maintained" [108]. It lubricates and cools the bearings, and also gives an indication of bearing condition. Oils can be analysed at intervals to determine contaminants and levels of additives.

Training and Personnel

"The most effective tool available for preventative or corrective maintenance is well trained and qualified operators" [108]. Training may consist of visits to the manufacturers training center, class room and instructor based on-site instruction. Vendors should be able to supply a modular video tape learning system complete with supervisor guides. Training is particularly important for small operations where the operators get less experience. This is especially important when maintenance times may be well separated and be preceded by many hours of low key activities.

Due to the close interfaces between the GT and the auxiliaries a single team is more likely to cooperate fully. Also experience has shown that good performance is obtained by a team specifically allocated to work on the plant and which operates and maintains the plant. The team then has a direct personal involvement in the success of the operation.

Vendor

The supplier should provide some sort of feed back facilities for problems, and product service bulletin updates. On going support, supply

of spare parts, and overhaul facilities are needed. As maintenance costs may be 10% of the purchase price per year and because downtime can be disproportionately effected by shipping times for components, the choice of engine supplier and contractual obligations might be as important as the engine performance.

Maintainability

The engine must be designed for maintenance, which must be addressed at procurement level. Ease of access and space are necessary, especially to high maintenance items such as fuel nozzles, combustor and first stage nozzles. Some engines have a separate cover for the first stage nozzles to allow inspection without disturbing the rest of the turbine. Access means permanent ladders, walk-ways, handrails, hinged inspection hatchways and other attention to detail. For base-load plant accessories and instrumentation must be able to be maintained without shutdown. This means dual filters and pumps, and access to items such as thermocouples.

Other items necessary for good maintainability are spare parts, tools and adequate documentation. Special tools may be needed and must be available, and documentation should include fault isolation logic sequences, task description and part lists. All this must be at a technicians level.

5.7 ON-CONDITION MAINTENANCE

'On-condition' maintenance as opposed to 'hard time' or 'fixed time' maintenance is where the maintenance is not scheduled on a strict time interval or running time but on the actual condition of the engine. It is based on the premise that time based maintenance must involve unnecessary downtime and expense in maintaining the engine before degradation has occurred.

On-condition maintenance involves monitoring engine operating parameters closely to determine when an engine is operating inefficiently or is in distress. Various performance monitoring practices have been developed where measured parameters such as pressures, temperatures and vibrations are collected and compared in trend plots or with other methods of analysis.

The on-condition techniques use the measurable 'dependant' parameters to estimate the unmeasurable 'independent' parameters such as component efficiencies. There are several techniques including Fault Matrix, Gas Path Analysis, Trending, Parameter Change Trees and Knowledge Based Systems. A performance simulation program can be used to find the measurable effects of small changes to engine parameters. A fault matrix can then be formed similar to the following example;

	TET	Power	m	rp	Vib.
Nozzle Erosion	up	up	up	down	No
Rotor Damage	up	up	up	up	Yes

The problem is that only one degradation can be present at one time for the method to be reliable. Gas path analysis is a technique where a set of linearised equations are formed linking the small changes of the independent and unmeasurable variables to the dependant measurable ones.

The matrix formed is called the 'Influence Coefficient Matrix'. The measurements and influence coefficient matrix are solved to give an estimate of the engine condition. It's use in practice can be very complex as the matrix may be larger than 30×30 , and the instrument noise and inaccuracies may be as large as the changes to be measured. In order to be successful the base line has to be measured for each engine in it's installed position with the instruments to be used for monitoring.

Currently the typical practice is to use trend plots. Parameters measured usually include speeds, fuel signals, vibration signatures, wheel space temperatures, exhaust temperatures, start/shutdown benchmark times and other variables. Data must be recorded at a steady state, which may mean minimum variation in say wheel space temperatures over a 15 minute period. Trend plots are not the best answer insofar as maintenance scheduling is concerned, because of the difficulties in interpreting what the deviations in parameters mean in terms of actual engine duress. Normal condition monitoring parameters in the LM2500, for example, are;

- * Gas generator speed
- * Power turbine speed and load
- * Gas generator exhaust temperature
- * Gas generator inlet conditions
- * Vibration
- * Oil pressures and temperatures
- * Variable stator vane angle
- * Fuel pressure

Modelling of deteriorated engines is complex but an established technology and can be made available. For an aero-derivative the use of gas path analysis can allow detection of degradations at the module level. The techniques can also assist in assessing the consequences of operating with deteriorations, to component lives.

6 STEAM TURBINES

6.1 INTRODUCTION

The steam turbine has been around as we know it since 1884 and the first commercial power generation was in 1890. Many early pioneers were involved in developing the turbine including notably C.A. Parsons, de Laval, Curtis and Rateau. Early turbines developed along various lines, and not until the second world war was there a serious attempt to standardise the operating conditions and certain physical characteristics. Standardisation was not so rigid to prevent improvements which have continued endlessly.

In the first forty years as many as thirty seven varieties of turbine appeared, but from the beginning two major categories emerged. The first is the so called impulse turbine, where the fixed stages convert the heat to velocity and there is no pressure drop over the rotating blades. The moving blades just change the direction of the velocity. The second type is the so called reaction turbine where some of the conversion of heat to velocity is done in the moving blades, and a pressure drop occurs.

Steam conditions have increased steadily until recent years with operational plant above 920 K. Practical turbines have used a maximum of about 840 K but most turbines have settled on 810 K for a number of years. The sizes have increased up to 1200 MW, but has stagnated at these large sizes largely for cost and planning reasons. The technological limitation on size is due to the size of the last blade row, and this annulus area limited by blade stress. Sizes up to 2000 MW are possible with current technology.

Pressure increases cause an increase of cycle efficiency but can cause increased losses due to small blading if the rating is not increased as well. Pressures of 24 MPa have been used in large plant although smaller combined cycle plant are more likely to be 4 to 8 MPa. Increased pressures have also been accompanied by reheat to reduce the low pressure stage blade wetness. Wetness, or condensing liquid, in low pressure stages causes erosion and reduced blade efficiency and so must be limited.

Efficiency has been the driving force in improvement, and is largely due to increases in steam conditions, increase in rating, increase in blade efficiency, use of reheat and feedheating. As a rule of thumb a 1% increase or decrease in efficiency is equivalent to 10% of the purchase price. The best practical efficiencies in 1923 were 17.8% and in 1953 30.4%. Currently the practical efficiencies are about 37-38% for a steam temperature of 810 to 840 K. Increases in efficiency are currently technically possible but at a price, by reducing condenser pressure, increasing temperature, reducing pressure drops in pipe work, increasing feedheater numbers, and others. Efficiencies have been demonstrated as high as 41.6%, but in reality these plant are not used. It is usual that the most efficient plant is not the most economic. Moves in the future will be towards supercritical cycles, using double reheat and steam temperatures of 890 to 920 K and give efficiencies of 39 to 41% for pulverised coal and flue gas desulphurization (PC/FGD).

Figure 6.1 Steam Turbine Cycle - Temperature vs Entropy

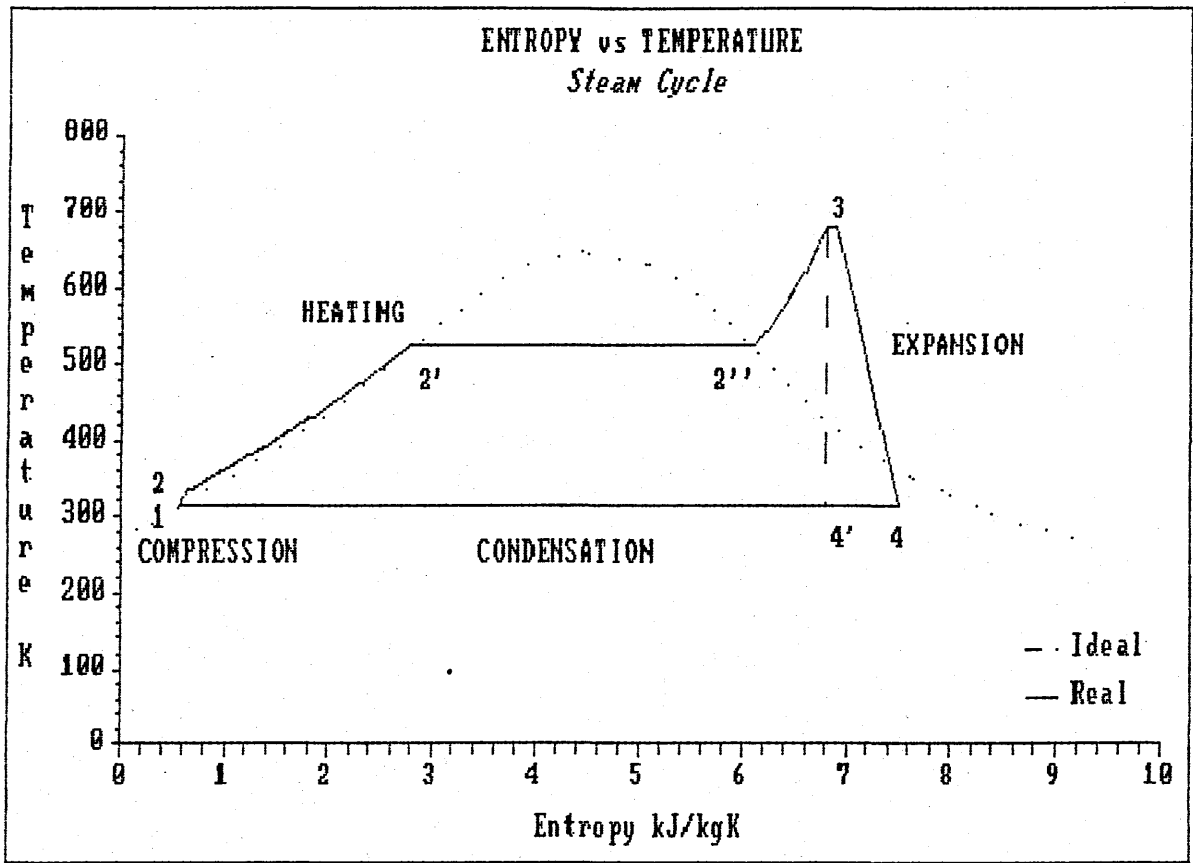
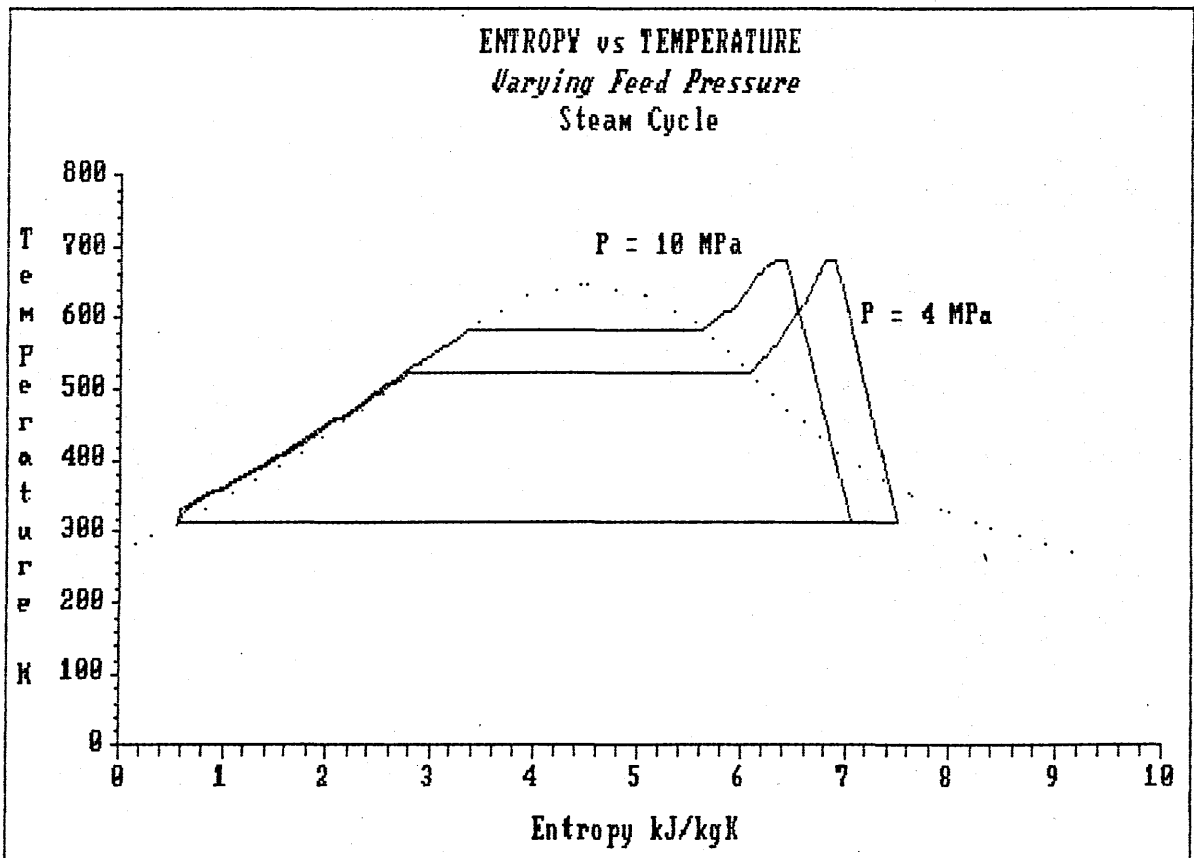


Figure 6.2 Temperature vs Entropy - Varying Maximum Pressure



Modern blade profiles show a much reduced increase in losses with changed steam inlet angle. This improves part load efficiency. The design point efficiency can also be compromised to improve the part load efficiency. Although development is continuing the steam turbine is a mature technology and significant improvements are slow, with steam conditions stagnating for several years.

6.2 STEAM TURBINE CYCLE

The steam turbine cycle consists of compressing a cool liquid water to high pressure. This water is then heated, changing it's phase and then heated again to a superheated vapour. The process of heating the liquid water is called feedheating and is done in the feedheaters or economiser. The boiling is done in the evaporator and the superheating in the superheater. The high pressure and high temperature steam is then expanded in a turbine to extract work. The pressure and temperature at inlet to the turbine is called the throttle conditions. The steam is discharged into a condenser, which is at a vacuum relative to atmospheric pressure, and then condensed back to cool water.

The cycle is shown in figure 6.1. If the kinetic energy changes and isentropic inefficiencies in the turbine are ignored then the heat transferred is represented by the area under the T-S diagram. This means to maximise the thermal cycle efficiency the area under the line 2-2'-2''-3 which is heat input must be maximised while the area under 1-4 or heat rejected must be minimised.

6.2.1 Analysis

The fluid properties h and s give the work and heat transfers by using the energy equation and assuming adiabatic conditions and negligible changes in kinetic energy. This is approximately correct as the heat losses through pipes and turbine walls are small relative to the heat flows through them, and the turbine efficiency definition includes the kinetic losses in the turbine exit. The properties are given in steam tables for all the ranges of pure vapour and liquid. In the region of mixed phase the 'quality' is defined as follows;

$$\text{Quality } x = \frac{\text{Mass of vapour phase}}{\text{Total mass}} \quad \dots 6.1)$$

This leads to the following;

$$h_{\text{MIXED}} = h_f + x(h_g - h_f) \quad \dots 6.2)$$

$$s_{\text{MIXED}} = s_f + x(s_g - s_f) \quad \dots 6.3)$$

where f is saturated liquid and g saturated vapour,
and h and s are enthalpy and entropy respectively.

The method of calculation for the steam turbine cycle is as follows. Notation is from figure 6.1.

$$\text{Pump work is; } W_{\text{PUMP}} = \dot{m}_s(h_2 - h_1) \quad \dots 6.4)$$

$$\text{but specific work } w = \int_1^2 v \, dp \quad \text{and } v \approx \text{constant} \quad \dots 6.5)$$

thus ideally specific pump work $W_{PUMP} = V_f(P_2 - P_1)$ and $s_2 = s_1$... 6.6)

As the work is small relative to other heat and work processes it is within the desired accuracy to just include an approximate isentropic efficiency, η_{1P} , on this work.

$$W_{PUMP} = \dot{m}_S V_f (P_2 - P_1) / \eta_{1P} \quad \dots 6.7)$$

The properties at outlet of the boiler are found from the steam tables for the temperature and pressure at outlet. A loss in pressure is included after the boiler for piping and throttling which modifies the outlet entropy and therefore enthalpy.

Throttle pressure $P_3 = P_2(1 - x_P)$ where x_P is the loss fraction ... 6.8)

The boiler heat input is $Q_{BOILER} = \dot{m}_S(h_3 - h_2)$... 6.9)

The turbine work is found from the ideal work. The ideal work is where entropy is constant and there are no irreversibilities. Thus;

$$\text{isentropic expansion} \quad s_4' = s_3 \quad \dots 6.10)$$

$$\text{isentropic quality} \quad x_4' = \frac{(s_4' - s_f)}{(s_g - s_f)} \quad \dots 6.11)$$

$$\text{isentropic enthalpy} \quad h_4' = h_f + x_4'(h_g - h_f) \quad \dots 6.12)$$

The actual entropy is found from the ideal entropy drop and the isentropic efficiency η_{1t} .

$$\text{Enthalpy drop} \quad (h_4 - h_3) = \eta_{1t}(h_4' - h_3) \quad \dots 6.13)$$

The actual quality is found from the real enthalpy;

$$\text{quality} \quad x_4 = \frac{(h_4 - h_f)}{(h_g - h_f)} \quad \dots 6.14)$$

The turbine work is found from the enthalpy drop;

$$\text{work output} \quad W_{TURBINE} = \dot{m}_S(h_3 - h_4) \quad \dots 6.15)$$

The condenser rejected heat is found from;

$$\text{condenser} \quad Q_{REJECT} = \dot{m}_S(h_4 - h_1) \quad \dots 6.16)$$

Overall cycle efficiency

$$\text{steam cycle efficiency} \quad \eta_{ST} = \frac{W_{TURBINE} - W_{PUMP}}{Q_{BOILER}} \quad \dots 6.17)$$

The mathematical model is summarised as follows;

h_{IN} , s_{IN} , and $\dot{m}_S$ are known from throttle pressure and temperature, and HRSG output. Condenser pressure or temperature gives s_f , s_g , h_f , and h_g at outlet.

$$\begin{aligned}
 x_{out}' & \text{ from equation 6.11, } x_{out}' = (s_{in} - s_f)/(s_g - s_f) \\
 h_{out}' & \text{ from equation 6.12, } h_{out}' = h_f + x_{out}'(h_g - h_f) \\
 \Delta h' & = h_{in} - h_{out}' \\
 \Delta h & = \eta_1 \Delta h' \\
 h_{out} & = h_{in} - \Delta h \\
 x_{out} & \text{ from equation 6.14, } x_{out} = (h_{out} - h_f)/(h_g - h_f)
 \end{aligned}$$

The above theory is used in the combined cycle model as discussed in the combined cycle section. The mathematical model is applied in appendix A.

6.2.2 Variables and Improvements

Exhaust quality;

The exhaust quality represents the amount of liquid in the steam in the low pressure stages. The liquid water tends to decrease the stage efficiency and introduce erosion problems. Droplets tend to move slower than the main stream and cannot turn as sharply striking blades. The minimum quality acceptable is approximately 87.5-90.0%, but tolerance varies with the low pressure stage design. A rule of thumb is a loss of 1% stage efficiency per 1% reduction in quality.

Inlet Pressure;

The higher the inlet pressure the higher the line 2-2'-2''-3 on figure 6.1. This is shown in figure 6.2. It can be seen that for constant superheat temperature the area enclosed remains approximately constant while the rejected heat reduces. This indicates an increase in cycle efficiency. A problem is reduced quality of exhaust reducing the low pressure stage efficiencies. Another problem is the increased pressure means smaller high pressure blades and reduced high pressure blade efficiency, coupled with the tendency for higher pressures to have lower stage efficiencies anyway. Large steam turbine plants use pressures of up to 24 MPa, but with reheat.

Condenser Pressure;

The exhaust pressure is close to the condensing pressure which is dictated by the condensing temperature. This is in turn dictated by the cooling water temperature and the condenser heat exchanger size. In practice a 10-20 °C temperature difference is needed between the cooling and condenser temperatures, thus the condenser pressure is normally in the range of about 3-12 kPa absolute. The process used for rejecting the heat affects the pressure such as cooling towers, sea water, river water or air cooled heat exchangers. The lower the condenser pressure and temperature the smaller is the area below line 1-4 on figure 6.1 and the higher the cycle efficiency. The side effects are slightly lowered quality of exhaust and therefore slightly lowered low pressure efficiency on the turbine. But the main side effect of lowered pressure would be to require an increased size or speed of the low pressure stage.

Superheat Temperature;

The higher the initial temperature the larger the area under 2-2'-2''-3 on figure 6.1. This is shown in figure 6.3 and means a higher cycle efficiency. The higher temperature will also increase the exhaust quality and improve the low pressure stage efficiencies. The maximum temperature is limited by material strength. Currently the highest economical temperature has settled on 810 K, but a few turbines operate up to 840 K and turbines have been run at 920 K. Increased pressure has

Figure 6.3 Temperature vs Entropy - Varying Maximum Temperature

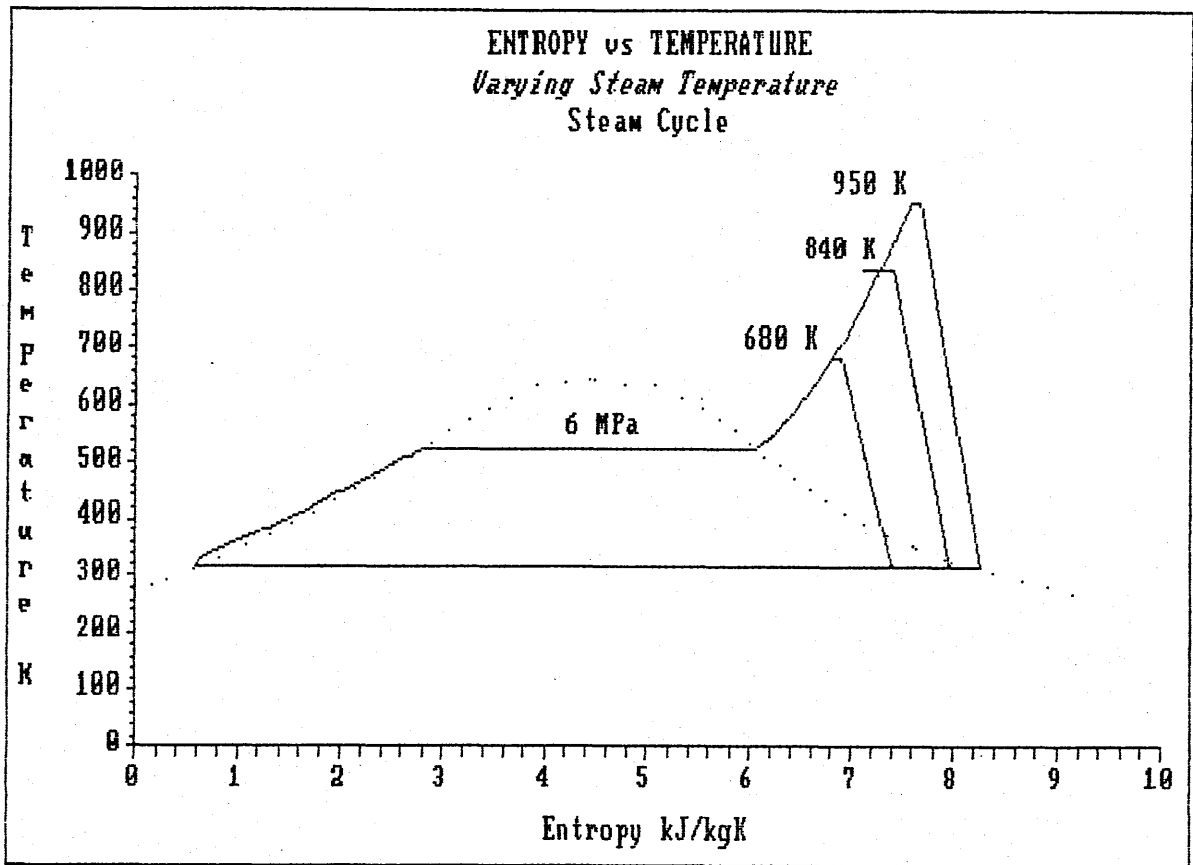
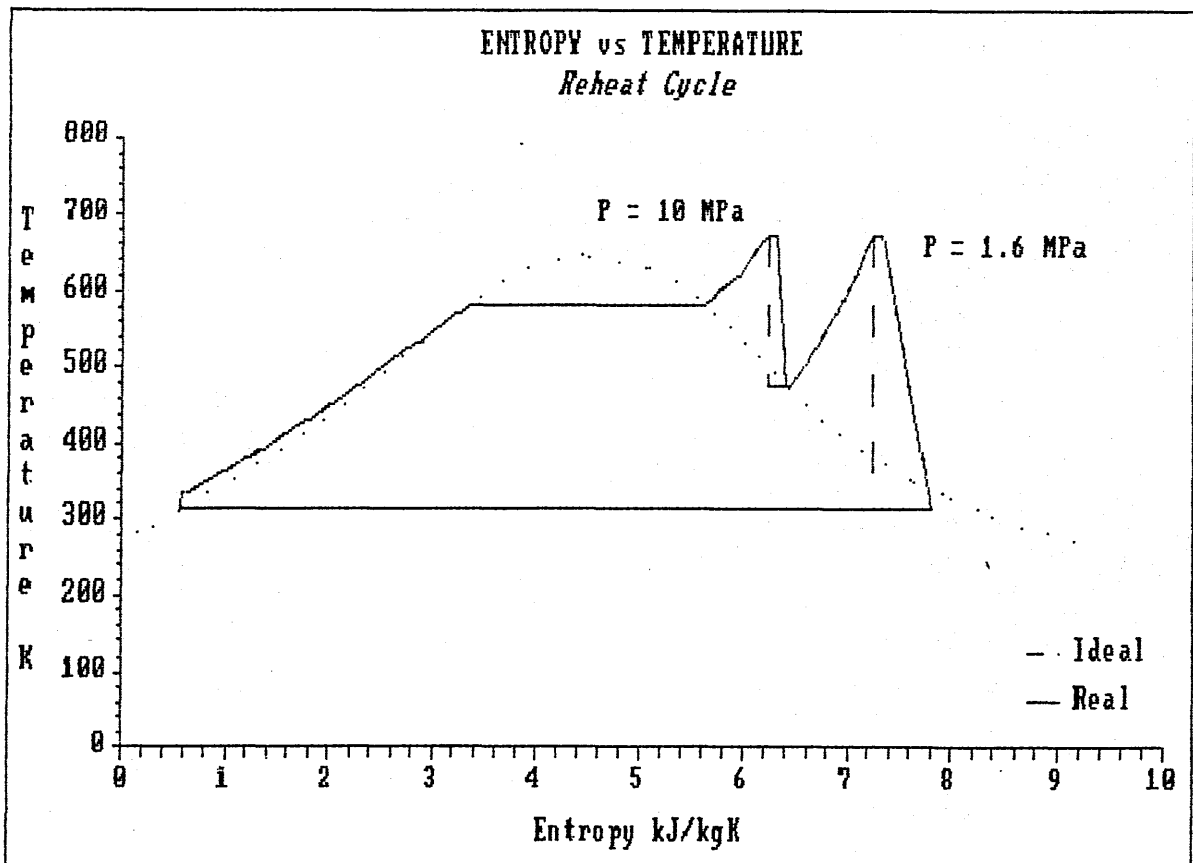


Figure 6.4 Temperature vs Entropy - Reheat

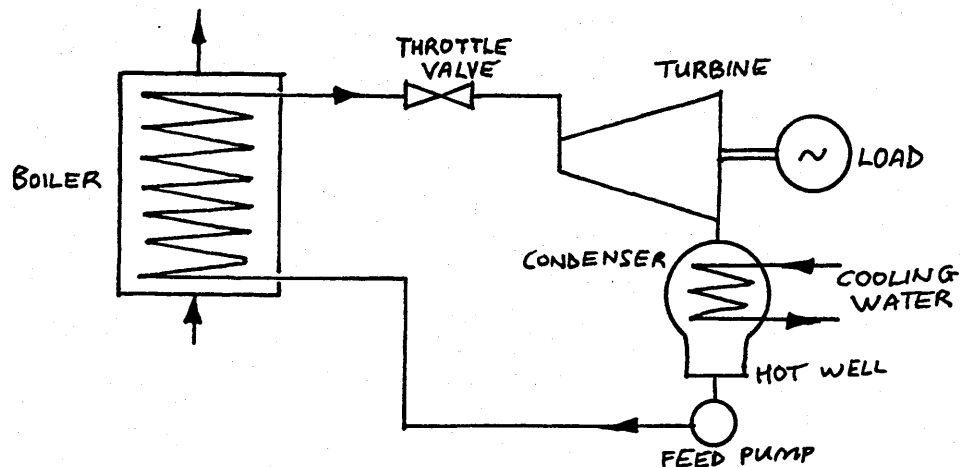


paralleled the improvements in temperature as the exhaust quality is a limiting variable.

Throttling:

Pressure is lost in the pipe work from the boiler to the turbine, and in the control valve. This generates entropy and reduces the available work. The inlet point of the turbine moves to the right and slightly down on the T-S diagram, figure 6.1. This increases the quality of the exhaust slightly. The lost work or Δh_4 is approximately $T_3 \Delta s$.

Figure 6.5 Simplified Steam Cycle



Improvements to the Simple Cycle

Reheat:

Reheat is the addition of more heat to the steam part way through the expansion. This is shown on figure 6.4 and involves piping the steam to and from the boiler a second time. The selection of reheat pressure must be selected carefully to prevent lowering the average temperature of heat addition. If the parameters are chosen correctly the cycle efficiency can be improved. The main improvement is the increased quality of exhaust and this allows significantly higher initial pressures. Reheat is usually required for throttle pressures in excess of 8-10 MPa. Typically the reheat pressure is about 15-35% of the inlet pressure. Double reheat is used in the very high pressure plant, where reheat is at typically at 30% and 10% of the pressure. Reheat is expensive in capital cost of the extra piping, controls and heat exchangers. The extra steam stored in the pipe work will require extra regulating valves to prevent over speed by stored steam in event of sudden load shedding. The extra pipe work will also be of larger size due to the reduced pressure, and involves extra pressure losses. These complications and costs have excluded practical triple reheat plant.

Regenerative Feedheating:

Regeneration or steam extraction for feedheating is where some of the steam flow is extracted from the turbine at various temperatures, part way through the expansion, to heat the feedwater. This heating is an attempt at reducing the temperature differences of heat transfer and so reducing the irreversibilities. A large modern steam plant may have 6-8 feed heater extraction points and the same number of heaters. This may represent up to 20-30% of the total steam flow. In conventional boilers

air preheaters in the boiler mean the degree of regeneration does not effect the stack gas temperature and thus does not effect the boiler efficiency. In a heat recovery steam generator the heat recovery efficiency may be reduced by regenerative feedheating and the overall effect must be optimised. Extraction reduces the size of the condenser, and the size of the last stages of the turbine.

6.3 TYPES OF TURBINES AND COMPONENTS

The turbine uses the steam pressure to transform thermal energy to kinetic energy and a change in momentum to provide forces on the blades. The major classifications are Impulse and Reaction.

The impulse turbine allows use of disks on a small diameter shaft as there is no pressure difference to transmit axial forces on the shaft. The disks can even include holes to equalise the pressure. Reaction turbines usually have blades mounted on drum type rotors to reduce axial thrusts on the shaft. In reality the degree of reaction, or heat drop in the moving blade increases from root to tip unless a twisted blade is used. The use of twisted blades is frequently not economic in high pressure stages and so the reaction varies. Current practice is to use root section reactions of up to 20% in 'impulse' turbines and reactions as low as 30-40% at the root of 'reaction' turbines. The long and twisted blades in low pressure stages are nearly indistinguishable between reaction and impulse machines and further reduce the distinction. This means that turbines are better separated into disk (diaphragm) types of low reaction and drum types with high reaction. Both types represent large sections of the current turbine population, with no clear superior form. Comparisons of the two main categories of impulse and reaction turbines are as follows. For the same work impulse stages require a higher velocity thus more frictional losses, but in practice blade speeds are at their maximum and reaction blades tend to have less work per stage. Impulse blades have lower tip losses due to low or zero pressure differences over the blades. The seals on impulse turbines also tend to be on the smaller diameter shaft and therefore reduce seal losses here to.

Other classifications are pressure compounded, Rateau, and velocity compounded. Velocity compounded impulse or Curtis stages use one stage of nozzles then several stages of impulse blades. The stators in these later stages just redirect the flow. The velocity compounded stages are of low efficiency. These stages are sometimes used as the first stages as the penalty in efficiency may be worth the rapid reduction in pressure and temperature.

Turbines may use a velocity compounded first stage to reduce the temperature and pressure quickly in the first nozzle. This reduces the pressure losses such as leakage and temperature effects such as creep on the rotors. The next few stages may be impulse stages to minimise tip leakage at the higher pressures, at the expense of high velocities and the remaining stages may be reaction stages for lower velocities and higher efficiency.

Partial arc admission;

This is used on small turbines where full admission would cause impractically short blades, and is used for part load control on many larger turbines. It is not practical in a reaction turbine stage.

Condensing, Back Pressure and Extraction Turbines;

Condensing turbines exhaust at very low sub atmospheric pressures, into a condenser. The back pressure turbine exhausts at an elevated pressure, usually to supply steam for a process in cogeneration. An extraction turbine is a condensing turbine but where higher pressure steam is extracted through bleeds part way through the expansion.

Deaerator:

At high pressures aggressive dissolved gasses such as carbon dioxide and oxygen need to be removed. Above 3 to 4 MPa the level should be < 20 ppb. The level of gases is more critical as the pressure increases.

Air extractor:

To maintain a vacuum in the condenser the gases must be continually condensing. Any noncondensable gases must be removed, such as entrained air. This is achieved by an air ejector as shown in figure 6.7 or by use of air pumps.

Some of the various layouts and types are shown in figure 6.8 and 6.9.

Figure 6.6 Feedheaters

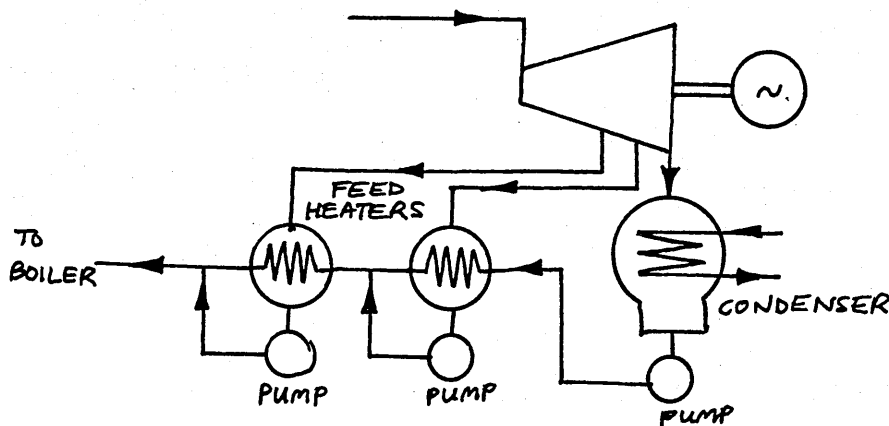


Figure 6.7 Air Extraction

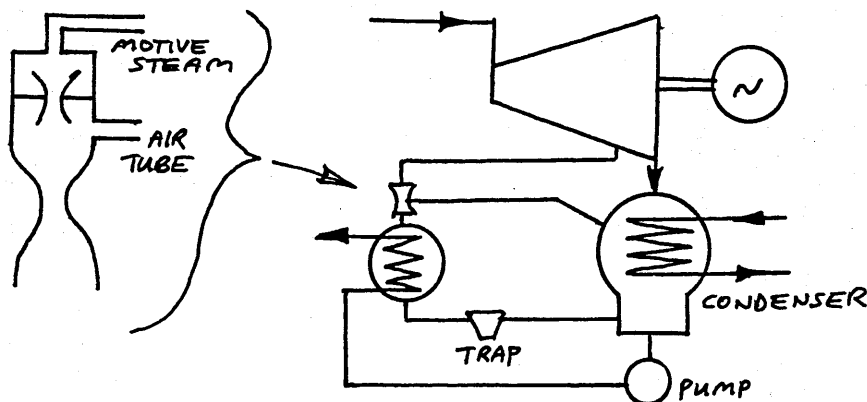


Figure 6.8 Turbine Layout

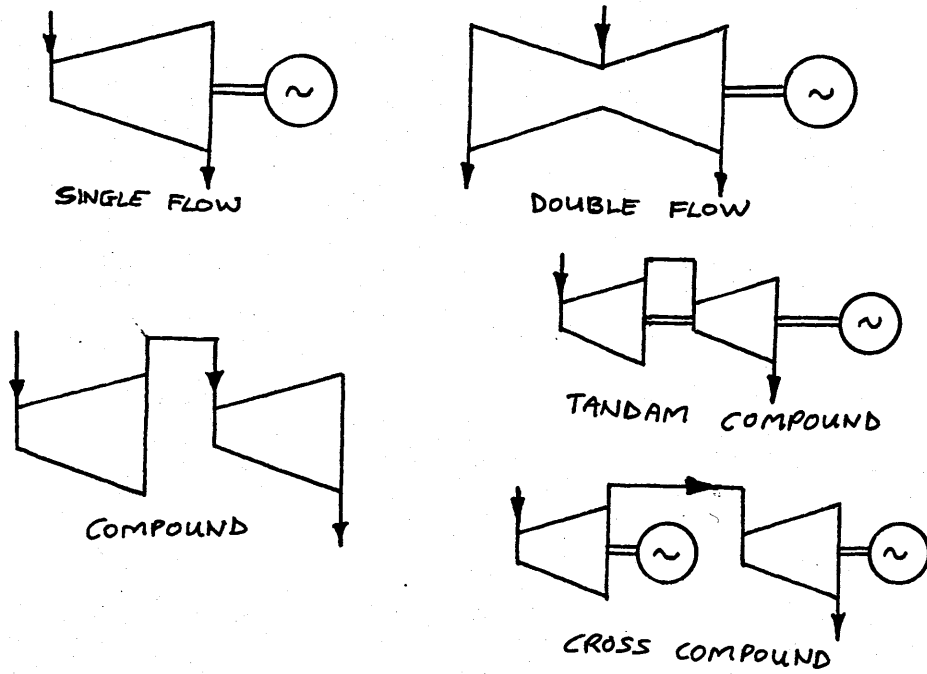
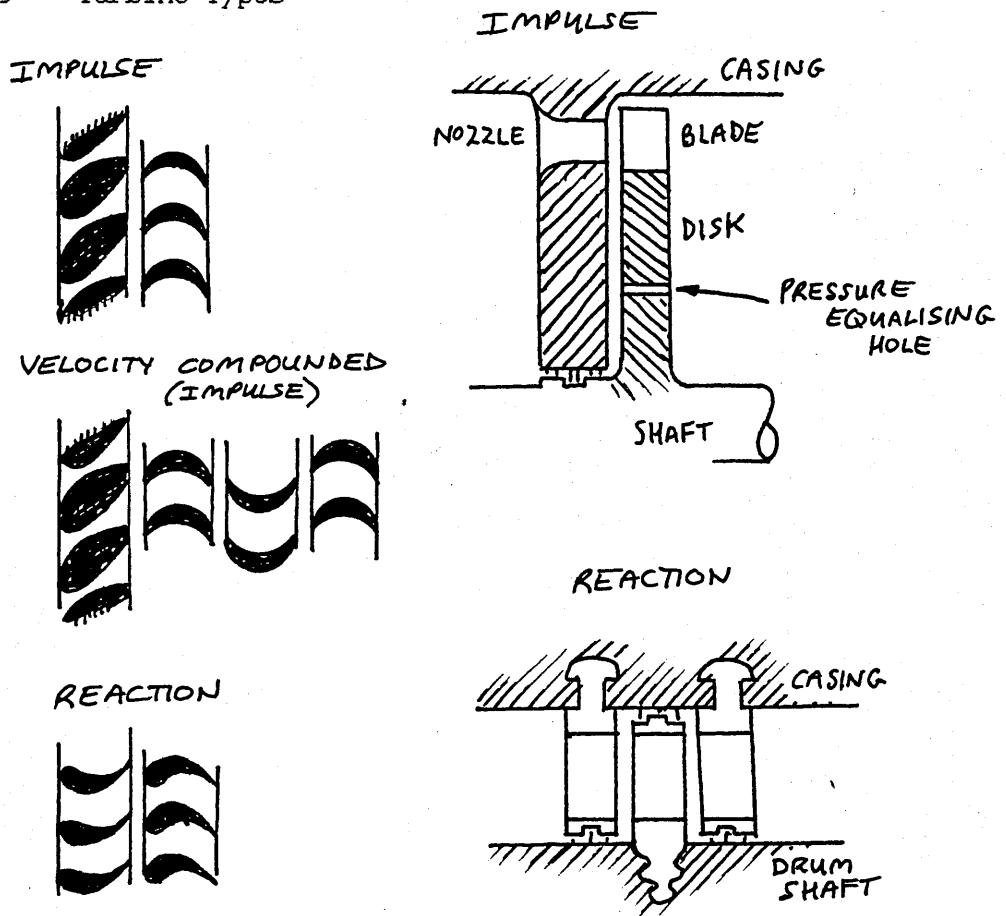


Figure 6.9 Turbine Types



6.4 PART LOAD PERFORMANCE

To understand part load behaviour the losses in the turbine need to be defined. Losses cause increases in entropy, decreased availability and reduced work.

Losses include;

Leakage	shaft-gland packing (around stage) labyrinth seals joints balance pistons blade tip
Rotational	disc friction or windage (skin friction) fanning action of idle blades (partial arc admission) bearing friction water seal power losses when used
Pressure	strainer governing valve throttle losses pipe work exhaust or hood loss in condenser
Heat	radiation and conduction from case (usually negligible but appreciable on very small turbines with higher surface to volume ratios)
Pumping	oil pumps circulation pumps
Moisture	liquid droplets striking blade and lost flow

Some losses decrease with load such as fanning of idle blades in partial arc admission. Some loads remain constant such as bearing, and pumping losses. Other loads increase with load such as leakage and disc friction. Exhaust losses and generator losses are proportional to the square root of the load.

Control at part load can be by closing the governing valve and throttling the steam. Pure throttling is unsatisfactory as the losses are high thus steam is often supplied from carefully designed groups of ports to reduce pressure loss. Partial arc admission is often used and is where steam is supplied to a reducing number of groups of inlet nozzles to remove most of the throttling losses. Sliding pressure is where the boiler controls the inlet pressure, reducing it at part load, and ideally the inlet valves remain wide open.

At part load the exhaust wetness increases. This accounts for a large percentage of the part load performance reduction. A simple rule of thumb for moisture loss is 1% reduction in stage efficiency for each additional 1% moisture. One commentator puts the figure at 1.15% per 1% moisture [160].

Several different efficiencies have been defined. Nozzle efficiency is the efficiency of the nozzle in converting pressure and heat to kinetic energy;

$$\eta_N = KE_{\text{ACTUAL}} / KE_{\text{ISENTROPIC}} \quad \dots 6.18)$$

This is linked to the nozzle coefficient K_N , $V_{\text{ACTUAL}} = K_N V_{\text{ISENTROPIC}}$

Blade efficiency is the blade work output to kinetic energy input from

the nozzle;

$$\eta_b = \text{work out of blade} / \text{energy input to blade} \quad \dots 6.19)$$

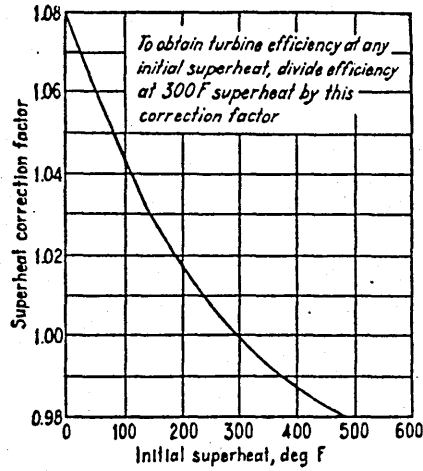
The combined nozzle and blading efficiency is η_{NB} , and if losses due to drum rotation, tip friction, and steam leakage are neglected then this is the stage efficiency. This efficiency is about 0.65 for small turbines and rises to 0.9 in large ones. Stage efficiency is proportional to the square root of pressure and so is worse in high pressure stages. Many other efficiencies are defined such as the carry over efficiency η_{co} which can be found in the texts such as [157], [158], [159] and [160]. It is beyond this thesis to list them all and would not assist in finding a model for part load performance.

Part load effects on the efficiencies and losses are not well documented in the literature. Effects on individual efficiencies are documented such as losses due to wetness, proportionality powers and design point effects. But the over all part load effect is too complex to produce satisfactory results purely from theory using a simple mathematical model. Turbine performance can only be satisfactorily determined by tests on full scale machines. Even with a large numerical model dealing with the turbine stage by stage it would be insufficient without experiential backup, especially as the effects would vary between design types.

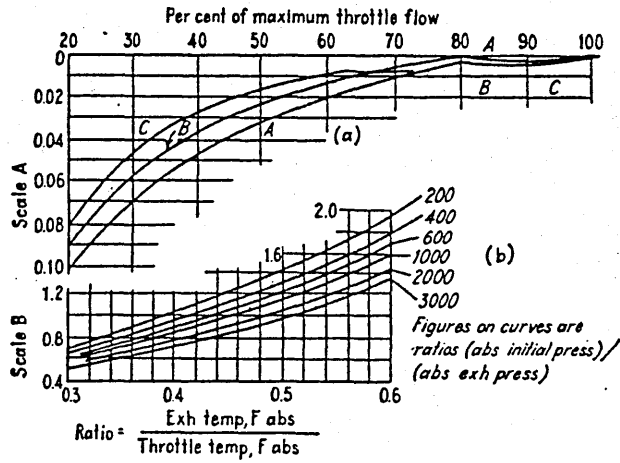
The reference [161] has fully usable and inclusive part load efficiencies, figure 6.11, but is dated at 1941. In spite of their age these part load efficiencies are used as they are in a usable form, based on real turbines, and available in the time available to the author. The results will have the correct trends but are presumably pessimistic, as improved modern blade profiles are less sensitive to angle of incidence and therefore will have better part load performance. The part load effects are used as part of the combined cycle section and are thus even more acceptable as they only represent a small part of the result, the main part being the off design behaviour of the heat recovery boiler and gas turbine.

Figure 6.11 Part Load Correction Factors

TURBINE PERFORMANCE



Throttle-superheat correction factors. (Warren and Knowlton.)



(c) Efficiency correction factor for part loads = 1.00 - (Scale A × Scale B)	Maximum throttle flow, cu ft per sec			
	Curve	3,600 rpm	1,800 rpm	Steam conditions
	A B C	50 or less 50-125 125 and up	150 or less 150-375 375 and up	All All Used only up to 825 F

Partial-load correction factors for condensing turbines.

7 COMBINED CYCLES

7.1 INTRODUCTION

Many advantages of the combined cycle compared to the steam cycle or gas turbine have been dealt with previously in section 2. The high efficiency of the combined cycle is due to the high temperature of heat utilised in the start of the gas turbine expansion and the low temperature of heat exhausted in the steam turbine.

The two cycles compliment each other for a simple reason. Due to the low expansion ratio and high TET of the gas turbine, the gas leaves the turbine still at high temperature. The gas is expanded back to ambient pressure but the gas turbine has no way of expanding below this pressure to extract more work. The steam turbine on the other hand has a moderate inlet temperature but a large expansion ratio. This is due to the large initial pressure and the condensing of the exhaust vapour to form a vacuum. Expansions can be down to pressures as low as 5 kPa absolute. The large expansion ratio means a large temperature drop and low exhaust temperature. The remaining task is to transfer the heat from the end of the gas turbine expansion with the start of the steam expansion, which is done by the Heat Recovery Steam Generator (HRSG).

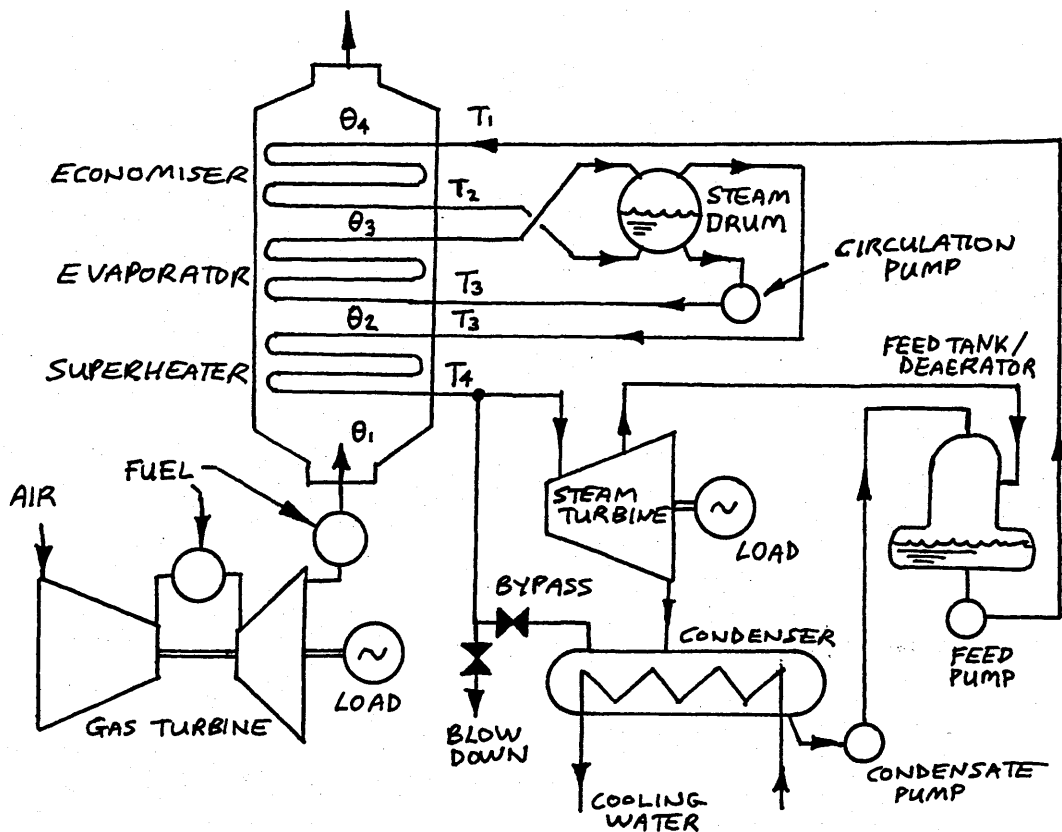
Typical current values for a combined cycle gas turbine might be a pressure ratio of 10-20 and TET of 1100-1500 K. The exhaust is usually about 700-870 K. The steam cycle would use a maximum TET of approximately 810 K and a pressure of say 8-10 MPa. The condenser pressure might be say 8 kPa absolute giving a pressure ratio of about 1000 and exhaust temperature of about 315 K. It will be shown that the typical gas turbine exhaust temperature is a little too low for an optimum steam cycle. Practical combined cycles were delayed until TET's exceeded something like 1350 K using the typical pressure ratios of 10-13 [2].

In this section combined cycles are presented, starting at market trends, and uses. These include 'topping', 'bottoming, and cogeneration. A discussion on configuration includes a table of the effect on availability and maintenance in multi-engine plant. The theory of combined cycles is looked at in an attempt to understand how they work and why they are designed as they are. A hypothetical simple cycle gas turbine and single pressure steam cycle is analysed in detail. The combined cycle performance depends largely on matching the cycles so the heat recovery steam generator is presented in detail. Discussion on supplementary firing, reheat, regeneration and two pressure boilers is included as means of improving efficiency. A method is given for estimating the combined cycle performance of a given gas turbine. The off design characteristics are investigated including engine configuration effects.

7.2 CYCLES

A simple combined cycle layout is given in figure 1.1. This does not include internal details of the boiler, control lines, or engine operation systems. A basic but more complete diagram is shown in figure 7.1. The gas turbine exhaust is ducted directly into a boiler, called the Heat Recovery Steam Generator (HRSG). The HRSG transfers the heat of the exhaust into the high pressure feed water and generates dry high

Figure 7.1 Single Pressure Combined Cycle



pressure superheated steam. The steam is passed through a steam turbine and into the vacuum created by the condenser. The condenser creates a vacuum by condensation of the water from a vapour to liquid in an enclosed space. The fluid is passed on as a liquid to the condensate pump. The deaerator reduces the oxygen and carbon dioxide content before the feed pump repressurizes the feed water.

The flow of energy is shown in figure 7.2. This example is for the hypothetical gas turbine in section 3.4 and a single pressure HRSG at 6 MPa. The air is ingested and energy added as it is compressed. The fuel is added and burnt giving the air an energy level 140% of the fuel energy at the turbine entry. The temperature at this stage is 1400 K and the gas is at 1200 kPa. The gas turbine extracts about half of the energy as work but about half of this work is required by the compressor. The remaining energy which is 66% of the fuel is exhausted as near ambient pressure combustion products at about 830 K. The HRSG transfers about 70% of the exhaust heat to the water and rejects the final low grade exhaust heat to the atmosphere at about 420 K. The steam cycle only sees about 45% of the initial fuel energy at about 800 K and converts 33% of this energy to work, rejecting the rest to the cooling water at 315 K (40 °C).

The thermodynamic processes can be separated for analysis, as in figure 7.3, into the gas turbine cycle, the heat recovery system, and the steam cycle. The heat source is the fuel (Q_{H1}), at the temperature of gas turbine inlet, and the heat sinks are firstly the atmosphere for the

Figure 7.2 Flow of Energy - Combined Cycle

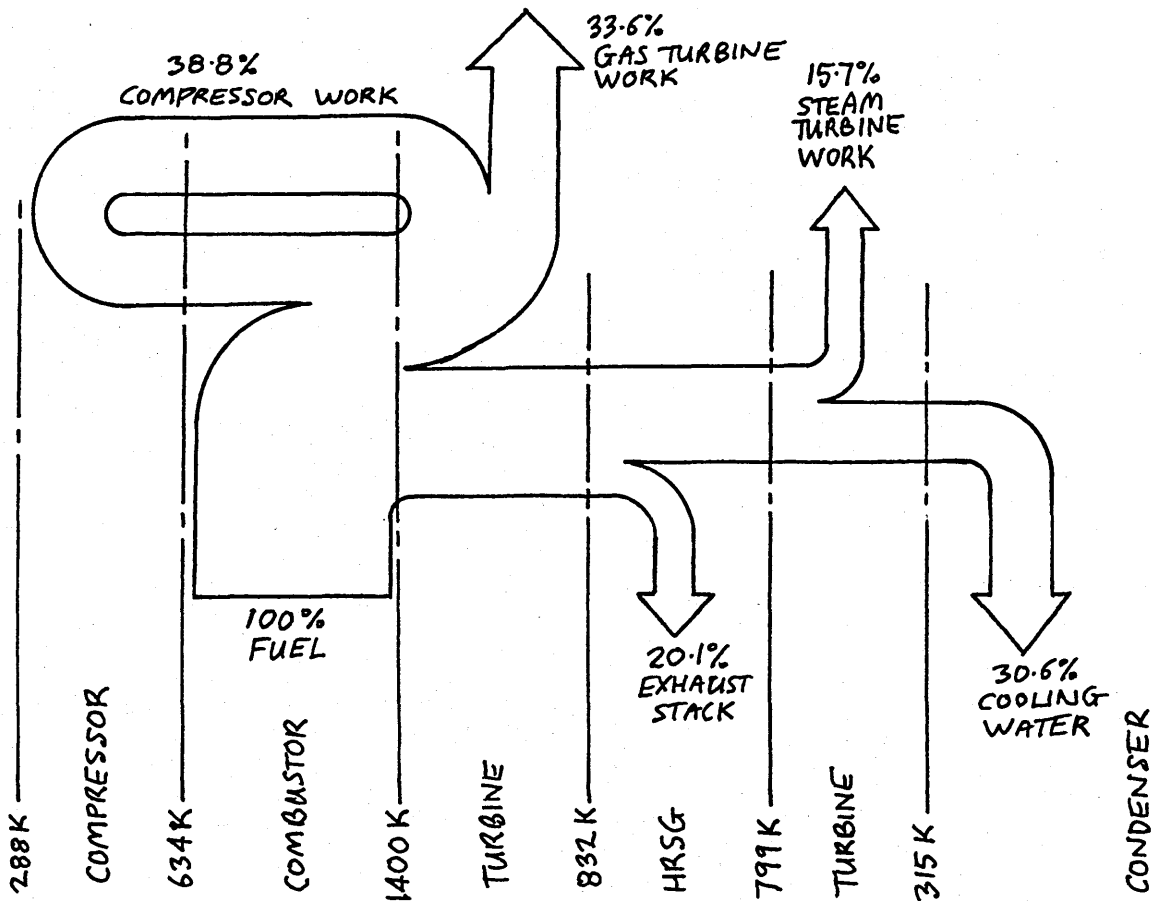
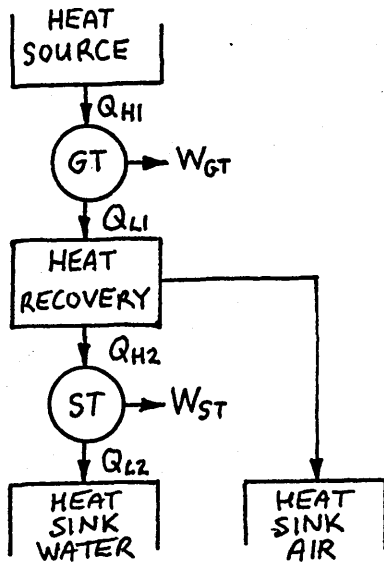


Figure 7.3 Efficiency and Energy Flows



$$\eta_{GT} = \frac{W_{GT}}{Q_{H1}}$$

$$\eta_{HR} = \frac{Q_{H2}}{Q_{L1}}$$

$$\eta_{ST} = \frac{W_{ST}}{Q_{H2}}$$

$$\eta_{CC} = \eta_{GT} + \eta_{HR}\eta_{ST}(1 - \eta_{GT})$$

losses in the heat recovery ($Q_{L1}-Q_{H2}$) and then the cooling water for the steam turbine cycle (Q_{L2}). The following efficiencies are defined;

$$\text{Gas Turbine Cycle } \eta_{GT} = W_{GT}/Q_{H1} \quad \dots 7.1)$$

$$\text{Heat Recovery } \eta_{HR} = Q_{H2}/Q_{L1} \quad \dots 7.2)$$

$$\text{Steam Cycle } \eta_{ST} = W_{ST}/Q_{H2} \quad \dots 7.3)$$

and noting that the combined cycle work is; $W_{CC} = W_{GT} + W_{ST} \dots 7.4)$

it is easily shown by combining the above equations that;

$$\text{Combined Cycle } \eta_{CC} = W_{CC}/Q_{H1} \quad \dots 7.5)$$

$$= \eta_{GT} + \eta_{HR}\eta_{ST}(1 - \eta_{GT}) \quad \dots 7.6)$$

This equation indicates that the maximum combined cycle efficiency is not necessarily at the maximum gas turbine efficiency. The bottoming cycle cannot make up for all the gas turbine inefficiencies, $(1-\eta_{GT})$, as the term $\eta_{HR}\eta_{ST}$ is less than unity. Varying gas turbine efficiency implies varying the exhaust temperature and as is shown later this affects η_{HR} or the $\eta_{HR}\eta_{ST}$ product. The optimisation for overall efficiency is not obvious from the equation as a high gas turbine efficiency may mean a low heat recovery efficiency. The performance section 7.6 attempts to show how the variables effect each other. Equation 7.6 does not apply when energy is added at points other than in the gas turbine, such as in supplementary firing.

The term bottoming cycle efficiency is defined for future use as the product;

$$\eta_{BC} = \eta_{HR}\eta_{ST} \quad \dots 7.7)$$

7.3 APPLICATIONS

7.3.1 Market trends

Combined cycle power plant are projected to form a significant portion of electricity capacity installed over the next decade [8]. Combined cycles offer flexibility in staged construction so suit many growth patterns. A common growth pattern is installation of a simple cycle gas turbine with provision for future conversion to combined cycle.

The current combined cycle market has clear trends in applications [179]:

Base load power generation: This is popular in the 300-400 MW size range using typically two gas turbines with separate unfired dual pressure HRSG's, and a single steam turbine. The plant may be developed in stages to ease procurement costs and the fuel is mostly natural gas or residuals.

Upgrading existing plant: This is adding a topping or bottoming cycle. In the middle east where simple cycle gas turbines are common, there is interest in adding steam turbines or cogeneration such as water desalination. In the western world there is a strong interest in topping or repowering of steam boilers with a gas turbine, or to convert

existing construction programs from nuclear or coal to natural gas combined cycles.

District heating: Usually a combined cycle with steam extraction for the district heating.

Total energy: Includes cogeneration schemes on industrial sites, for instance in the United States typically 200-500 MW selling the electricity to the electricity utility and the steam to one or more industrial sites.

Some further application information and existing combined cycle plant is available in references [179], [219], [213], [212], [223] and [197].

7.3.2 Topping and Bottoming

The terms topping and bottoming are used for addition of power cycles to other plant. The top cycle is the cycle burning the fuel and using the higher temperature heat input. In the combined cycle this is the gas turbine. The bottoming cycle is the cycle utilising the rejected heat of the top cycle. Generally all commercial bottoming plant have used steam due to it's advantages, low cost, established technology, stability, high specific heat and high heat transfer rates. Steams disadvantages are it's low molecular weight, high latent heat and high critical pressure. Other bottoming cycle fluids have been suggested but have not succeeded commercially.

Conversion of an Existing Gas Turbine

'Bottoming' is where rejected heat from a high temperature process is used for further processes such as power generation. Conversion of an existing simple cycle gas turbine to a combined cycle increases the power output by about 50% for no increase in fuel consumption. The reference [212] (Alsthom VEGA) compares the generating costs for various configurations. Using a reference of a 65 MW conventional steam plant operated at base-load (6000 hr/yr), the generating cost of a simple cycle gas turbine is +5%, a combined cycle -25% and the incremental running cost of conversion of an existing simple gas turbine cycle to combined cycle is -80%.

The benefits of conversion of an existing gas turbine to combined cycle are said to be attractive if the gas turbine is used for more than about 2000-3000 hr/yr [212]. Although the operating costs are advantageous the capital cost is high, and about double the cost per kW of a new combined cycle. Also the life expectancy of the gas turbines must be assessed before embarking on such new capital outlay. Old gas turbines with low TET and exhaust gas temperature may have improvements in efficiency with supplementary firing but may still be less profitable than new plant [12]. However the pay back period for simple cycle conversion can be satisfactory if the plant is operated for reasonable hours. To minimise pay-back the added bottoming cycle is often of the simplest configuration, single low pressure, and without deaerator/feedheater. As many existing gas turbines are relatively small the conversion to a combined cycle may mean a high speed steam turbine geared to the generator. Layout problems nearly always exist but can usually be overcome.

Conversion of Existing Steam Turbine

'Topping' is where the prime plant is a steam turbine or industrial process and a gas turbine is used to 'top up' the heat input. The gas turbine exhaust is added to the boiler flow and replaces some of the boilers fired fuel. Repowering is where a topping gas turbine is used to replace most or all of the fuel fired in the boiler of an existing plant. The range of gas turbine input can be from use as a forced draft fan to supplying the full heat input into the steam cycle. The HRSG in a combined cycle is only a fraction of the size of a fired steam boiler, say 1/8th in a 400 MW plant, but due to the large rating of steam plant relative to available gas turbine sizes a number of gas turbines may be needed for full repowering.

The main problems encountered with repowering are that the steam turbine will be existing and may or may not be optimised for the relatively low steam conditions required by the combined cycle. Conventional steam plant tend to have the highest practical pressure for the temperature and number of reheats used, whereas a packaged combined cycle has an optimised and lower steam pressure for maximum heat recovery efficiency. Older steam plant may well have aging boilers but good condition steam turbo-sets, and many older steam plant are designed for the low steam pressure and temperature as used in modern combined cycles. The low steam condition and hence lower efficiency may not justify replacement of deteriorating boilers [12], but may correspond well to the requirements of a combined cycle. As a rule of thumb the space released by removal of the fired boilers is sufficient for the gas turbines and the HRSG's [12]. This means no extra land is required and also planning approval should be easier.

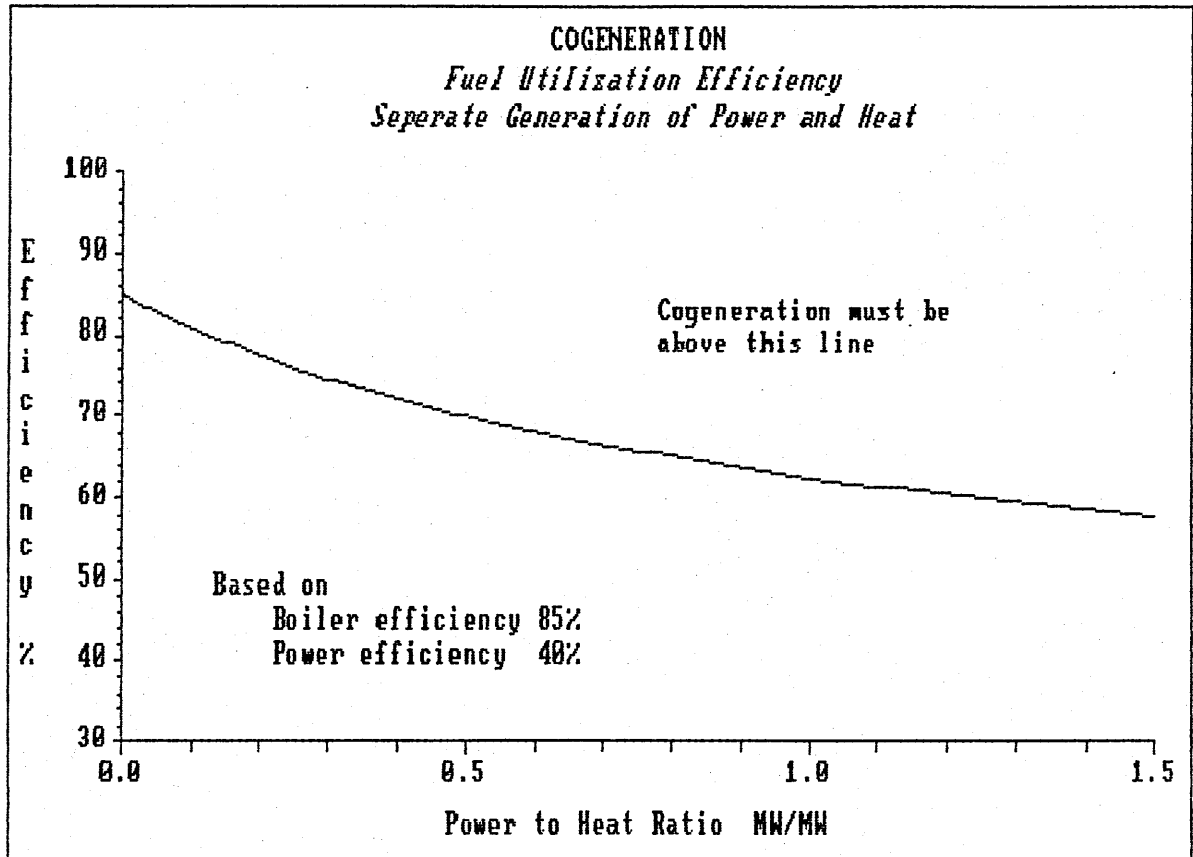
Other problems in repowering are that the existing steam plant will have regenerative feedheaters, which reduce the low temperature heat transfer in the heat recovery boiler, and will most likely have reheat, which increases the heat transfer at high temperatures. This is satisfactory in a conventional fired boiler but reduces the heat recovery efficiency in a combined cycle. Existing steam plant may also be of a large rating, say up to 1000 MW, and full repowering could mean up to three times the original power capacity. This large extra output may not be needed, even if it is at a higher efficiency. The fuel of the repowered plant will need to be burnt in the gas turbine instead of a boiler thus repowered coal plant will need to change fuel and reduce in fuel flexibility.

Repowering of utility steam plant has been discussed for many years but there are relatively few examples except notably in the Netherlands.

7.3.3 Cogeneration

Combined cycles have applications in more than just power generation. Due to the generation of steam, the combined cycle can also be integrated in other steam or heat using processes in what is called cogeneration. The combined production of power and heat is called Combined Heat and Power or CHP. CHP can provide for the maximum utilization of fuel, therefore minimising consumption of resources. The combined cycle on it's own has one of the highest thermal efficiencies of any power plant and may achieve efficiency of more than 50% when new and clean. CHP can achieve fuel utilization of up to 85% at design point, which is similar to a boiler efficiency but involves power

Figure 7.5 Fuel Utilisation for Separate Production of Heat and Power



production as well as heat so is more valuable. However average operating efficiency may be somewhat less than design point due to load profiles. Historically Europeans developed cogeneration applications before the Americans and it is a recognised and well developed technology. Uses include industrial steam, district heating, desalination, drying, oil refineries and glass house heating/CO₂ enrichment.

In industrial CHP applications the power requirement is usually electricity and the heat is in the form of steam. The electricity can inevitably be purchased from an electricity utility but the steam must be generated. Here justification for the CHP system would concentrate on the cost of the electricity generated as additional cost to that cost as if the steam were generated in a conventional boiler. In this case the efficiency of the electricity produced, η_E , can be defined as follows;

$$\eta_E = \frac{W_E}{Q_F - Q_P/\eta_B} \quad \dots 7.8)$$

where Q_F and Q_P are the fuel and process heats respectively, W_E is the electrical power output and η_B is the efficiency of a separate heat production boiler (typically 0.85).

Clearly industrial CHP is only of interest if it saves significantly over separate generation of electricity and heat, especially as it is less flexible and more capital intensive. Conventional direct fired

boilers produce steam from fuel at an efficiency of about 85%, and assuming separate power generation is at 40% efficiency a simple plot of fuel utilization for separate power/steam production can be made, figure 7.5. The utilization of fuel in the cogeneration plant must lie well above this line to be of interest, suggesting that CHP is only viable with the higher power to heat ratios.

To encourage industrial CHP any surplus power should be able to be exported back to the national grid. This is so that the highest power to heat ratio can be selected, and so that heat requirements can be matched leaving the mismatch of power to be satisfied by selling or buying power. Purchase of this power can only introduce complication for the electricity utility. It must normally have the connected capacity for the full load yet supply less, and as a reasonable price will be required by the industry some government incentive must be added to encourage utilities. Commonly the price paid by the utility is the cost avoided or fuel cost of the electricity that would have been purchased, which is too small to pay capital on the CHP but may allow export of power at low demand just to keep the plant running.

Steam injection cycles for power augmentation are well suited to CHP situations where steam demand fluctuates. The rate of steam injection can be varied from zero to full steam production to absorb any the steam surplus to process requirements. Supplementary firing can improve flexibility even further. The main problems are that the largest power production is at the lowest steam demand, which is likely to also be the lowest power demand, and the injected steam is lost requiring large quantities of treated make up water. The steam injection cycle is discussed more fully in section 7.12.

Efficiencies of centralised industrial steam production can be deceiving. After the steam is produced it must be transmitted around the factory with radiation, leakage and other losses. Total efficiency for heat from fuel to process location may be much less than the boiler efficiency of 85%. Measurements [225] have shown that typical steam production efficiencies, including steam distribution losses and boiler efficiency, are more like 60% at high load (working shifts) and 25% at low load (unproductive shifts). The average efficiency for three plant were 34% (one shift per day), 40% (two shifts) and 55% (continuous). CHP is only suitable for this traditional concept of centralised steam production but if moves are made towards decentralised use of gas appliances for direct heating and steam production [226] then CHP suffers a serious handicap on efficiency and the high cost of steam piping.

District heating is where cogeneration heat is piped to consumers for domestic heating. It is well accepted in Scandinavia and used in several other European countries, notably the Netherlands and Germany. It is difficult to justify district heating on purely business grounds as the initial investment is very high and returns are over a long period. Pay-back periods may be in excess of 20 years. For this sort of investment to occur it must be encouraged by central government. The district heating plant may typically be a combined cycle with steam extraction for the hot water heating circuit. The distributed water is typically at 100 to 105 °C and returned at about 50 to 70 °C. The plant may run at combined cycle efficiencies producing electricity during the warmer months and at some higher CHP efficiency during the colder

months. References are [253] and [219].

The CHP system design would vary depending on the ratio of power to heat required, what part load requirements are expected, geography, fuel price, fuel availability, required return on investment, existing plant, surplus power sale agreements and taxation requirements. The control is also changed if the power generation has to take part in grid frequency control or is free to change load. Owing to the very wide ranging design requirements, CHP design is not dealt with in detail here except to say power to heat ratios are possible in the ranges of table 7.1.

Table 7.1 Power to Heat Ratios for Common Power Plant

Power Plant	Power over Heat Ratio
Combined cycle	0.7 - 0.8
Gas turbine and HRSG	0.35 - 0.5
Back pressure steam turbine	0.25 - 0.3
Extraction condensing turbine	0.3 - 1.0
Steam injected gas turbine	0.35 - ∞

The combined cycle could extend it's power to heat ratio range if equipped with an extraction condensing steam turbine. The power to heat ratio can also be varied if supplementary firing is used. The pressure range of steam is usually in the range 0.5-40 bar. Greater than 40 bar is normally not economical in cogeneration [223].

7.4 CONFIGURATION

Combined cycles come in various configurations. The configurations looked at here will be the conventional combined cycle using a liquid or gas fired gas turbine exhausting to a boiler. The configurations of integrated coal burning are looked at in there own section.

7.4.1 Single Shaft and Multi-engines

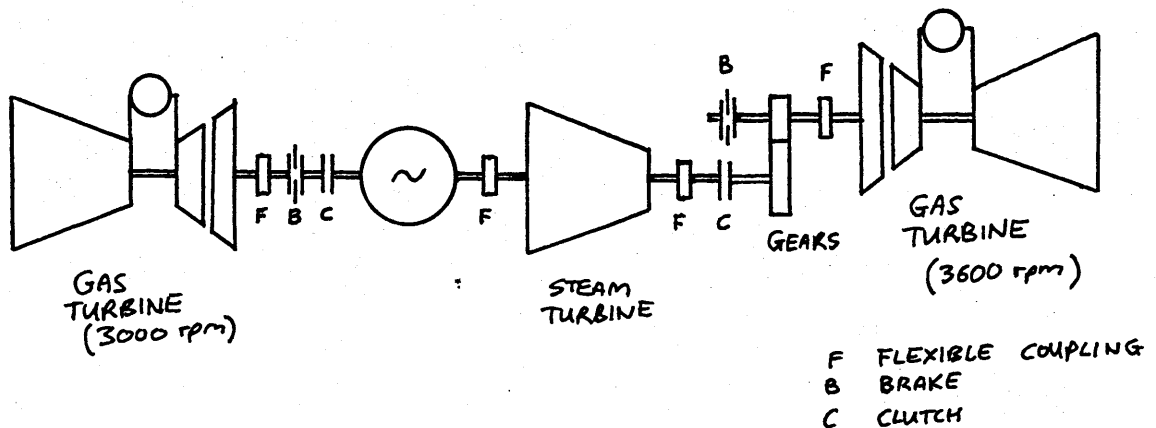
A single shaft combined cycle has the gas turbine and the steam turbine on the same shaft, usually on either side of a double ended generator. The single shaft combined cycle has advantages in load following as the steam turbine does not need speed or load control. It also has higher efficiency due to only one large generator and the single generator is a lower cost than several smaller ones. But a disadvantage of a single shaft is that the cycles cannot easily run separately when one is out of service, requiring clutch and brake pairs to do so. Also the plant is more difficult to start, requiring a large starter power. For a single shaft configuration a small steam turbine size might require gearing to run at the required generator speed. Reference [248] puts the minimum sized 3000/3600 rpm steam turbine at approximately 35 MW, but smaller turbines at this speed have been used such as in reference [253].

A multi-shaft combined cycle has the steam turbine physically separate from the gas turbine. This leaves the option of several gas turbines feeding one steam turbine, which allows a single larger steam turbine with efficiency improvements of scale. For plant with multiple engines a common format of notation has developed. For example, a 2 gas turbine and 1 steam turbine plant is called a 2+1 block configuration. In these multi-engine blocks the single steam turbine not only has higher

efficiency due to scale effects but may be a lower cost than several smaller ones, especially if generator numbers are also reduced.

The use of multiple gas turbines and one or more steam turbines is a flexible option for operational reasons. One gas turbine can trip off line and leave the others running, and at part load the gas turbines can be shut down one by one. For instance a two gas turbine set may shut down one gas turbine at 50% load and regain near design point efficiency. The installation of separate exhaust stack and diverter damper will allow the gas turbine to be operated in simple cycle mode while the steam turbine or HRSG are off line. This adds flexibility, and historically increases availability by 0.5-2.0%, although at the cost of reduced efficiency while in operation as a simple cycle.

Figure 7.6 Example 2 + 1 Block for LM2500



Usually each gas turbine has its own HRSG. This is for operational reasons, for instance if several gas turbines are linked to one HRSG and one is stopped or trips off line the hot exhaust gases of the remaining gas turbines may blow backwards through the stationary gas turbine causing heat damage. Thus if there is only one HRSG the gas turbines require automatic shut off doors with associated fast controls and reliability problems. Also if there is only one HRSG and it is off line for cleaning or maintenance it would put the whole steam turbine capacity off line. Generally the steam lines are connected and any gas turbine and HRSG pair can feed the steam turbine(s).

Multi-shaft installations allow staged modular construction. The steam turbine calls for heavier investment than the gas turbine and yet is only 1/3 of the output. Gas turbines are usually standardised and some gas turbine manufacturers carry turbines as stock items, allowing unusually short completion times [12]. The gas turbine has a low efficiency on its own but is cheap and quickly installed starting cash

flow, and allowing future conversion to combined cycle and full efficiency as load grows and money is available. The steam turbo-sets are usually manufactured to order and combined cycle completion times are correspondingly longer than gas turbines.

An example of an innovative 2+1 combined cycle block combining the efficiency and cost gain of a single generator but retaining the flexibility of independent gas turbine operation is given in figure 7.6 and reference [253]. In this installation the single large generator has an efficiency of 98.3% as opposed to three smaller ones which would have 97.6%. The single generator is also very much cheaper, and only requires one protection and control unit, and one set of switch gear and transformer units. The efficiency loss of the one GT operating through a reverse rotation gear is offset by the increased performance due to it's higher allowable speed. The flexibility and availability of separate multi-shaft units is largely retained by the use of clutch and brake units between the gas turbine and the steam turbine and generator set. The set can be started and operated with only one gas turbine.

7.4.2 Multienzyme Operational Consequence

The use of multiple engine installations has effects on maintenance, availability and the consequences when a component is off line. Table 7.2 presents some of the implications for various configurations. Assumptions are as follows;

- | | | |
|-----------------|---------------|-----|
| 1) Availability | Gas turbine | 90% |
| | HRSG | 98% |
| | Steam turbine | 99% |
- 2) Part load efficiency of steam turbine at;
- | | | |
|--|---------------|-----------------------|
| | 3/4 mass flow | 0.995 of design point |
| | 2/3 mass flow | 0.990 of design point |
| | 1/2 mass flow | 0.975 of design point |

Note; These figures are from reference [161] and would vary from turbine to turbine with different design requirements.

- 3) HRSG steam flow output is in proportion to GT exhaust mass flow.
- 4) Gas turbines have exhaust stacks and diverter dampers for single cycle operation while the HRSG is off line.
- 5) The maximum power available is used, so gas turbines are operated as simple cycles if necessary.
- 6) The gas turbine represents exactly 2/3 of the design point output.
- 7) Loss of one component does not increase the probability of another component failing.
- 8) All turbines, including the steam turbines, cannot be operated at higher output than the design point value.

The availability equation used is;

$$P = \frac{N!}{K! (N-K)!} (1-A)^K A^{N-K} \quad \dots 7.9$$

Where P = Fraction of time K engines are shut down
 N = Total number of engines
 K = Number of engines shut down
 A = A Single Engine Availability

Table 7.2

Component Number Effect on Availability and Performance

MULTIPLE ENGINE BLOCK CONFIGURATIONS										Based on Gas Turbines 2/3 and Steam Turbines 1/3 of total power output												
CONFIGURATION		GAS TURBINE			HRSG			(avail 98%)			STEAM TURBINE			(avail 99%)			OVERALL					
Number of:		Probability of:			If one GT down:			Probability of:			If one HRSG down:			Probability of:			If one ST down:			Availability allowing:		
GT	HRSG	ST	Zero down	One down	Power output	Heat rate increase	Zero down	One down	Power output	Heat rate increase	Zero down	One down	Power output	Heat rate increase	Zero down	One down	Power output	Heat rate increase	Full Load only	Up one down	Up to one component down	
1	1	1	0.900	0.100	0.000	N/A	0.980	0.020	0.667	1.500	0.990	0.010	0.667	1.500	0.990	0.010	0.667	1.500	0.873	N/A	0.997	
2	1	1	0.810	0.180	0.496	1.008	0.980	0.020	0.667	1.500	0.990	0.010	0.667	1.500	0.990	0.010	0.667	1.500	0.786	0.990	0.984	
2	2	1	0.810	0.180	0.496	1.008	0.960	0.039	0.829	1.206	0.990	0.010	0.833	1.200	0.990	0.010	0.833	1.200	0.770	0.990	0.981	
2	2	2	0.810	0.180	0.500	1.000	0.960	0.039	0.833	1.200	0.980	0.020	0.833	1.200	0.980	0.020	0.833	1.200	0.762	0.990	0.978	
3	1	1	0.729	0.243	0.664	1.003	0.980	0.020	0.667	1.500	0.990	0.010	0.667	1.500	0.990	0.010	0.667	1.500	0.707	0.972	0.965	
3	3	1	0.729	0.243	0.664	1.003	0.941	0.058	0.888	1.126	0.990	0.010	0.888	1.126	0.990	0.010	0.888	1.126	0.679	0.972	0.954	
3	3	2	0.729	0.243	0.664	1.003	0.941	0.058	0.889	1.125	0.980	0.020	0.889	1.125	0.980	0.020	0.889	1.125	0.672	0.972	0.951	
3	3	3	0.729	0.243	0.667	1.000	0.941	0.058	0.889	1.125	0.970	0.029	0.889	1.125	0.970	0.029	0.889	1.125	0.666	0.972	0.949	
4	1	1	0.656	0.292	0.749	1.002	0.980	0.020	0.667	1.500	0.990	0.010	0.667	1.500	0.990	0.010	0.667	1.500	0.637	0.948	0.939	
4	4	1	0.656	0.292	0.749	1.002	0.922	0.075	0.916	1.091	0.990	0.010	0.916	1.091	0.990	0.010	0.916	1.091	0.599	0.948	0.920	
4	4	2	0.656	0.292	0.749	1.002	0.922	0.075	0.917	1.091	0.980	0.020	0.917	1.091	0.980	0.020	0.917	1.091	0.593	0.948	0.917	
4	4	4	0.656	0.292	0.750	1.000	0.922	0.075	0.917	1.091	0.961	0.039	0.917	1.091	0.961	0.039	0.917	1.091	0.581	0.948	0.911	
5	5	1	0.590	0.328	0.800	1.000	0.904	0.092	0.933	1.071	0.990	0.010	0.933	1.071	0.990	0.010	0.933	1.071	0.528	0.919	0.881	
5	5	2	0.590	0.328	0.800	1.000	0.904	0.092	0.933	1.071	0.980	0.020	0.933	1.071	0.980	0.020	0.933	1.071	0.523	0.919	0.878	
5	5	5	0.590	0.328	0.800	1.000	0.904	0.092	0.933	1.071	0.951	0.048	0.933	1.071	0.951	0.048	0.933	1.071	0.508	0.919	0.867	
6	6	1	0.531	0.354	0.833	1.000	0.886	0.108	0.944	1.059	0.990	0.010	0.944	1.059	0.990	0.010	0.944	1.059	0.466	0.886	0.839	
6	6	2	0.531	0.354	0.833	1.000	0.886	0.108	0.944	1.059	0.980	0.020	0.944	1.059	0.980	0.020	0.944	1.059	0.461	0.886	0.835	
6	6	6	0.531	0.354	0.833	1.000	0.886	0.108	0.944	1.059	0.941	0.057	0.941	1.059	0.941	0.057	0.941	1.059	0.443	0.886	0.820	
7	7	1	0.478	0.372	0.857	1.000	0.868	0.124	0.952	1.050	0.990	0.010	0.952	1.050	0.990	0.010	0.952	1.050	0.411	0.850	0.794	
7	7	7	0.478	0.372	0.857	1.000	0.868	0.124	0.952	1.050	0.932	0.066	0.952	1.050	0.932	0.066	0.952	1.050	0.387	0.850	0.771	
8	8	1	0.430	0.383	0.875	1.000	0.851	0.139	0.958	1.043	0.990	0.010	0.958	1.043	0.990	0.010	0.958	1.043	0.363	0.813	0.748	
8	8	8	0.430	0.383	0.875	1.000	0.851	0.139	0.958	1.043	0.923	0.075	0.958	1.043	0.923	0.075	0.958	1.043	0.338	0.813	0.721	
9	9	1	0.387	0.387	0.889	1.000	0.834	0.153	0.963	1.038	0.990	0.010	0.963	1.038	0.990	0.010	0.963	1.038	0.320	0.775	0.702	
9	9	9	0.387	0.387	0.889	1.000	0.834	0.153	0.963	1.038	0.914	0.083	0.963	1.038	0.914	0.083	0.963	1.038	0.295	0.775	0.671	
10	10	1	0.349	0.387	0.900	1.000	0.817	0.167	0.967	1.034	0.990	0.010	0.967	1.034	0.990	0.010	0.967	1.034	0.282	0.736	0.656	
10	10	10	0.349	0.387	0.900	1.000	0.817	0.167	0.967	1.034	0.904	0.091	0.967	1.034	0.904	0.091	0.967	1.034	0.258	0.736	0.623	

Explanation of table 7.2

In table 7.2 the values are fractions relative to the total time, design point power or design point heatrate. The relative heatrate values reflect the increase in fuel costs per unit output for the particular event. The table has the gas turbines, HRSG's and steam turbines listed separately and then the overall effect. Under each there are four columns. The first two are the probability of all the particular component type being available, and the probability of one being unavailable. The next two columns show the effect on power and heat rate relative to design point if one of the components are off line. The far right three columns are the availability of 'all the components being operational', of 'all or all less one gas turbine operational', and of 'all or all less any one component operational', respectively.

Example using table 7.2

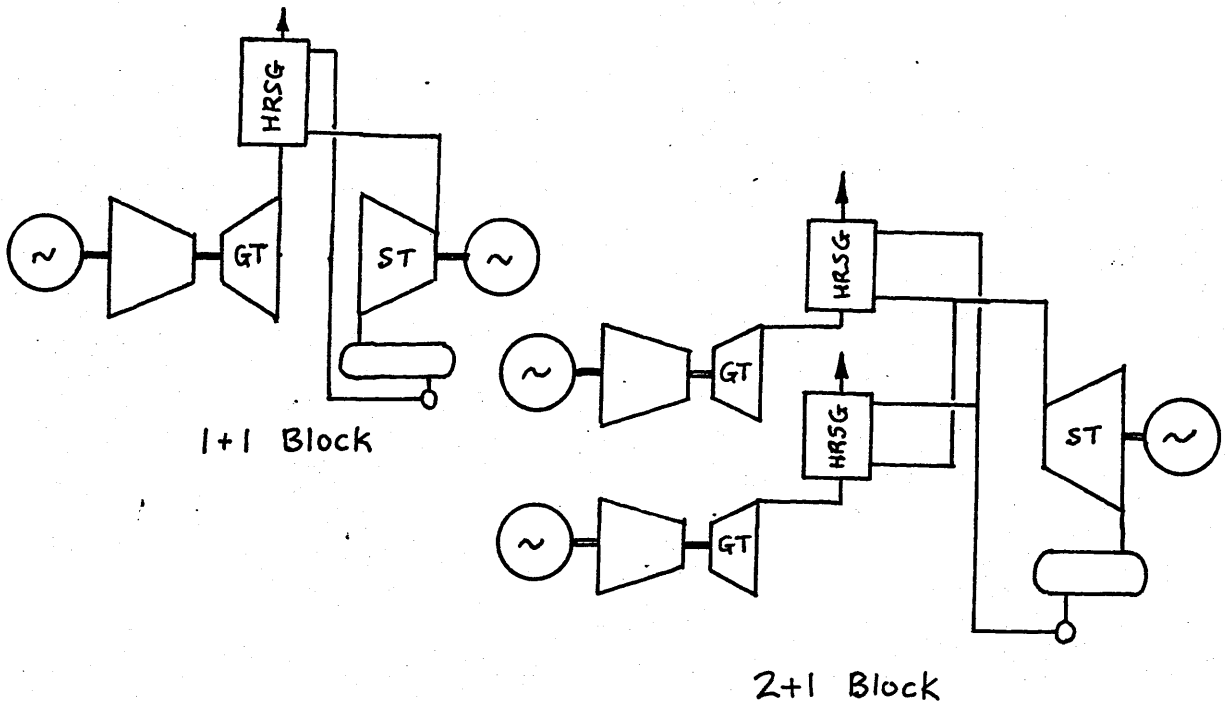
Two of the most popular configurations are the 1+1 and the 2+1 block with 2 HRSG's. These two are shown in figure 7.7 and are the 1,1,1 and 2,2,1 block configurations in table 7.2. From the table it can be seen that for a 1,1,1 block the gas turbine is operational for 90% of the time but has zero power output when off line. For a 2,2,1 block both gas turbines are available together for only 81% of the time and one only is unavailable for 18%. The power is reduced to just under 50% for 18% of the time, and of all the power is lost for just 1% of the time (100%-81%-18%). In the 2,2,1 block when one gas turbine is off line the fuel cost per unit output increases by just under 1%. This is due to the steam turbine being at part load, but the value of the loss will vary from one turbine to another.

The loss of a HRSG has a greater effect on the 1,1,1 block than on the 2,2,1. On the 1,1,1 block the HRSG is available for 98% of the time whereas on the 2,2,1 both the HRSG's are available together for only 96% of the time. The consequences of the HRSG off line on the 1,1,1 block is only 2/3 of the power and 50% greater fuel cost per unit of power. On the 2,2,1 block the loss of a HRSG with it's gas turbine on simple cycle mode gives 83% power with a fuel cost increase per unit of 20%. Obviously the 2,2,1 block could shut down the gas turbine as well for a HRSG off line and have little penalty on heat rate, but at a loss of 50% of the power.

For unavailability the percentage heat rate increase and therefore the fuel cost increase is not the only cost increase of the electricity. The electricity cost includes capital and maintenance costs, and the increase due to an outage includes the cost of additional power required from other stations. From table 7.2 the overall availability of all the components together for the 1,1,1 and 2,2,1 blocks are 87% and 77% respectively. But the 2,2,1 block would require twice as many gas turbine and HRSG parts, thus it's capital and maintenance costs would be higher.

In reality the availability of components are not independent of each other, as assumed in this analysis. For instance the HRSG might be cleaned at the same time as the maintenance of the GT, increasing availability. Also the efficiency of multi-engine installations will be less than large single engine installations as the sizes go down.

Figure 7.7 Various Block Configurations



7.5 HEAT RECOVERY

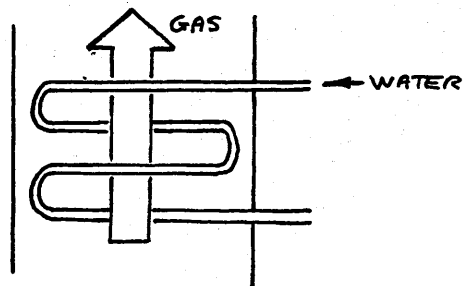
7.5.1 Heat Recovery Description

Both the heat recovery and steam cycle efficiency is governed by the gas turbine exhaust. The following section deals with the heat recovery efficiency. Analysis is initially restricted to a basic single pressure HRSG. Dual pressure HRSG's and other refinements are discussed later in section 7.7.

The process of heating the feed water at say 315 K and 6 MPa to the superheated steam conditions of say 800 K involves a phase change. This process is shown as the process 2 to 3 in the T-S diagram on figure 6.1. The water has heat added and increases temperature until it reaches boiling point, 2-2'. The boiling point is elevated by the high pressure. The temperature remains constant as more heat is added, 2'-2'', as the water boils and then the temperature continues to rise after the last of the liquid has changed to vapour, 2''-3. Optimisation of a HRSG differs from that of a direct fired boiler, as the heat supplied is of lower temperature. The decreasing temperature of the exhaust gas as heat is removed requires careful matching to the steam temperatures.

The HRSG is just an enlarged section of the exhaust duct, either vertical or horizontal, containing the heat exchange surfaces. The water is pumped through the pipe heat exchange surfaces, which usually comprise a bank of serpentine coils. This arrangement is

Figure 7.8 Serpentine Coil Arrangement



shown diagrammatically in figure 7.8. The coils are finned on the outside, and usually are connected in parallel between inlet and outlet manifolds. The arrangement means that although the individual flow is cross-flow to the exhaust gas, the bulk flow is counterflow to it.

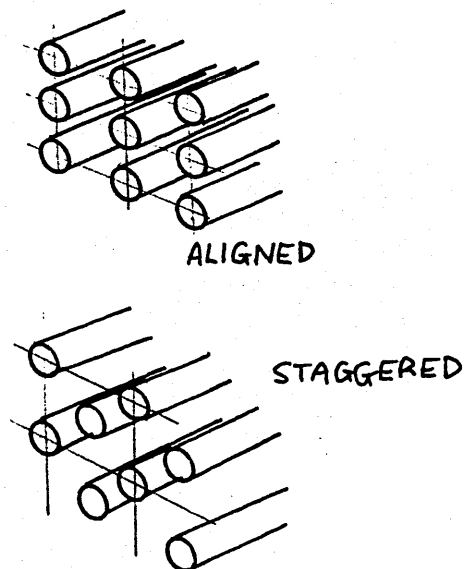
In construction of real boilers the processes of water heating, boiling and then superheating are usually kept separate. The economiser heats the water, and this process is called feed heating. Evaporation is done in an evaporator and the superheating in a superheater. The layout of these components is shown in figure 7.1. Flow from the economiser is invariably injected into a storage pressure vessel called the steam drum. To prevent evaporation occurring in the economiser there is usually a small temperature difference called the approach point between the economiser discharge temperature and the boiling temperature. This means a small amount of feed heating occurs in the evaporator. The evaporator may require a forced circulation pump as shown in figure 7.1, or it may use natural circulation in the bi-drum arrangement of conventional boilers. In the forced flow design the evaporator flow might be 3 to 5 times the feed flow. Circulation pumps introduce a further component for reliability, and the possibility of pump failure must be designed into the evaporator. The heat exchange characteristics of the forced or natural circulation arrangements are basically the same. The superheater takes its flow from the space above the water level in the steam drum.

The design of the HRSG can take various forms, tube sizes and fin configurations, and optimisation studies may be used for each design. The boilers can be of single or two pass design, figure 7.10, depending on space considerations. Fire tube arrangements have been used with the water on the outside of the tubes and the exhaust on the inside. This is usually used when integrated with processes of very high pressure gases, and in combined cycles the water tube arrangement is inevitably used.

The fins are used to increase the effective area of the tubes in contact with the exhaust. Typical fin pitch varies for gas turbine heat recovery from about 4 to 8 mm. Clean fuels such as natural gases allow closer fin packing. The fin thickness may be 1 to 3 mm and the height 12 to 20 mm. Fin temperatures must be controlled, so with high temperatures, such as with supplementary firing, the first rows of tubes may be bare with initially low density on subsequent rows. Later rows would have the full density of fins. Serrated as well as plain fins are used.

The tube bank can be staggered or in-line, figure 7.9. The staggered bank has a better heat transfer characteristic, but a higher gas pressure drop. As the efficiency of the gas turbine was found in section 3.4.5 to drop by 0.5% per kPa of exhaust loss the layout needs to be considered in the design optimisation. Typical pressure losses in a HRSG are 1.5 to 2.0 kPa. In line tubes are usually selected for poor fuels as they are easier to clean. Generally a vertical evaporator

Figure 7.9 Aligned and Staggered

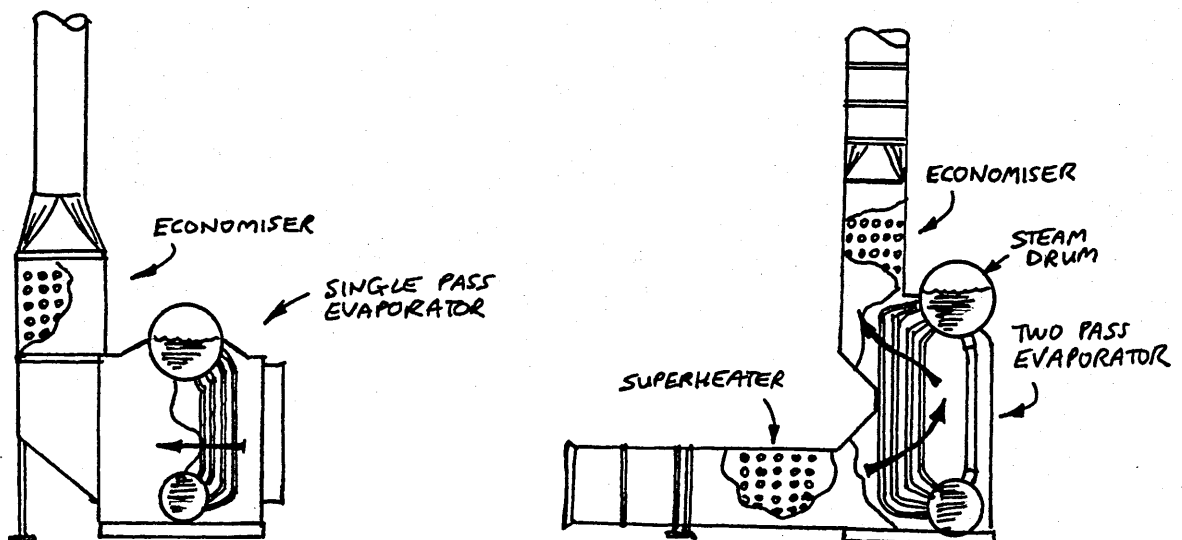


tube can withstand a higher heat flux than horizontal tubes due to the nature of vapour formation, therefore evaporator tubes are mounted vertically. The duct work is insulated and in supplementary fired boilers the higher temperatures mean a more expensive insulation, such as ceramic fibres with a stainless steel liner.

At part load boiling may occur in the economiser. This is especially a problem in supplementary fired boilers operated in an unfired mode. If this is likely to cause a problem then provision can be made for bypass of some of the economiser at low loads using a damper.

Tube bundle vibration is where vortex shedding off the tubes causes resonance. Noise is generated when vortex shedding coincides with the acoustic frequency. Baffles can be introduced to counter this but selection of tube size and pitch can avoid vibration.

Figure 10 HRSG Example Arrangements



7.5.2 HRSG Performance

The HRSG flow is essentially counterflow and the temperature profile for an example HRSG is shown in figure 7.11. This diagram is the most useful representation of the heat recovery process, and is plotted as the temperature of the fluids against the heat transferred between the fluids. Sometimes these plots are against the fraction of the total heat transferred but this does not differentiate the amount of heat. In this figure the energy transferred is the energy per kilogram of exhaust gas mass flow.

The temperature profile in figure 7.11 contains all the necessary information for the heat recovery efficiency of the HRSG. The design of this temperature profile is fundamental to the bottoming cycle efficiency. The slope of the lines is $1/(mC_p)$, thus their relation is effected by the heat capacities and the mass flow rates and cannot be chosen arbitrarily. The relative mass flow rate is defined here as μ ;

$\mu = \dot{m}_S / \dot{m}_G$ where $\dot{m}_S$ and $\dot{m}_G$ are the steam and exhaust gas mass flows respectively. ... 7.10)

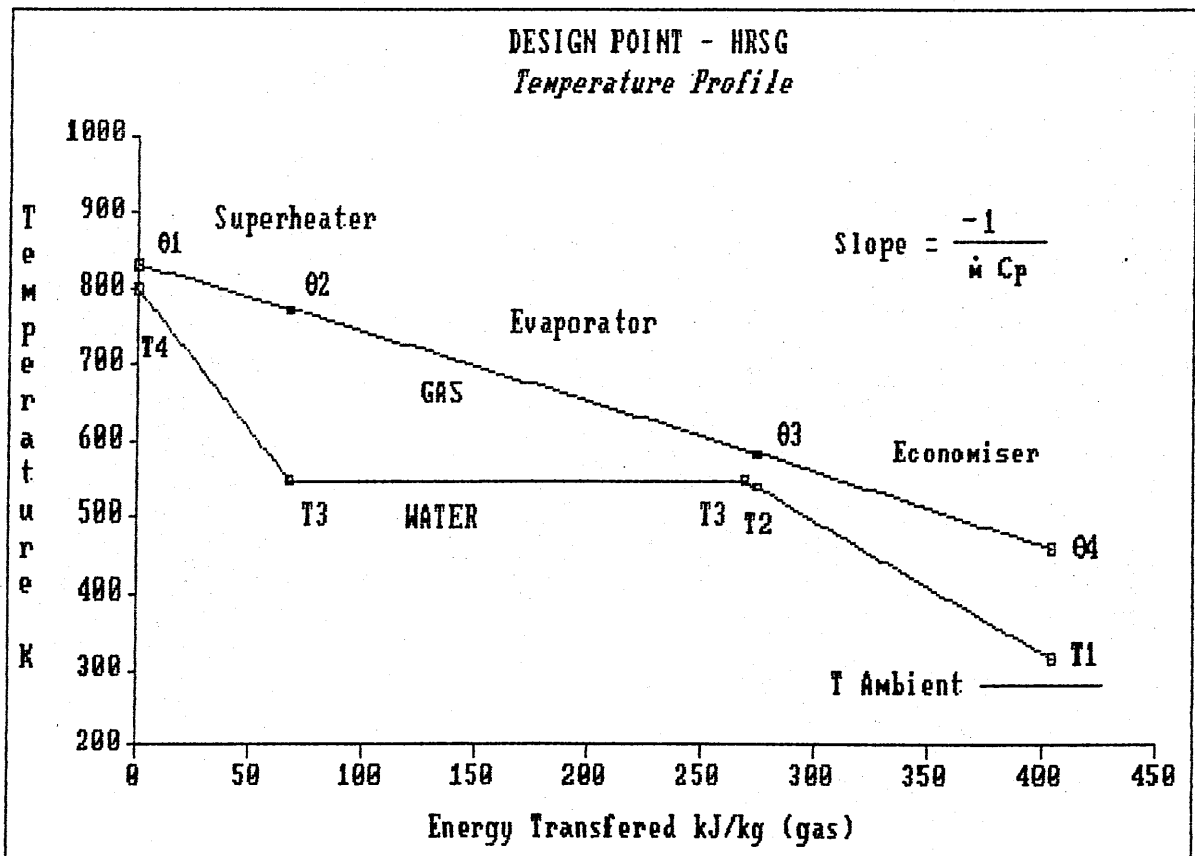
Another variable defined is the ratio of flow heat capacities, E;

$$E = \frac{\dot{m}_{COLD} C_{PCOLD}}{\dot{m}_{HOT} C_{PHOT}} \quad \dots 7.11)$$

In figure 7.11 E gives the information on the relationship of the two lines. When E is unity the lines are parallel. It should be recalled that entropy is defined as heat transfer over a temperature difference. To minimise the heat transfer irreversibility losses the temperature differences in figure 7.11 should be minimised, although reductions in differences requires more tube surface area and so more capital cost. The most efficient heat exchanger would have parallel lines, and $E = 1$ throughout, but this is impossible due to the infinite heat capacity of the boiling process. The profile must be adjusted to the best compromise. It can be seen that it is not possible to look at the HRSG from inlet and outlet conditions only as a critical pinch point may be internal.

It is important to note that the HRSG heat exchanger effectiveness is not the same as the heat recovery efficiency. The effectiveness is the ratio of the temperature drop of the hot gas to the maximum possible, which would be to the feed water inlet temperature. The heat exchanger

Figure 7.11 Example Single Pressure HRSG Temperature Profiles



efficiency is the ratio of the heat transferred to the heat available in the hot gas. The heat available is related to the ambient temperature of the intake of the gas turbine. Referring to the notation of figure 7.11;

$$\text{Heat exchanger effectiveness } \epsilon = \frac{(\theta_1 - \theta_4)}{(\theta_1 - T_1)} \quad \dots 7.12)$$

Heat recovery efficiency

$$\eta_{HR} = \frac{\text{heat transferred}}{\text{heat of exhaust gas}} \approx \frac{(\theta_1 - \theta_4)}{(\theta_1 - T_{AMB})} \quad \dots 7.13)$$

The limiting variables at design point in the heat exchange process are listed here individually.

Maximum Steam Temperature (T_4):

The outlet or maximum steam temperature is dictated by the materials of the steam turbine, but must be below the exhaust temperature as well. If the exhaust temperature of the gas turbine is large, the steam outlet temperature is reduced by increasing the feed flow and reducing the slope of the lines in figure 7.11. Typically the material maximum (T_{MAX}) is about 810-840 K.

Minimum Water Temperature (T_1):

The inlet or minimum water temperature is dictated by the condenser cooling water temperature and any regeneration in the steam cycle, such as a deaerator. The condenser would typically be at about 10-20°C above the cooling water temperature for practical designs, and so in the region of 315 K (40°C). The cooling water temperature depends on the source of cooling, such as cooling towers, river or sea water. For sulphur containing fuels with a tendency for low temperature corrosion the inlet feed temperature will need to be increased to minimise the likelihood of condensation.

Gas Maximum Temperature (θ_1):

The gas turbine exhaust temperature is the gas inlet temperature. This is chosen in gas turbine selection and dictated mainly by the gas turbine TET and the pressure ratio. At part load the exhaust temperature would fall.

Gas Minimum Temperature (θ_4):

Ideally the exhaust gas minimum temperature, or 'stack' temperature would be reduced to ambient, although it must be above the water inlet temperature. In reality the temperature is limited for two reasons. Firstly the temperature must be sufficient to prevent condensation of corrosive exhaust compounds, especially when burning poor fuels containing sulphur, and secondly the size and cost of the heat exchanger rises exponentially with reduced minimum temperature. Typically the minimum temperature (θ_{MIN}) is kept above 100-150 °C, although with clean natural gas fuel a condensing temperature down to 90 °C may be used. The exhaust temperature may not be dictated by the low limitation but by the boiling temperature (T_3) and diverging lines in the economiser, figure 7.11.

Pinch Point (ΔT_{PINCH}):

The pinch point is the minimum temperature difference between the two

fluids in the heat exchanger, either or both of $\theta_1 - T_4$ and $\theta_3 - T_3$. The size and cost of the heat exchange surfaces rises exponentially with reduced pinch point thus a compromise must be made. Pinch points in real combined cycles range around 10-30 °C. There can be two pinch points if the steam maximum temperature is dictated by low exhaust temperature. If the exhaust temperature (θ_1) is well above the steam material maximum then there would be only one pinch point $\theta_3 - T_3$, and if the exhaust temperature was large enough then the economiser lines in figure 7.11 would converge towards θ_4 and there would be effectively no limiting pinch point (figure [7.25]).

Approach Point ($T_3 - T_2$)

To prevent evaporation in the economiser the temperature at outlet of the economiser (T_2) is designed to be just short of the boiling temperature (T_3). This difference is called the approach point.

Boiling Point (T_3):

The boiling point is fixed by the pressure in the steam drum. Increased steam pressure increases the boiling point. The limitations are the steam turbine exhaust quality limiting the maximum pressure, and low steam cycle efficiency limiting the minimum pressure.

Real steam cycles include deaerators, feedheaters, and feed storage tanks. Deaerating is required to reduce dissolved aggressive gasses such as oxygen and carbon dioxide. For example the maximum permissible oxygen content in the feedwater for a single pressure (3-4 MPa) is about 20 ppb [248]. Deaeration is more critical for higher steam pressures. The deaerator uses extracted steam and provides some feedheating. This increases the temperature of the water entering the economiser, which may be necessary with poor fuels to reduce the tendency for corrosion. Typically the feedwater must be heated to about 125°C for light fuel oils and 140°C for heavy oils. Some plant obtain cost reductions by eliminating the deaerator and providing deaeration in the condenser hot well, and with no feedheating the water inlet temperature is controlled by recirculation. The effect of regeneration (feedheating) on the performance is discussed in section 7.7.4.

7.5.3 HRSG Theory

A mathematical model of the heat exchanger has been developed. The model is integrated on a spreadsheet with the gas turbine and steam turbine models previously presented, and is included in appendix A. It calculates a mass flow of steam for maximum acceptable steam exit temperature using the energy balance and temperature profile limitations. The gas turbine exhaust temperature and the feed water inlet temperature are fixed external to the boiler, thus the two degrees of freedom within the HRSG are the steam outlet temperature and the gas outlet temperature. Notation was introduced in figure 7.1 and 7.11.

The maximum steam temperature is the gas turbine exhaust temperature less the pinch point, or the maximum acceptable steam temperature, whichever is the smallest. This temperature can be thought of as being controlled by the feed mass flow rate. Using the notation of figure 7.11 we get;

$$T_4 = \theta_1 - \Delta T_{\text{PINCH}} \quad \text{but if } T_4 > T_{4\text{MAX}} \text{ then } T_4 = T_{4\text{MAX}} \quad \dots 7.14$$

where T_4 , T_{4MAX} and θ_1 are the steam exit, steam maximum and gas inlet temperatures respectively.

The exhaust gas temperature is guided by the pinch point on the evaporator boiling temperature, figure 7.11, or the minimum acceptable gas temperature whichever is the largest. If the lines in the economiser, figure 7.11, diverge away from the evaporator ($E_{ECON} < 1$), the exhaust gas minimum temperature is controlled by the steam drum pressure. This is normally the case in unfired HRSG's. The energy balance is used, which for the economiser using the notation of figure 7.11 is approximately;

$$\text{energy balance economiser } \dot{m}_G C_{pG}(\theta_3 - \theta_4) = \dot{m}_S C_{pS}(T_2 - T_1) \quad \dots 7.15)$$

where $\dot{m}_G$, C_{pG} and $\dot{m}_S$, C_{pS} are the mass flow and heat capacities of the exhaust gas and water feed respectively.

The energy balance gives θ_4 (gas outlet temperature) from T_3 and ΔT_{PINCH} which is tested against the minimum acceptable temperature.

$$\text{Limiting relation if } \theta_4 > \theta_{MIN} \text{ then } \theta_3 = T_3 + \Delta T_{PINCH} \text{ else } \theta_4 = \theta_{MIN} \quad \dots 7.16)$$

where ΔT_{PINCH} is the second pinch point and θ_{MIN} is the minimum allowable exhaust temperature.

This method is summarised as follows (using notation of figure 7.11;

- T_1 given or known from the condenser pressure (see appendix B)
- θ_1 and $\dot{m}_G$ given from the gas turbine calculations
- T_3 (boiling point) from steam drum pressure (see appendix B)
- T_4 from equation 7.14
- if $\theta_4 - \Delta T_{PINCH} > T_{4MAX}$ then $T_4 = T_{4MAX}$ else $T_4 = \theta_4 - \Delta T_{PINCH}$
- Case 1, Evaporator limiting
 - θ_3 from $\theta_3 = T_3 + \Delta T_{PINCH}$
 - $\dot{m}_S$ from an energy balance over superheater and evaporator;

$$\dot{m}_G C_{pG}(\theta_1 - \theta_3) = \dot{m}_S C_{pS}(T_4 - T_3) + h_{fg} + C_{pF} T_{APPROACH}$$
 - T_2 from $T_2 = T_3 - \Delta T_{APPROACH}$
 - θ_4 from an energy balance over the economiser

$$\dot{m}_G C_{pG}(\theta_3 - \theta_4) = \dot{m}_S C_{pW}(T_2 - T_1)$$
- Case 2, Exhaust Temperature Limiting
 - $\theta_4 = \theta_{4MIN}$
 - $\dot{m}_S$ from energy balance over entire HRSG

$$\dot{m}_G C_{pG}(\theta_1 - \theta_4) = \dot{m}_S (h_{S4} - h_{S1})$$

where C_{pG} , C_{pS} , C_{pF} and C_{pW} are the average heat capacities of the gas, steam, saturated liquid water and condensed water respectively for the temperature ranges applicable. And h_{fg} , h_{S4} and h_{S1} are the enthalpies of boiling, steam at exit and steam at inlet to the boiler respectively.

The smallest feed water flow, $\dot{m}_S$, from case 1 or 2 used and it's corresponding temperatures. Therefore $\dot{m}_S$, T_1 , T_4 , θ_1 and θ_4 are known.

7.5.4 HRSG Design Point Results

The above mathematical model is used for the design point calculations. The feed water mass flow rate used in this model is such as to give the maximum permissible steam temperature. The design point performance of the HRSG is then effected mainly by the exhaust temperature and the steam pressure. These two variables are varied in the calculations and the results shown in figures 7.12 and 7.13. This example is based on the following values.

Supplementary firing	nil
Steam drum pressure (P_{S3})	6 MPa abs
thus boiling temperature (T_3)	548.79 K
Pinch points, both (ΔT_{PINCH})	33 °C
Minimum allowable exhaust temperature (θ_{4MIN})	150 °C
Feed heating in economiser ($\Delta T_{APPROACH}$)	10 °C
Maximum allowable steam temperature (T_{4MAX})	530 °C
Condenser pressure	8.5 kPa abs
thus feed water temperature (T_1)	315.58 K

Given the exhaust conditions of the hypothetical gas turbine in section 3.4 of 153.2 kg/s at 831.9 K the following results are found;

Exhaust Stack Temperature (θ_4)	186 °C
HRSG effectiveness (ϵ)	72.1 %
Heat Recovery Efficiency (η_{HR})	68.5 %
Steam Flow	19.7 kg/s at 525.8 °C

The resulting temperature profile is in figure 7.11.

Varying the steam drum pressure varies the boiling temperature, and it's effect is seen on figure 7.12. It is seen that a low steam drum pressure is better for heat recovery as it allows the exhaust temperature to be reduced to a lower level. Here the exhaust temperature is not low enough for the minimum limit to be active, and is pinch point controlled. An important effect is that the boiling temperature does not vary as much as the pressure, and a 100% increase in pressure has reduced the heat recovery by 6%. The pressure change also changed the latent heat of vaporisation by 20% and the mass flow change is too small to be noticeable on the slopes. Increased steam pressure decreases heat recovery efficiency but would increase the steam cycle efficiency, and effectively η_{ST} increases while η_{HR} decreases, only slightly effecting the bottoming efficiency $\eta_{BC} = \eta_{ST}\eta_{HR}$.

Effects of changes to the design point gas turbine exhaust temperature can be seen in figure 7.13. At higher exhaust temperatures the ratio μ of mass flow of steam to exhaust gas is higher, and leads to the slopes of the graph being more parallel in the economiser. This reduces the exhaust temperature and improves the heat recovery. The change from exhaust temperature of 700 to 900 K changes the effectiveness ϵ from 48% to 81%, and the heat recovery efficiency η_{HR} from 43% to 74%. The higher exhaust temperature also means a higher steam temperature if the exhaust is sufficiently low to have been limiting it. Effectively the increase in exhaust temperature increases heat recovery efficiency η_{HR} and often steam cycle efficiency η_{ST} , and so generally increases bottoming efficiency η_{BC} .

Figure 7.12 HRSG Temperature Profile - Varying Feed Pressure

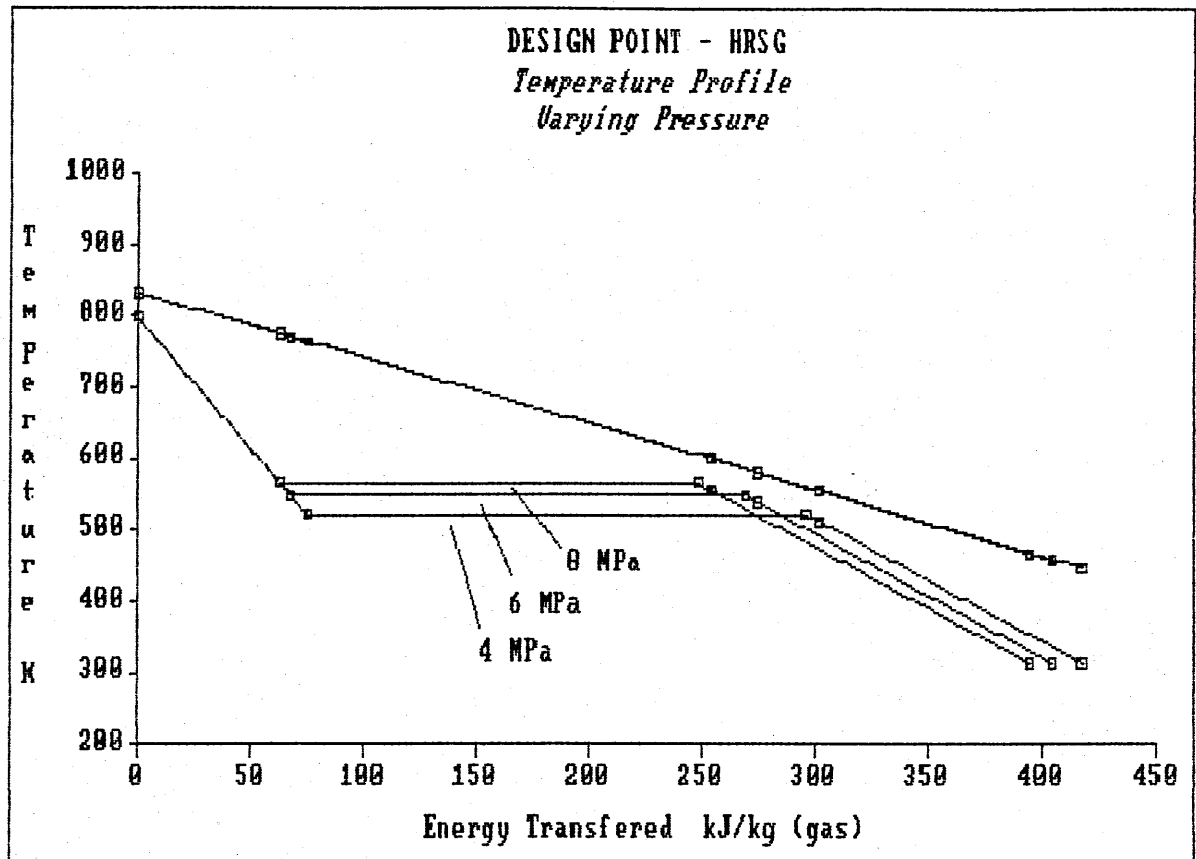
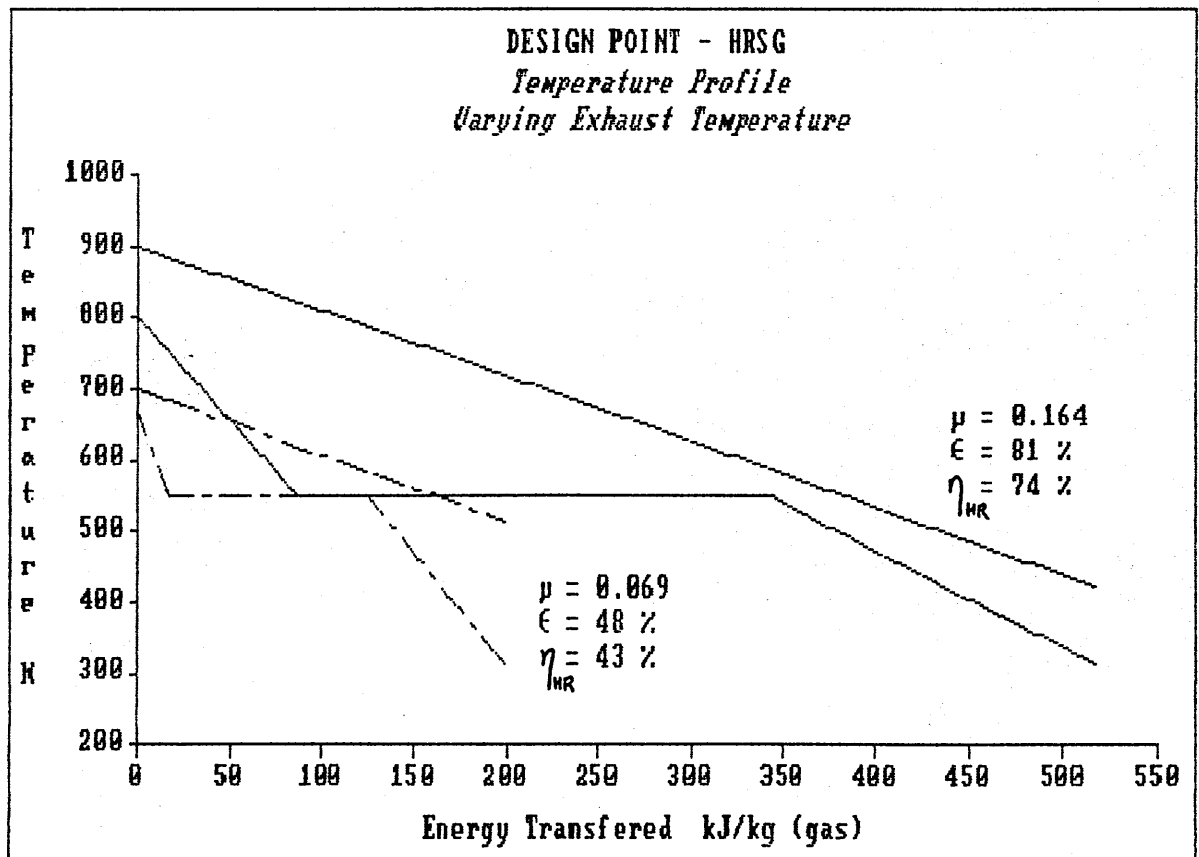
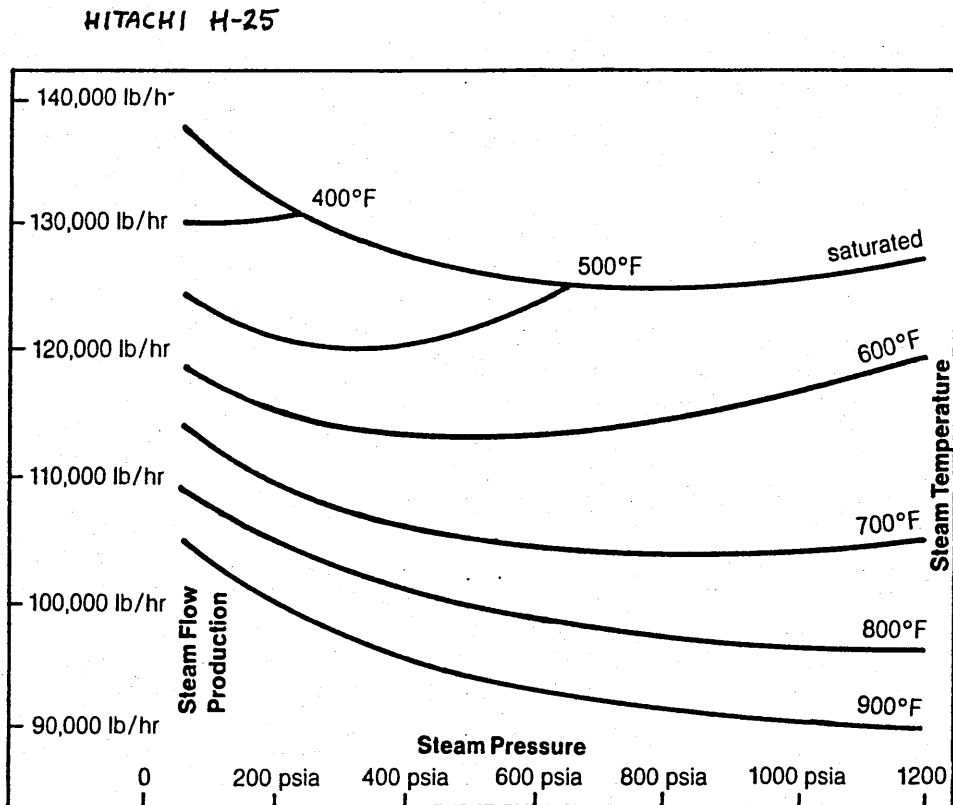


Figure 7.13 HRSG Temperature Profile - Varying GT Exhaust Temperature



A diagram can be produced giving the potential steam production from a specific gas turbine. A typical diagram is shown in figure 7.14, reference [138].

Figure 7.14 Steam Potential Production Map



7.6 COMBINED CYCLE DESIGN POINT

The combined cycle design point is calculated for a single pressure steam cycle from the gas turbine theory of section 3.4, steam turbine theory of section 6.2 and the above heat recovery theory. The analysis of a dual pressure cycles are given in a similar way in reference [193] which is discussed more fully in section 7.7.

The mathematical model of the combined cycle is the model already used for the gas turbine, section 3.4.2, HRSG, section 7.5.3 and the steam turbine 6.2.1. It is on a spreadsheet and included in appendix A. The steam properties, h and s , for the steam turbine comes from the pressure and temperature of the steam and polynomials fitted to steam tables over the ranges of interest. This is discussed more fully in appendix B, where it is shown that the error from the tables in the ranges used is less than 1%.

Using the mathematical model the GT pressure ratio and TET were varied for the design point, and the results are plotted in figures 7.15 and 7.16. The cycle values used were as in the design point of the gas turbine in section 3.4.3 and the HRSG in section 7.5.4. Additional assumptions were as follows;

Throttle pressure losses	5%
Steam turbine isentropic efficiency	80%
Condenser pressure	8.5 kPa
Feed pump isentropic efficiency	80%

The plots 7.15 and 7.16 are comparable to gas turbine performance figures 3.11 and 3.12 and represent combined cycles using the same gas turbines. These are design point results.

At low pressure ratios the lower gas turbine efficiency is accompanied by a greater exhaust gas temperature (figure 3.16) and a greater bottoming cycle efficiency. For higher TET's such as 1400 K the effect of gas turbine pressure ratio on the overall efficiency is weak and the efficiency is nearly independent of pressure ratio over a wide range. This means from equation 7.6, that for reducing pressure ratio the reduction in η_{GT} is compensated for by the rapid increase in η_{HRGT} . The small diversion to lower efficiency at very low pressure ratios is due largely to the steam temperature being limited to 800 K.

Reference [185] comments that there are various different conclusions between commentators for the best GT pressure ratio in a combined cycle. This is not surprising given that the efficiency is nearly independent of pressure ratio at practical TET's and that the function is dependant on so many variables. The author believes the optimum pressure ratio is different for different steam cycles and heat recovery designs. It would be better to use an optimum exhaust temperature for each steam cycle.

The power output varies nearly linearly downwards from low pressure ratios to high pressure ratios. This effect is due to the constant size of gas turbine in the model, which with higher pressure ratios has less temperature rise available for the fuel input so has reducing fuel flow.

The split of power and efficiency with gas turbine pressure ratio between the gas turbine and the bottoming cycle can be seen in figures 7.17 and 7.18. This is not representative of size, as the gas turbine size is constant so power is giving a measure of specific power, whereas the steam turbine has nearly constant specific power and therefore it is a measure of varying size. It can be seen from figure 7.18 that the steam turbine is more dominant at very low pressure ratios, but as the pressure ratio climbs the steam turbine power and thus size gets smaller. This is due to the decrease of exhaust temperature as the pressure ratio and hence expansion ratio of the gas turbine increases. This was shown in figure 3.16.

Figure 7.19 shows the effect of gas turbine pressure ratio and hence exhaust temperature on the HRSG, condenser and steam turbine flow. As the pressure ratio increases and the exhaust temperature decreases the HRSG effectiveness, steam mass flow and condenser heat flow decrease and hence the size of the HRSG, steam turbine and condenser decrease. This model uses a constant steam drum pressure whereas in reality the steam pressure would be reduced at lower exhaust temperatures, but this effect is small and only noticeable at very high pressure ratios. Variable steam pressure has more effect at part load where exhaust temperatures get much lower still.

Figure 7.15 CC Design Point - Efficiency vs GT Pressure Ratio

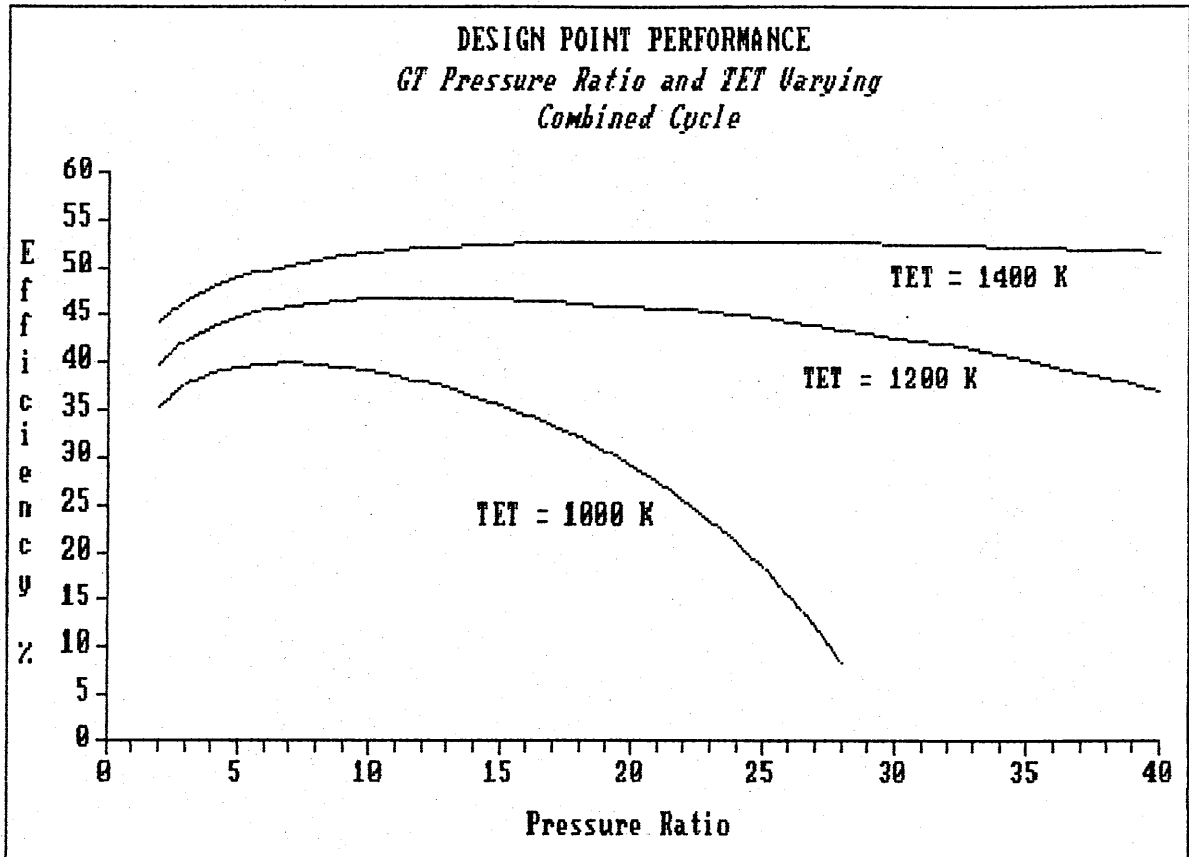


Figure 7.16 CC Design Point - Power vs GT Pressure Ratio

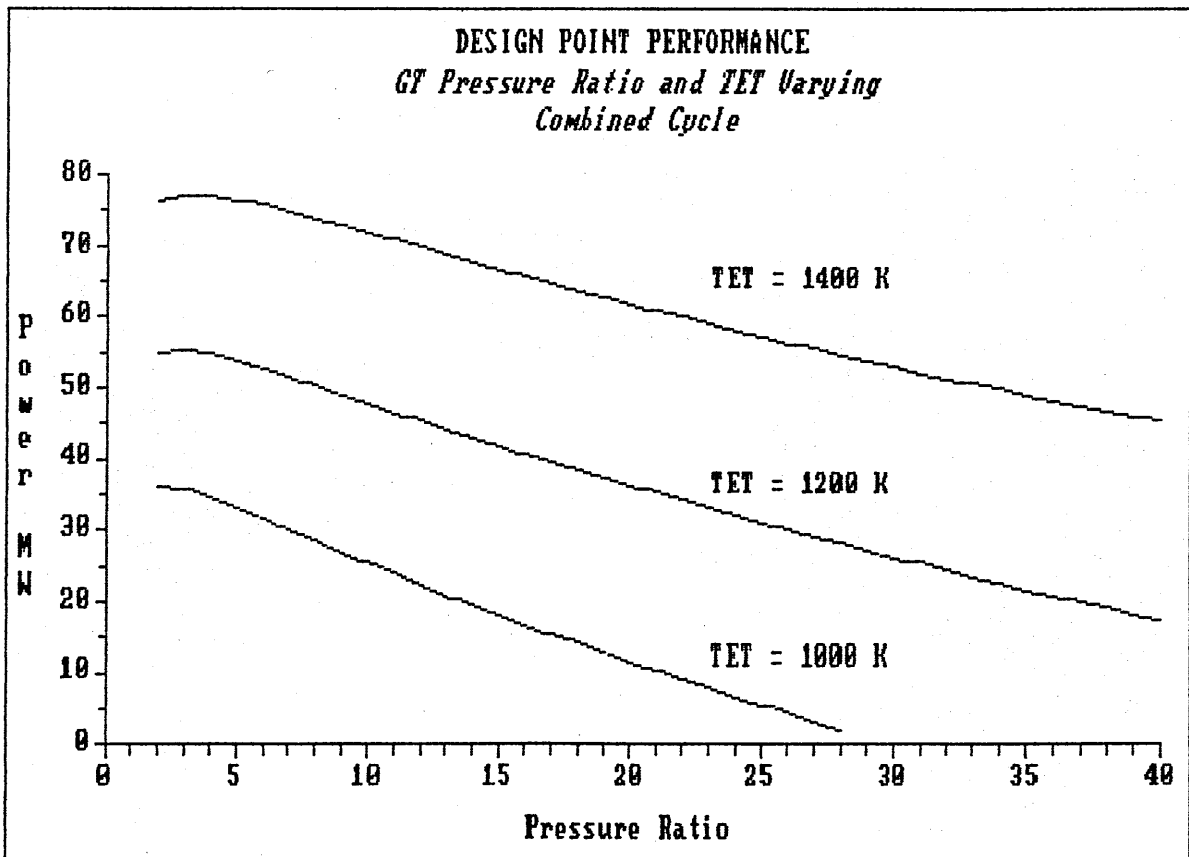


Figure 7.17 CC Design Point - Efficiency Split vs Pressure Ratio

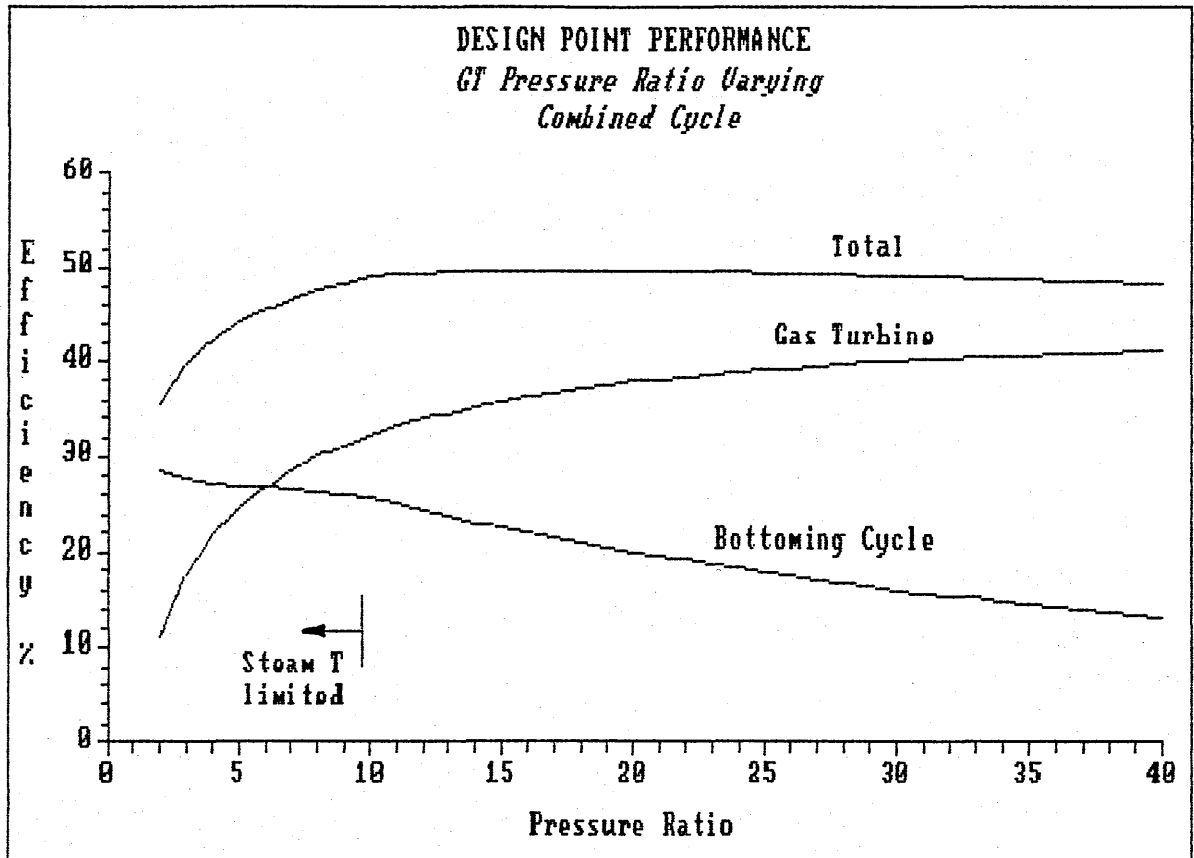


Figure 7.18 CC Design Point - Power Split vs GT Pressure Ratio

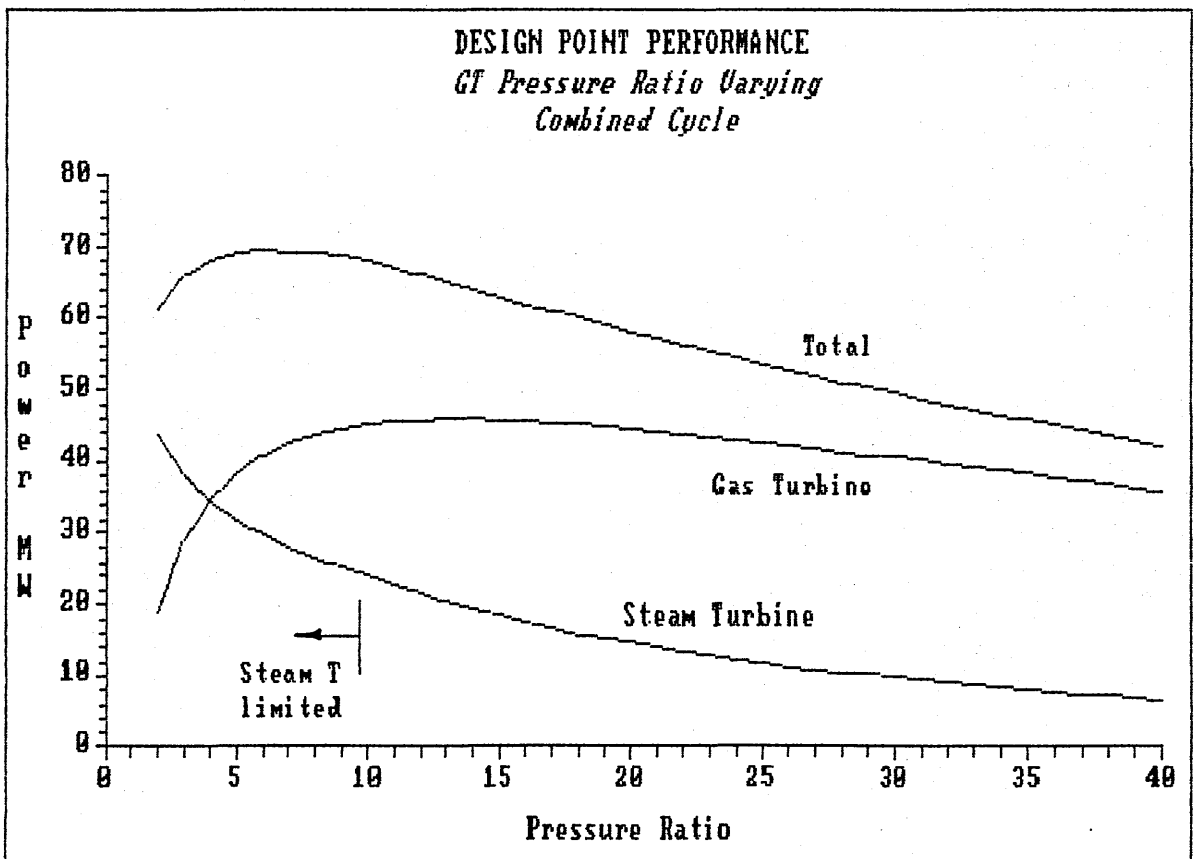
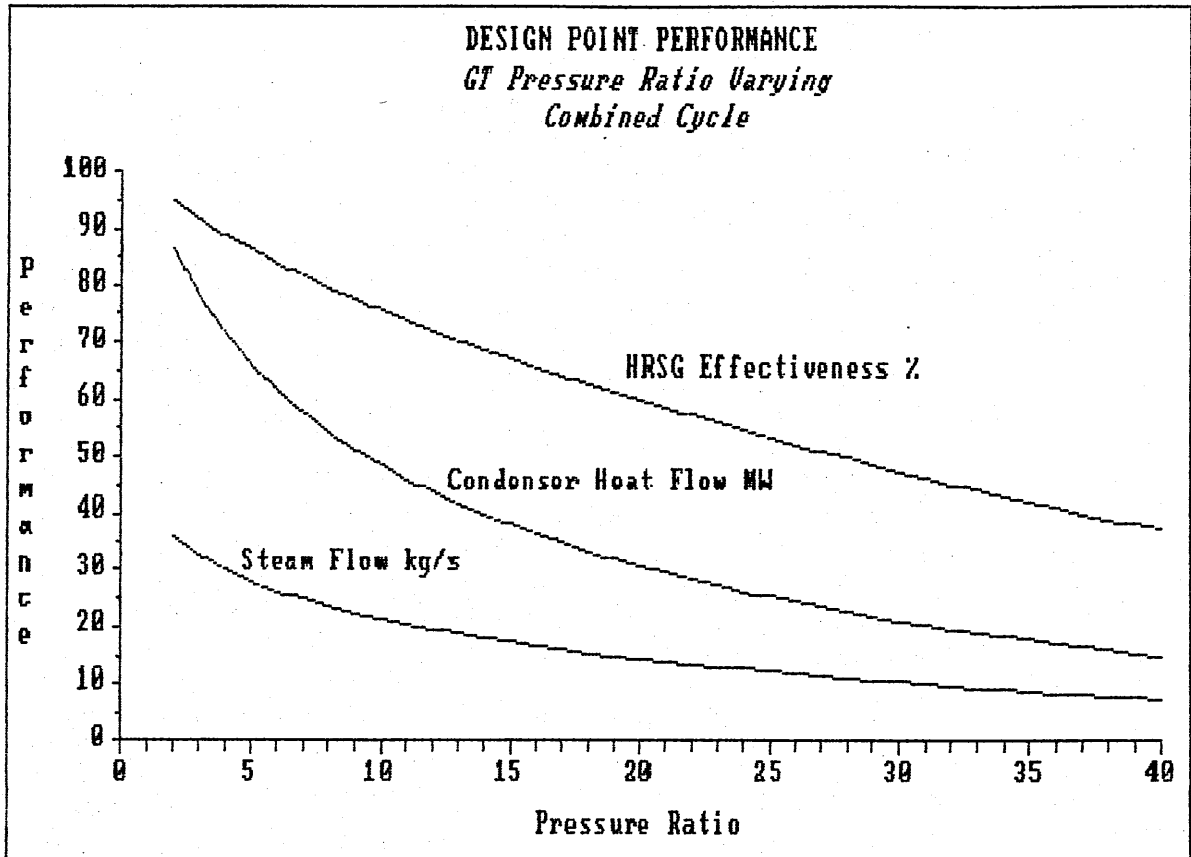


Figure 7.19 CC Design Point - Bottoming Cycle Characteristics vs GT r_p



In figures 7.20 and 7.21 the effect of gas turbine TET on the performance split of the gas to steam turbine is seen. The total performance effect has a similar shape to the effect on the gas turbine alone, figures 3.13 and 3.14. The steam turbine runs out of power at low TET in this example when the GT exhaust temperature is less than the steam drum boiling temperature. In reality the steam pressure would be reduced for reduced temperatures.

The effect of steam drum pressure can be seen in figure 7.22. Increasing steam pressure increases the steam cycle efficiency η_{ST} but reduces the heat recovery efficiency η_{HR} (section 7.5.4). The net effect on the bottoming efficiency, $\eta_{BC} = \eta_{ST}\eta_{HR}$, is increasing bottoming efficiency for increasing pressure when at higher gas turbine exhaust temperatures and reducing bottoming efficiency when at lower exhaust temperatures. In this example the net effect is a small positive gain. Higher steam pressures mean more expensive equipment, lower first stage isentropic efficiencies in the turbine and lower low pressure stage isentropic efficiencies due to lowered steam quality. The wetter low pressure turbine stages may require steam reheat, which again effects the heat recovery efficiency as will be seen later.

Figure 7.20 CC Design Point - Efficiency Split vs Gas Turbine TET

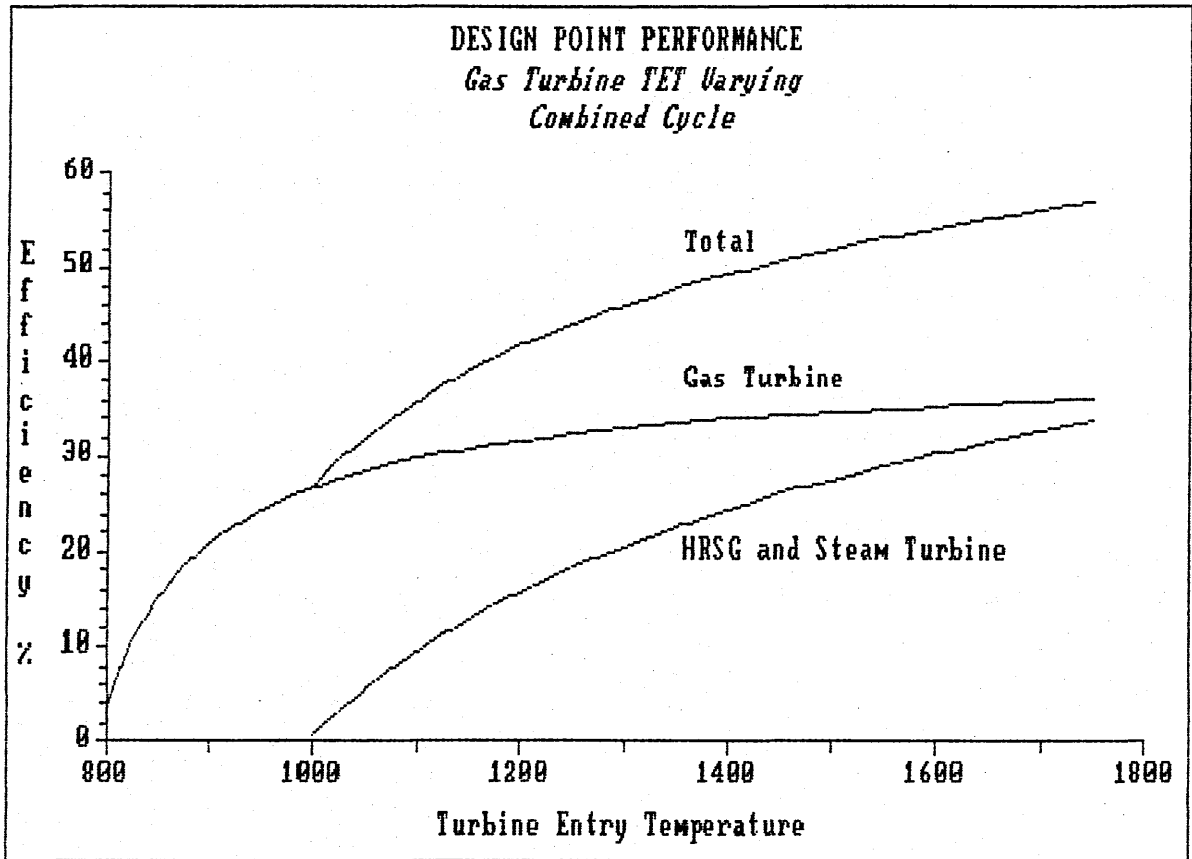


Figure 7.21 CC Design Point - Power Split vs Gas Turbine TET

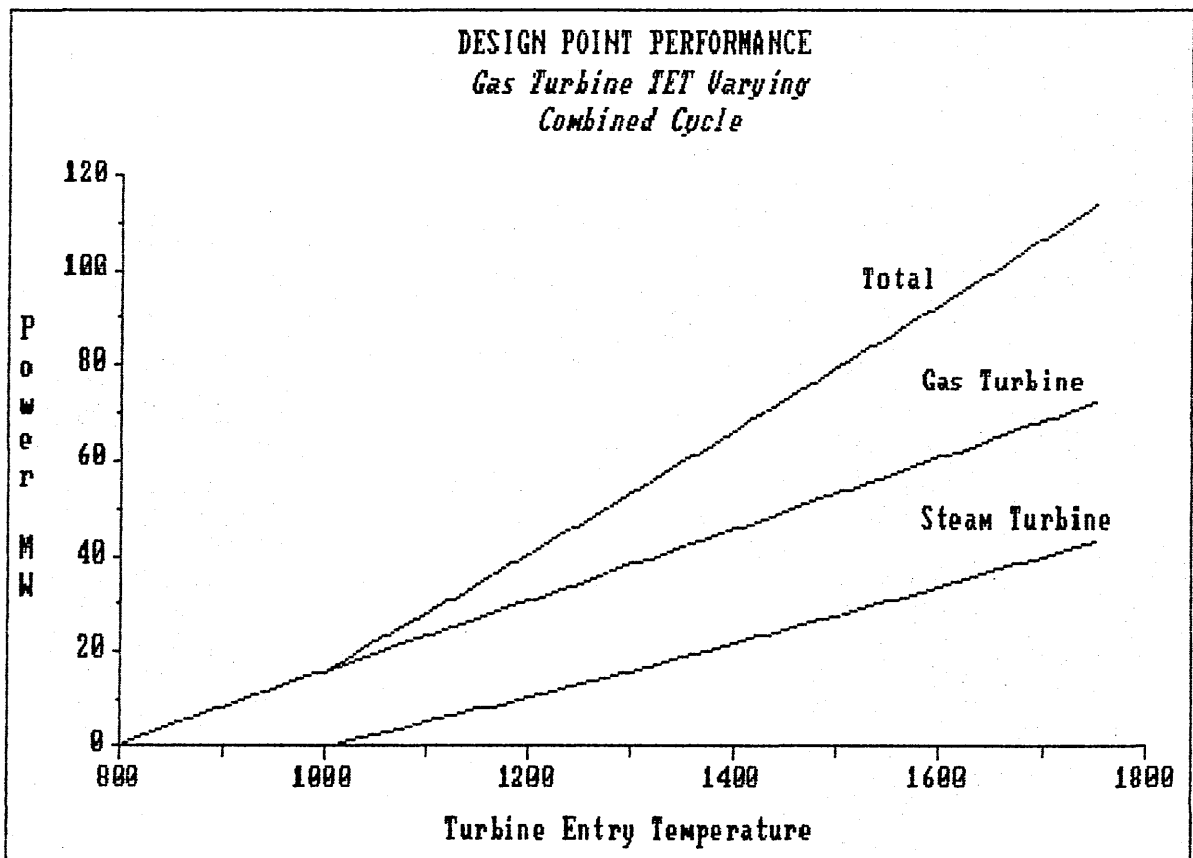


Figure 7.22 CC Design Point - Varying Steam Drum Pressure

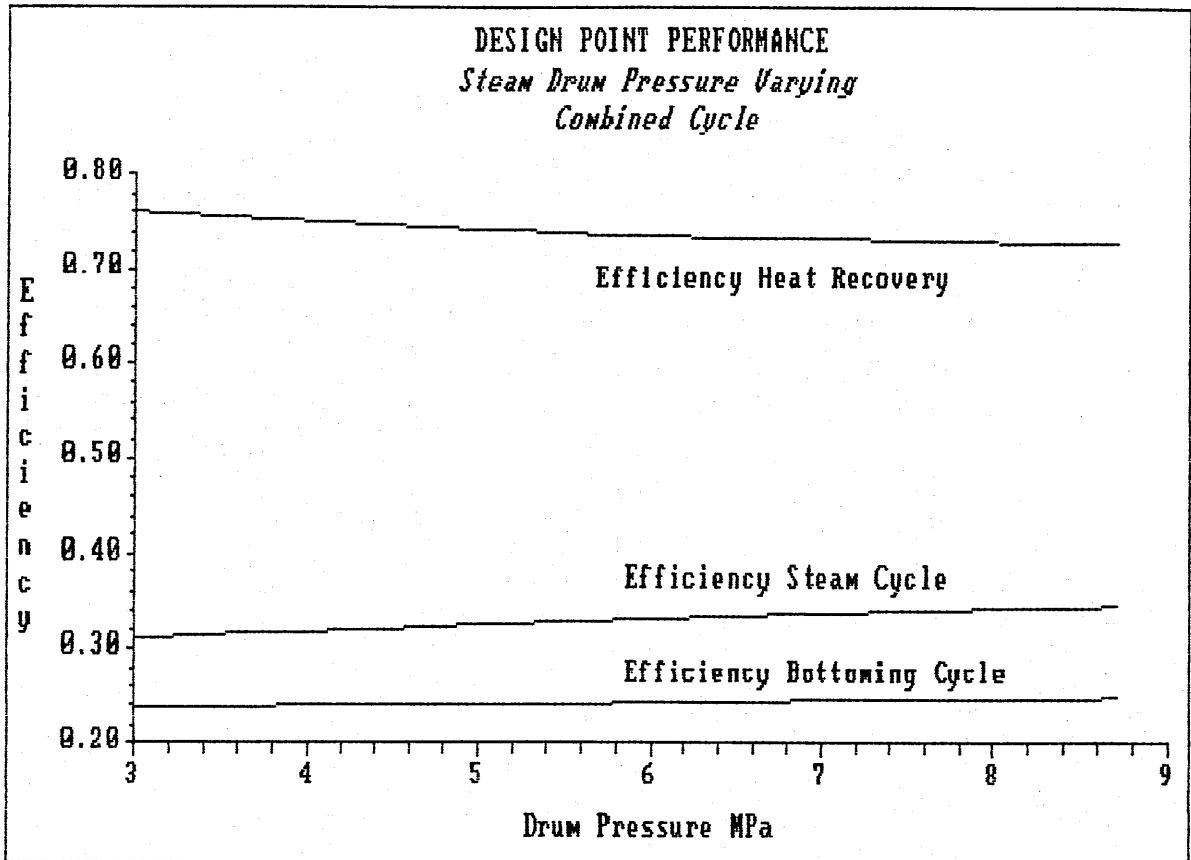
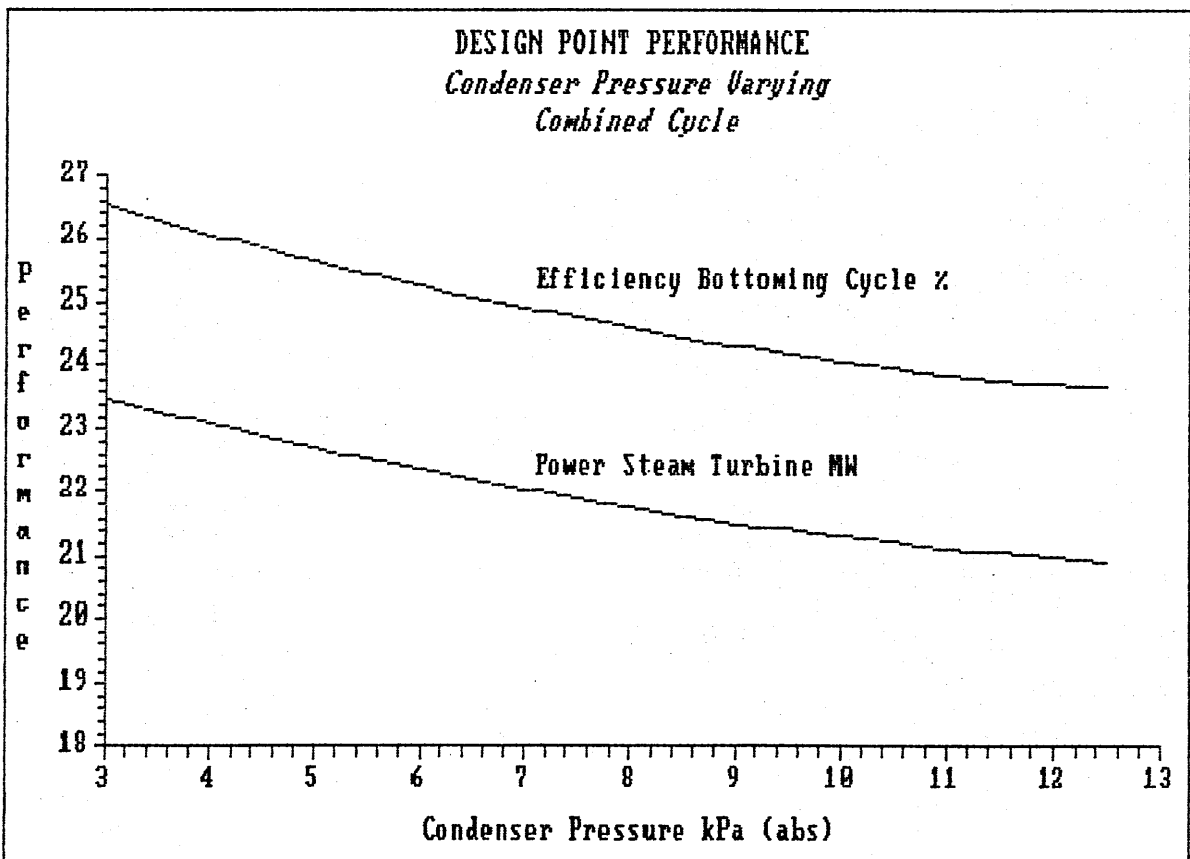


Figure 7.23 CC Design Point - Varying Condenser Pressure



The effect of condenser pressure can be seen in figure 7.23. Decreasing the condenser pressure improves the steam turbine pressure ratio and improves the steam cycle efficiency. The reduced pressure requires a reduced temperature, and either colder cooling water or a larger condenser heat exchanger. This also reduces T_1 on figure 7.11 and may increase the heat recovery efficiency. Another effect is to decrease the quality of exhaust steam and to increase the required size of the last steam turbine stages. The cooling water temperature is dictated by the cooling process, either river water, sea water, cooling towers or air coolers. An example is a cooling tower in air at 15°C and 60% humidity may give about 7 kPa for a once through system with the water at 25°C.

7.7 IMPROVING THE HEAT RECOVERY

The low efficiency of the heat recovery process at low gas turbine exhaust temperatures can be improved in several ways. The exhaust gas temperature can be increased, or a more complex HRSG can be used to improve the temperature profiles. If more firing to boost exhaust temperature is to be used the common way is firing fuel in the exhaust, called supplementary firing. A better although unused way would be to use a reheat chamber part way through the gas turbine expansion. The most common way of improving the HRSG temperature profile is by two or three pressure HRSG's. Gas turbine TET's have tended to increase in the past years increasing the exhaust temperature, and this has improved the heat recovery process allowing more complex cycles.

7.7.1 Supplementary Firing

Supplementary firing is burning more fuel in the exhaust of the gas turbine to improve the temperature profiles in the HRSG, allow a more complex steam cycle or boost steam turbine power. The improvement in heat recovery efficiency and steam conditions may compensate for the addition of fuel heat at a lower temperature. The solution of using supplementary firing only works well for otherwise low exhaust temperatures, and is more expensive in capital and control facilities than unfired systems. Supplementary firing has the advantage of being able to vary the heat to power ratio in CHP situations, and can be configured to allow continuation of the process steam when the gas turbine is down for maintenance. This is one more degree of freedom in control for CHP, useful where the power is used for frequency control and cannot be varied at will.

The fuel is either burnt in the exhaust duct between the gas turbine and the HRSG or in the HRSG using conventional boiler burners and cooled combustion chambers. The two are roughly equal in price for supplementary firing at 1000 K but the cooled combustion chambers are more robust and are better suited to burning poorer fuels [197]. This means that supplementary firing can burn cheaper fuels of lower quality than are acceptable in the gas turbine. The limit in firing is the exhaust oxygen content, which is somewhere in the vicinity of 2000 K, but not practically attainable.

Figure 7.24 CC Design Point - Efficiency vs Supplementary Firing

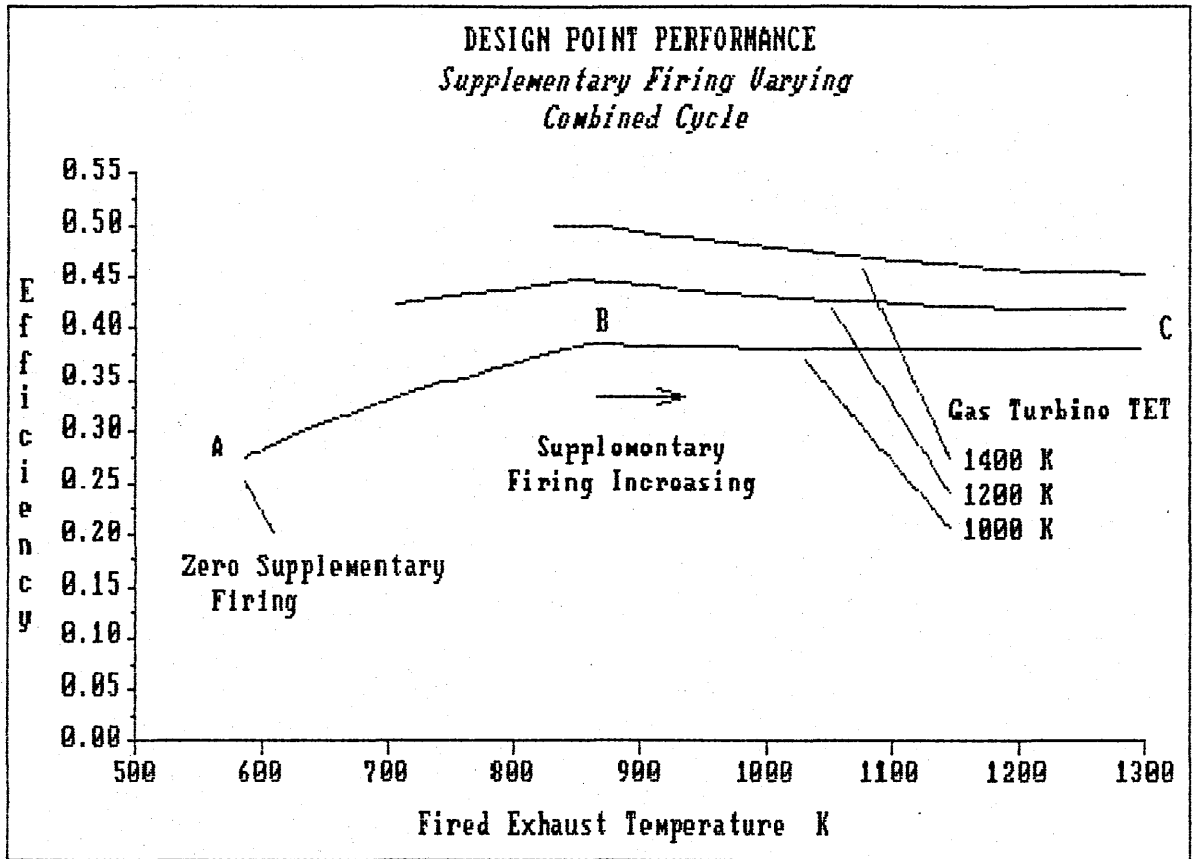
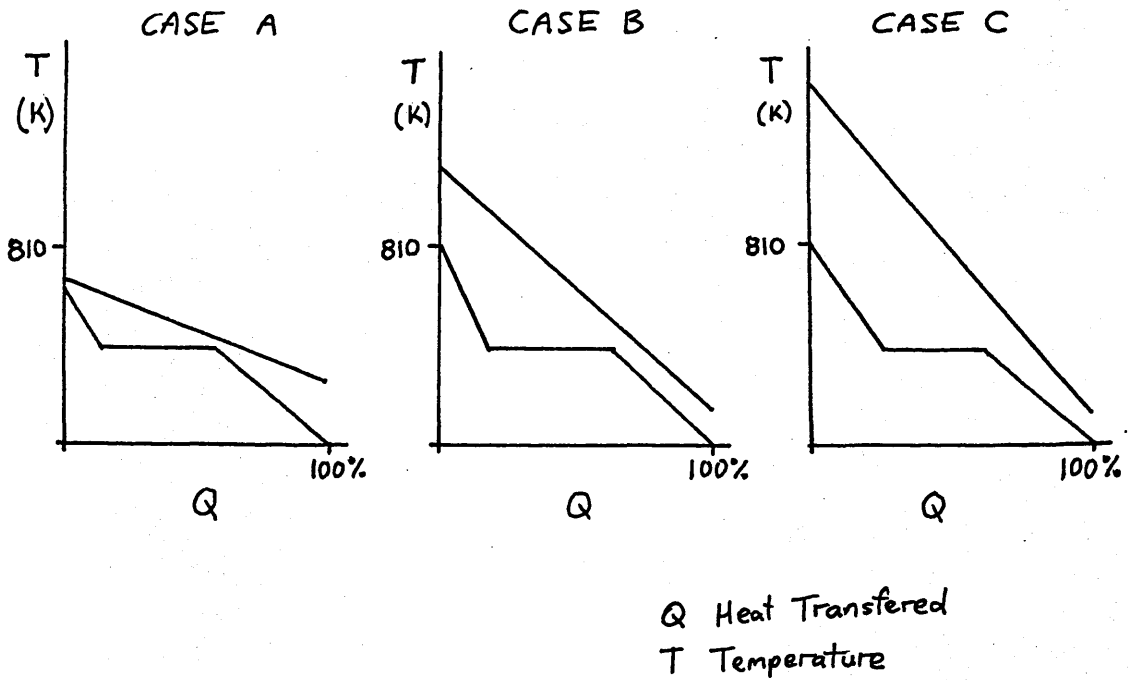


Figure 7.25 HRSG Temperature Profiles - Varying Supplementary Firing



The mathematical design point model in appendix A was adjusted to include supplementary firing, and a plot of the effect on performance is given in figure 7.24. This plot is a varying design point plot, and not an off design plot. The effect is that the efficiency of the combined cycle improves until the exhaust temperature reaches some maximum, after which more firing would reduce the efficiency again. The three plots are for pressure ratios of 12 and varying TET, hence varying exhaust temperature. The plots start with no firing on the left and increasing firing as they traverse to the right. The initial efficiency gain is due to the effect of heat recovery improvement being stronger than the lowered efficiency due to heat addition at a lowered temperature. The effect on the temperature profiles can be seen in figure 7.25 and shows how the peak efficiency occurs when the temperature profiles of the economiser are parallel, $E_{ECON} = 1$.

The low exhaust temperature gas turbine had the greatest improvement in performance for supplementary firing, whereas a gas turbine with exhaust temperature already at the peak performance temperature has no improvement. The apparently large performance gain in the 1000 K combined cycle is due to the very poor performance at the initially low exhaust temperature.

Supplementary firing also has the effect of increasing the steam turbine power output relative to the gas turbine, and in the extreme tends to the steam cycle efficiency. These plots would indicate the peak efficiency performance is at about 850 K fired exhaust temperature, whereas the figure used by the company ABB is 1000 K [197]. The actual peak performance figure would be different for different steam cycles, particularly if steam reheat was used. The ABB figure is based on a real steam cycle and experience, while the figure here is based on a simple HRSG and a simple mathematical model. However the trends are the same as those published by ABB [197].

The constant exhaust firing temperature for maximum efficiency in the supplementary firing would appear to indicate that the optimum pressure ratio in the gas turbine is that which gives this optimum exhaust temperature. This would mean no supplementary firing is necessary and maximises the temperature of heat addition. A pressure ratio greater than this would give a lower exhaust temperature and mean the cycle could be improved even at the expense of lower temperature heat addition. A lower pressure ratio than this would mean the exhaust temperature was too high and would be improved by conversion of more work in the GT at higher temperatures.

Supplementary firing loses its importance with regard to efficiency as development of gas turbines progresses giving improved exhaust temperatures.

7.7.2 Gas Turbine Reheat

Reheating the gas part way through the gas turbine expansion is a better solution to raising gas turbine exhaust temperature than supplementary firing. It increases the temperature of heat addition. Obviously raising the gas turbine TET would be the best solution but this is limited by material strength. Reheat results in more improvement than for supplementary firing and over a much more general range of exhaust temperatures. Ideally the cycle performance is also less sensitive to

the peak temperature. The optimum pressure ratio increases with the number of reheats and both the specific power and the efficiency are improved [193]. Reheat also improves part load efficiency by maintaining better part load exhaust temperature, which assists the steam cycle to maintain part load efficiency.

In external combustion gas turbines, such as with coal, the TET may be limited to a lower temperature of say 1100 K. This requires a low pressure ratio to give adequate exhaust temperature for the HRSG. The performance is then improved significantly by introducing reheat [220]. Although reheat has fundamental thermodynamic advantages in combined cycles, it has limiting problems in practice. The reheat combustor inlet air would be too hot for cooling purposes so all cooling would come from compressor bleed. With a larger proportion of the gas path exposed to elevated temperatures there is an increased cooling requirement and minimising these cooling losses is critical to realising the full potential of reheat cycles. The cooling penalty can result in lower optimum peak temperatures than in simple cycles and may offer little or no advantage without advanced cooling methods.

A reheat combustor is at a lower pressure so would be relatively large for the required flow velocities. The air to the reheat combustor has depleted oxygen which has flame stability ramifications, especially if two reheat combustors were used. Oxygen concentration into the reheat chamber might be say 11-15%. The inlet air to the reheat combustor would be at say 920 to 1030 K and Mach 0.4 - 0.5. This requires a diffuser and therefore added length, more cooled surface and pressure loss. The reheat combustor would also require a mixing pressure loss although the fundamental heating losses would be relatively small due to the smaller temperature rise. The location of the reheat might be limited as it can only be located between turbine stages, and probably after a gas generator. The reheat TET would be limited due to the reheated pressure ratio and the exhaust duct maximum material temperature.

In spite of the development problems reheat offers significant potential advantages in a combined cycle and much discussion and research has been completed. It is likely to be included in a future generation of commercial machines. The current notable example is the Japanese project for an advanced intercooled and reheat high pressure ratio combined cycle discussed in section 8.5.5. Some references for the reheat discussions are [191], [186], [187], [188], [259], [260] and [261].

7.7.3 Complex HRSG Temperature Profiles

Another way to change the temperature profiles in the HRSG, other than with changed GT exhaust temperature, is to change the temperature profiles by use of complex steam cycles. The most common method is the use of a two pressure boiler. The two pressure boiler comes in two main forms, with a two pressure steam turbine it increases heat recovery efficiency, and with a single pressure steam turbine and feed heating it decreases heat recovery. Increase in heat recovery is possible for the two pressure steam turbine as the exhaust stack temperature is reduced, figure 7.27, with a corresponding example temperature profile in figure 7.29. The two pressure boiler with single pressure turbine and feedheating has reduced heat recovery over the single pressure boiler as the exhaust stack temperature is raised, figure 7.26, with a temperature profile in figure 7.28. These temperature profiles are directly

comparable to the temperature profile for the single pressure HRSG in figure 7.11. The heat recovery efficiencies of these examples are;

Two pressure HRSG with feedheating	62.6%
Single pressure HRSG	68.5%
Two pressure HRSG and two pressure turbine at 3 and 12 MPa	73.6%

The single-pressure turbine and two pressure HRSG with feedheating is similar to the regenerative cycle that is described in the following section. It has a greater feedwater inlet temperature and does not reduce the gas turbine exhaust temperature as much as the other cycles. This is mainly of interest for oil fired plants with sulphur fuels and low temperature corrosion problems. For natural gas and sulphur free fuels it would not be used [251].

A dual pressure turbine and dual pressure HRSG increases the heat recovery efficiency from a single pressure HRSG, in this example by 5%, but splits the flow. Here only 67% is delivered at the full temperature, but at a higher pressure than the single pressure cycle (12 MPa instead of 6 MPa). The other 33% of the flow is delivered at a lower pressure (3 MPa) and also as saturated steam. This low pressure flow must be injected to the turbine at a low pressure stage doing less work than the high pressure flow. Overall the combined cycle efficiency improves about 3%.

A single pressure HRSG has an optimal exhaust stack temperature, which is not the lowest temperature, which depending on the pinch point lies between 160 to 200 °C [178]. This is because to reduce the stack temperature the steam pressure needs to be reduced. A dual pressure HRSG on the other hand allows reduction of the stack temperature to the minimum allowable by corrosion, with corresponding improved efficiency [178].

In real cases the improvement to combined cycle efficiency for dual pressure is about 3% over a single pressure HRSG [193]. The improvement is largely due to the reduction in heat transfer temperature differences, which represent entropy generation. The reduction in irreversibilities is about half, from about 15% of the exhaust gas energy to about 8% [193], in typical boiler designs. Three pressures are sometimes used, but generally to supply medium pressure steam for injection into the GT for NO_x reduction. Research has been carried out into use of a binary fluid mixture to give variable boiling temperature, notably the Katina cycle. This uses a water and ammonia mixture and uses varying composition to vary the temperature of boiling, and is discussed in section 8.5.6.

Figure 7.26 Example Dual Pressure HRSG – Single Pressure Steam Turbine

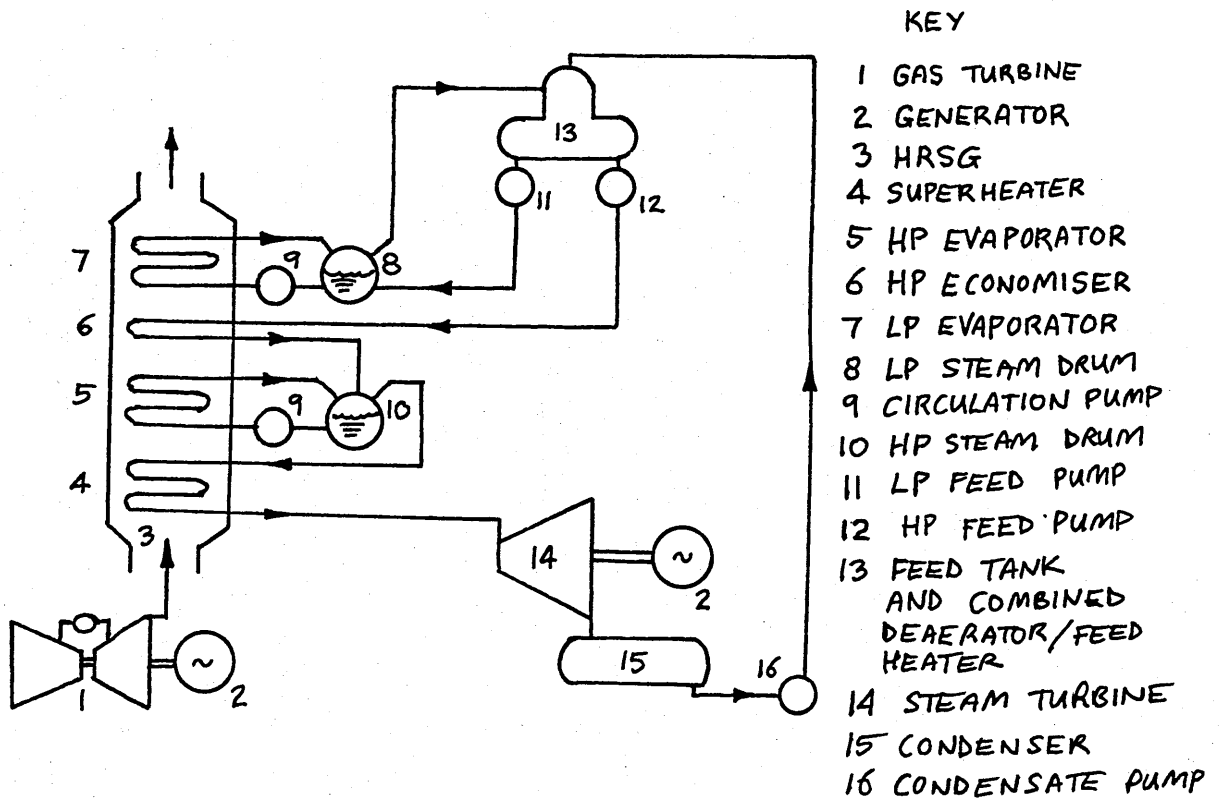


Figure 7.28 Dual Pressure HRSG - Single Pressure ST

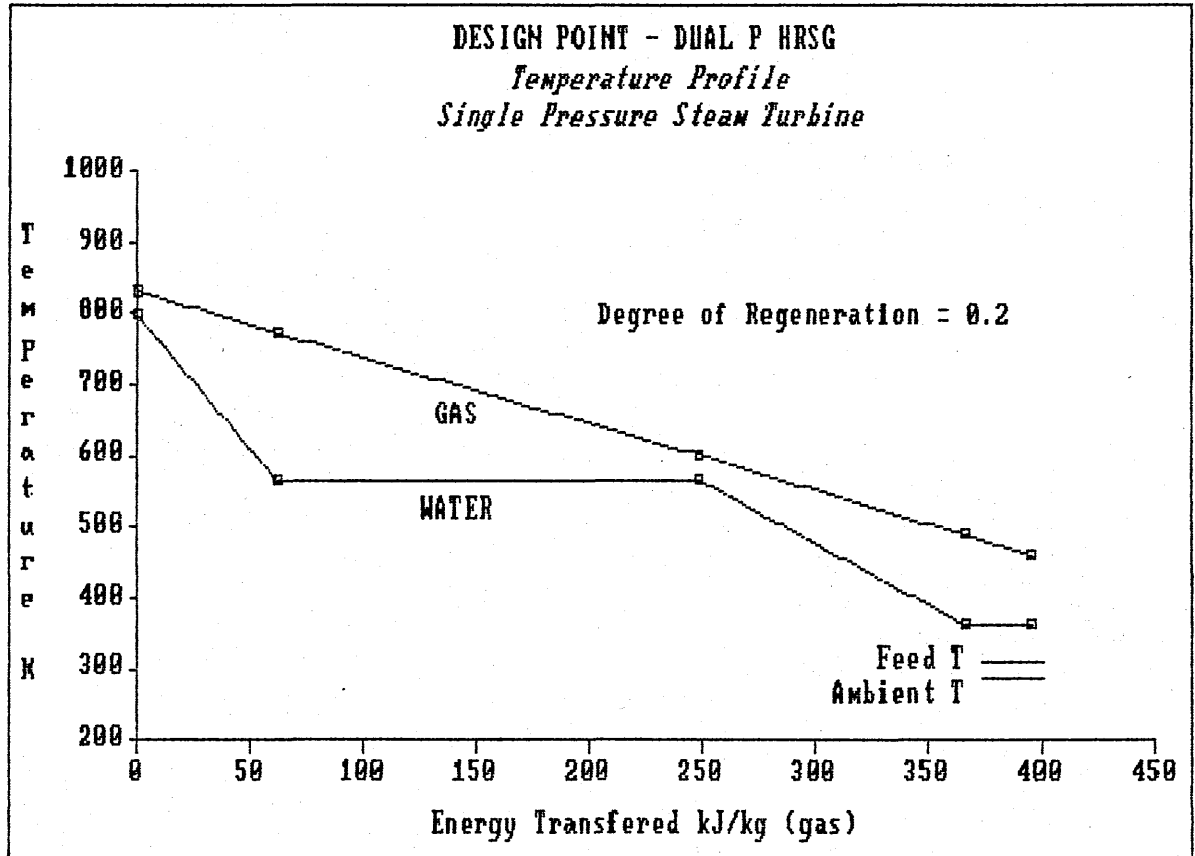


Figure 7.29 Dual Pressure HRSG - Dual Pressure Steam Turbine

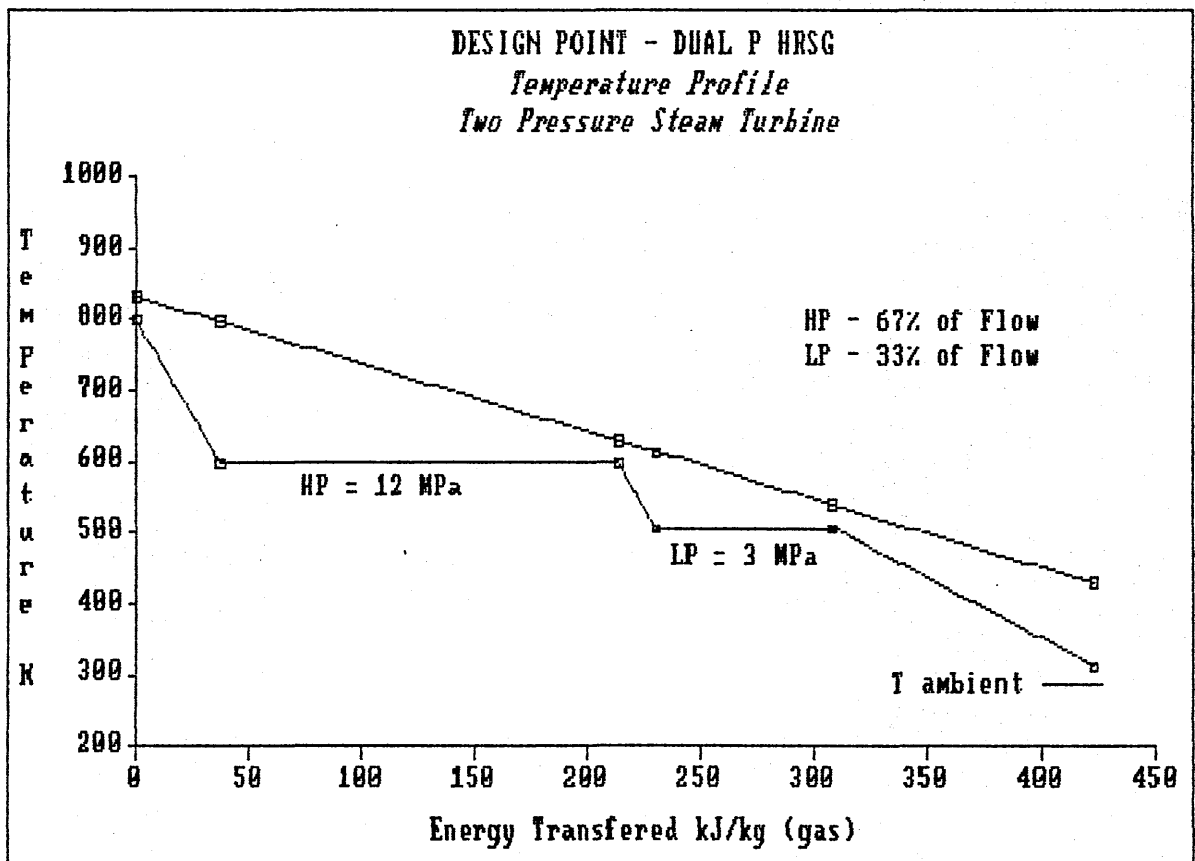
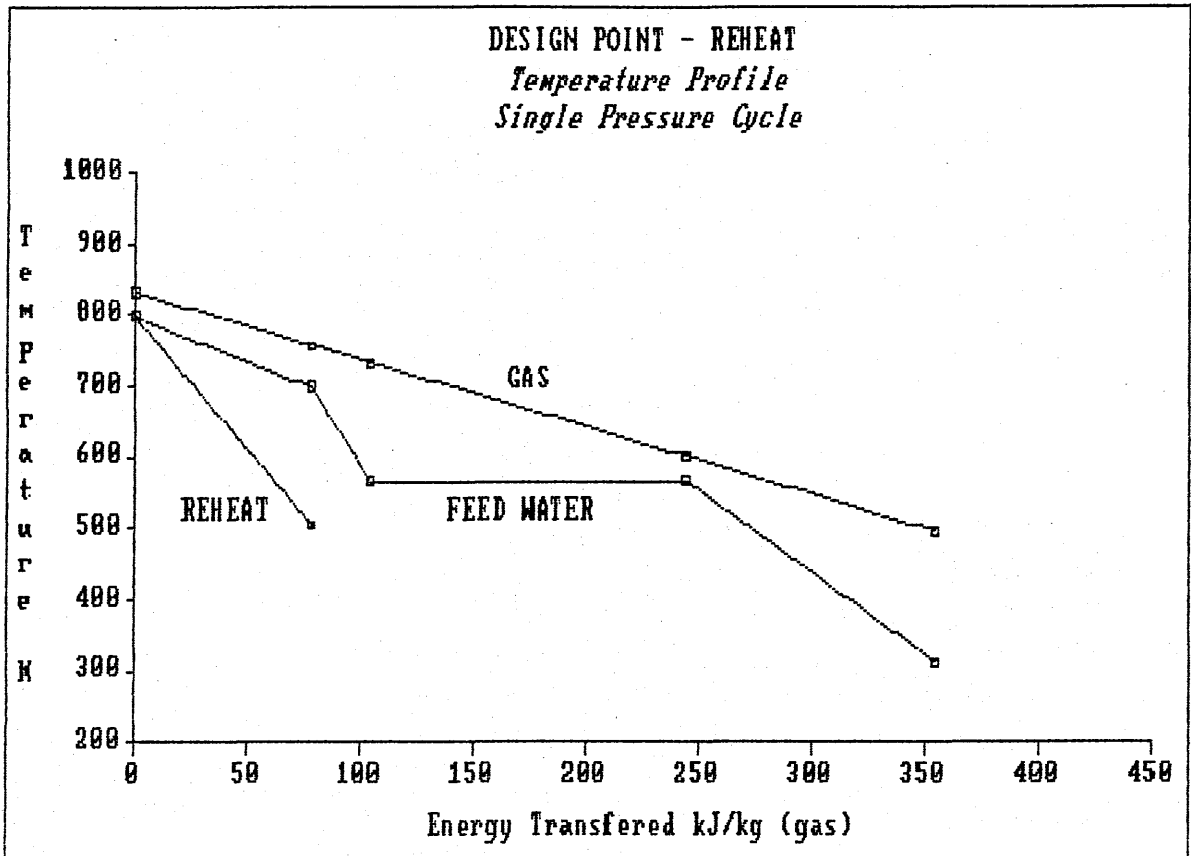


Figure 7.32

Reheat Steam Turbine HRSG Temperature Profile



These measures to reduce heat transfer losses introduce additional cost and complexity. As the temperature differences are reduced the entropy generation reduces but the necessary heat transfer surfaces increase. A single pressure HRSG may require 35–50 m²/MW whereas a dual pressure boiler may require more than 60 m²/MW. It was mentioned that about half the lost energy due to heat transfer temperature difference in a single pressure HRSG is recovered by a dual pressure HRSG, and it is expected that additional pressures or binary systems can further increase the efficiency by only approximately 1%. Often the cheaper simplicity is more beneficial than capital intensive performance, and single pressure heat recovery is widely used. Generally dual pressure steam turbines would tend to have higher pressures and be used for base load, while single and lower pressure cycles would be used for mid to peaking load ranges.

Reheat in the steam turbine is used for high pressure steam cycles to reduce the wetness of steam in the low pressure turbine stages. The steam is reheated after part of the expansion, and this doubles the flow through part of the superheater. This effect is shown in figure 7.32, and it can be seen the temperature profiles are improved. This means that steam turbine reheat can be advantageous for the heat recovery efficiency as well as for the steam turbine low pressure stage efficiency. A combined cycle with steam reheat may be say 170 bar, and give a further combined cycle efficiency gain over a two pressure non-reheat cycle of about 1%. Reheat is only normally viable for exhaust temperatures more than about 870 K and requires large unit sizes to

prevent HP steam blades from being too small. Although the efficiency gains are limited there are other advantages with steam reheat. The steam quality in the last stages of the turbine may increase from say 85% to 90%, with improved turbine efficiency and longer blade life. Also the overall heat transfer surface area is reduced. These benefits need to be traded off with the higher capital and maintenance costs, and the cycle efficiency is more sensitive to steam pressure.

7.7.4 Steam Cycle Regeneration

Regeneration, or feedwater heating is the process of using steam extracted from the turbine part way through the expansion to preheat the cold feedwater before entry into the economiser. On conventional steam cycles this increases efficiency by reducing temperature differences for heat transfer and thus reducing entropy generation. Regeneration increases the temperature of the cold end of the HRSG, which may be necessary to a small degree for low grades of sulphur bearing fuel but reduces heat recovery efficiency. Deaerators provide some regeneration as they use extracted steam for heating.

Regeneration degree has been defined as the ratio of the feedwater enthalpy rise due to regeneration, to the enthalpy difference between boiler saturation and the condenser feedwater pump outlet. On conventional steam plant the boilers air preheater means the gas stack temperature is independent of the degree of regeneration. This is not necessarily true of HRSG's, where simply increasing the temperature of the inlet water (T_1 figure 7.11) to the cold end of the boiler will increase the stack temperature (θ_4) and reduce the heat recovery efficiency.

The reference [184] analyses how regeneration effects the combined cycle performance. For high GT exhaust temperatures, regeneration is shown to improve the combined cycle efficiency for small regeneration degree, whereas with low exhaust temperatures regeneration is shown to reduce efficiency. The theory used was similar to the theory used in this thesis with the addition of the feedheaters and extraction points.

The example of reference [184] has a gas turbine TET of 1273 K, and with a pressure ratio below 6.5 regeneration was advantageous while above this regeneration was disadvantageous. The change over exhaust temperature at pressure ratio of 6.5 corresponds to where the steam temperature has reached it's material maximum of 810 K. The efficiency gain is small, for this example the improvement in combined cycle efficiency was about 1% (45% to 46%). This was using three feedheaters and a regeneration degree of 0.2. Improvements will be more for gas turbines with higher exhaust temperatures.

Assuming the GT exhaust temperature is sufficient for regeneration to improve efficiency, due to low GT pressure ratio, high GT TET or with supplementary firing, the main conclusions were;

- 1) The number of extractions raises the efficiency, but the efficiency gain becomes smaller as the number of extractions increases. It is suggested efficiency gains are minimal after three extractions.
- 2) Equally spaced regeneration temperatures are nearly optimal
- 3) When efficiency is improved then so is specific work.

- 4) Whatever the gas turbine temperatures the efficiency only improves for small degrees of regeneration ($R < \text{approximately } 0.2 - 0.3$)
- 5) The higher the GT exhaust temperature the less sensitive the degree of regeneration has on the efficiency.

7.7.5 Intercooling

Performance changes for intercooling a gas turbines with exhaust heat recovery is not straight forward. The first law only tells us the energy balance and that the compression work is reduced by intercooling. It does not tell us that heat rejected to the intercooler is at a high temperature and has a recoverable fraction which may have been recouped in the bottoming cycle. In cycles where heat rejection is constrained to ambient pressure regardless of temperature, as in a simple cycle GT, intercooling increases efficiency. In cycles where the steam cycle rejected heat is at lower temperatures than the intercooling water the efficiency may or may not increase.

Combined cycles using intercooling that disposes of the cooling water always produces a decrease in efficiency if the gas turbine pressure ratio is below a certain breakeven level. For pressure ratios above this breakeven, intercooling may or may not produce an efficiency increase depending on the number of intercoolers, their pressure drop, reheat configuration and heat recovery configuration. This can be followed more fully in [191].

7.8 ESTIMATING THE BOTTOMING EFFICIENCY

An established rule of thumb is that the steam bottoming cycle gives 50% more power to a simple cycle gas turbine, therefore the combined cycle efficiency is about 1.5 time the gas turbine efficiency. This is very approximate, as the previous analysis has shown us that the bottoming cycle efficiency is highly dependant on the exhaust temperature. A simple and better method is presented here. It is still approximate but a next order more accurate than the 50% rule.

An analysis of dual pressure bottoming cycles can be found in reference [193]. In this reference the optimum bottoming cycle efficiency was calculated as a function of the exhaust gas temperature considering single pressure reheat and dual pressure cycles. The results are plotted in figure 7.33 for steam temperature limits of 811 and 867 K. These curves combine when the exhaust gas temperature is insufficient to supply this steam temperature. The curves are from the equations in [193] as follows;

$$\begin{aligned} &\text{for maximum steam temperature of 811 K} \\ &\eta_{BC} = -0.7285 + 0.5855 \Gamma - 0.08081 \Gamma^2 \quad \dots 7.17) \end{aligned}$$

$$\begin{aligned} &\text{for maximum steam temperature of 867 K} \\ &\eta_{BC} = -0.6327 + 0.5094 \Gamma - 0.06578 \Gamma^2 \quad \dots 7.18) \end{aligned}$$

$$\text{where } \Gamma = T_{\text{EXHAUST}}/T_{\text{AMBIENT}} \text{ and } 2.337 < \Gamma < 3.652 \quad \dots 7.19)$$

The sensitivity curves for the analysis can be found in reference [193].

Figure 7.33 Bottoming Cycle Efficiency vs Exhaust Temperature

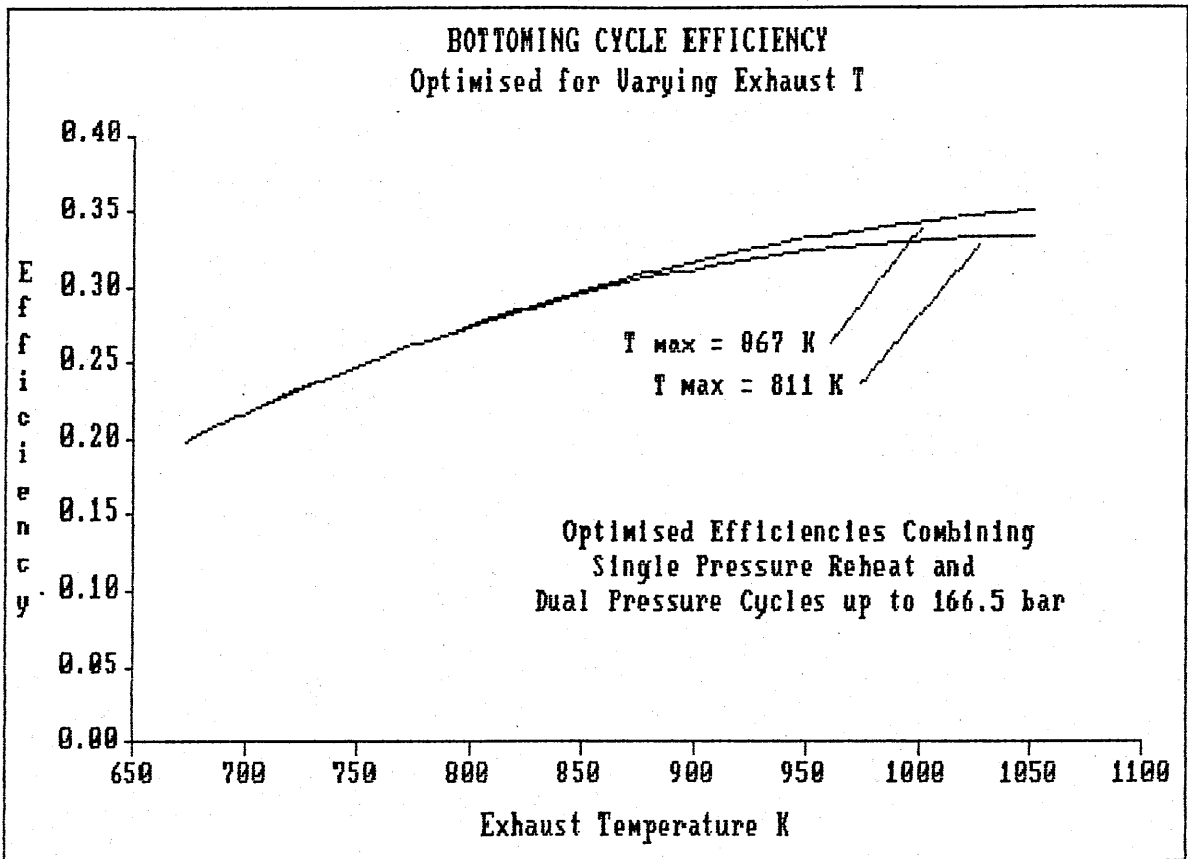


Table 7.3 Comparison of Quoted Efficiency and Calculated Optimal

Quoted information						Calculated, estimation method						50% method	
Engine Type	η_{GT}	Power GT	T_{EXH}	η_{CC}	Power GT in a CC	Power loss of GT	New η_{GT} in CC	T_{EXH}	η_{BC}	η_{CC}	Diff. from quoted	η_{GT} x1.5	Diff. from quoted
Dual Pressure Steam Cycle													
GT8	0.299	46.90	523	0.475	45.30	3.41%	0.294	796	0.272	0.486	2.3%	0.449	-5.9%
GT11N	0.306	81.60	515	0.485	78.10	4.29%	0.299	788	0.268	0.487	0.5%	0.459	-5.7%
GT13	0.305	98.20	489	0.481	94.00	4.28%	0.298	762	0.255	0.477	-0.8%	0.458	-5.1%
GT13E	0.328	147.90	516	0.506	142.10	3.92%	0.322	789	0.269	0.504	-0.4%	0.492	-2.8%
V94	0.337	149.25	545	0.511	140.00	6.20%	0.327	818	0.282	0.517	1.1%	0.506	-1.1%
V84	0.336	102.53	539	0.509	95.00	7.34%	0.324	812	0.280	0.513	0.8%	0.504	-1.0%
501F	0.337	145.00	572	0.502	136.20	6.07%	0.327	845	0.294	0.524	4.3%	0.506	0.7%
V84.2	0.331	103.00	540	0.509	100.00	2.91%	0.326	813	0.280	0.515	1.2%	0.497	-2.5%
FT8	0.381	25.40	450	0.505	24.80	2.36%	0.377	723	0.232	0.521	3.1%	0.572	11.6%
Single Pressure Steam Cycle													
5371PA	0.289	26.30	483	0.419	25.90	1.52%	0.287	756	0.203	0.432	2.9%	0.434	3.3%
6541B	0.314	38.34	539	0.451	37.70	1.67%	0.311	812	0.229	0.469	3.9%	0.471	4.2%
9161E	0.331	116.90	529	0.478	115.30	1.37%	0.329	802	0.225	0.480	0.3%	0.497	3.7%
MW-501	0.333	104.57	521	0.470	98.70	5.61%	0.324	794	0.221	0.473	0.6%	0.500	5.9%
MW-701	0.339	130.55	513	0.462	119.60	8.39%	0.325	786	0.217	0.471	2.0%	0.509	9.1%

Using the data from reference [193] the author has added an approximate equation for a single pressure bottoming cycle;

$$\eta_{BC} = 0.000467 T - 0.15 \quad \text{where } 750 < T < 975 \text{ K} \quad \dots 7.20)$$

The above optimised bottoming cycle efficiency can be compared to the manufacturers data. An example is as follows for the ABB GT8;

quoted simple cycle performance (appendix D)

$$\eta_{GT} = 0.299, \text{ power} = 46.9 \text{ MW}, \text{ TEXHAUST} = 523 \text{ }^{\circ}\text{C}$$

quoted combined cycle performance (appendix D)

$$\text{power GT} = 45.3 \text{ MW}, \text{ power ST} = 24.8 \text{ MW}, \text{ nCC} = 0.478$$

In the combined cycle the GT power is reduced due to the greater exhaust pressure loss over the HRSG. The combined cycle overall efficiency is given but not the GT efficiency. Using the rule of thumb that the percentage efficiency loss due to inlet/outlet pressure drop is half that of the power loss, the combined cycle gas turbine efficiency can then be guessed. In this case the power reduction from a simple cycle gas turbine of 46.9 MW to combined cycle gas turbine of 45.3 MW is a 3.41% loss, which gives an estimated GT efficiency loss of $3.41/2 = 1.7\%$ or 0.294 efficiency. The change in exhaust temperature is unknown but small. Using equation 7.17 the estimated optimal bottoming efficiency is 0.272. Equation 7.6 can be written as;

$$\eta_{CC} = \eta_{GT} + (1 - \eta_{GT})\eta_{BC} \quad \dots 7.21)$$

This gives an optimal combined cycle efficiency for the quoted GT8 of 0.486, which is 2.3% higher than the 0.478 quoted. This is because the actual steam cycle of the quoted GT8 is unlikely to be the same as the optimum one assumed in equation 7.17, and the GT8 is a small engine with efficiency penalties of small scale. Using the rule of thumb that combined cycle efficiency is 1.5 times the simple cycle we get a combined cycle efficiency of 0.449 which is 5.9% low. Other engines are compared in a similar fashion in table 7.3. It can be seen that the above method gives a slightly higher efficiency on most of the examples, and gives an approximate and simple method of calculating the potential maximum combined cycle efficiency from a given gas turbine. The 50% rule of thumb is consistently less accurate and tends to under estimate dual pressure industrial engines, and over estimate single pressure industrials. The 50% rule of thumb is especially inaccurate for the aero-derivatives with high pressure ratio and low exhaust temperature.

A summary of the method of estimation of the upper bounds of a practical optimal combined cycle efficiency given the gas turbine efficiency and exhaust temperature is as follows.

- 1) Adjust the performance of the GT for the increased pressure loss of the HRSG. Typically a 1.5 to 2 kPa loss over the HRSG and 0.5 kPa over the inlet. The reduction in performance is approximately 1/2% efficiency per kPa exhaust loss (1% power per kPa) and 1% loss of efficiency per kPa inlet loss (2% power).
- 2) Use the exhaust temperature and equations 7.17 or 7.18 to estimate the optimal bottoming cycle efficiency
- 3) Use the equation 7.21 to find the estimated optimal combined cycle efficiency

Values will be quoted on a particular fuel and fuel the type effects efficiency. The combined cycle should be de-rated for non natural gas fuels, with example factors as follows, based on the GE STAG 207EA;

	Power	Efficiency
Natural Gas	Ref.	Ref.
Distillate	-3.0%	-2.1%
Residual	-9.3%	-7.6%

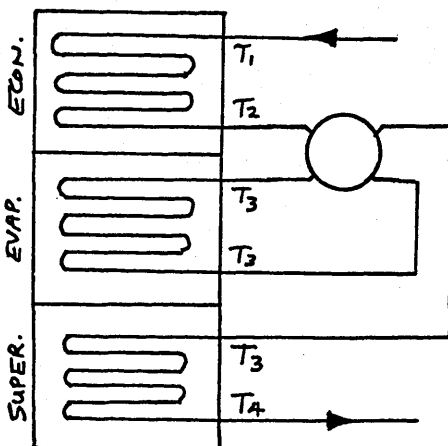
7.9 OFF DESIGN HRSG

The off design of the combined cycle involves the off-design of the three main components, the gas turbine, HRSG and the steam turbine. The gas turbine off design is handled by the Cranfield program TURBOMATCH as discussed in section 3.4.5. The key to the off design of the bottoming cycle is the off design of the HRSG.

Ideally the HRSG heat exchangers are sized so that at the design point conditions the economiser delivers the feed water to the steam drum at it's boiling temperature. In reality at part load the tendency is for the economiser to over heat the feed water and have some evaporation in the economiser before the flow reaches the steam drum. Usually the economiser is designed so that at design point the economiser only heats to some margin below boiling so that no evaporation occurs in the economiser and some feed heating occurs in the evaporator. This keeps boiling out of the economiser at least over most of the normal operating range. The heat exchange characteristics with boiling in the economiser or feed heating in the evaporator are different. Provided no evaporation occurs in the economiser the feed water is mixed into the steam drum, the temperatures in the evaporator remain constant, and the areas are constant. If evaporation occurs in the economiser then the area of the evaporator is effectively increased and the area of the economiser effectively decreased. This is shown in figure 7.34.

Figure 7.34 Part Load Effective Areas in HRSG

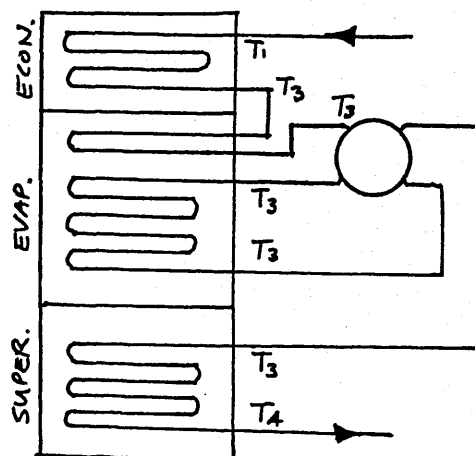
FEEDHEATING IN EVAPORATOR



$$A_{ECON} = A'_{ECON}$$

$$A_{EVAP} = A'_{EVAP}$$

EVAPORATION IN ECONOMISER



$$A_{ECON} = A'_{ECON} (1-i)$$

$$A_{EVAP} = A'_{EVAP} + A'_{ECON} i$$

7.9.1 Off Design HRSG Theory

The off design of the HRSG was carried out using a model developed on a spreadsheet using the common heat exchanger equations and applicable heat transfer correlations. The heat transfer correlations used were as follows;

External surface of a tube bank [206],

$$\text{Schmidt equation} \quad Nu_D = C(Re_D)^m(Pr)^n\phi^p \quad \dots 7.22)$$

where Nu , Re and Pr are the Nusselt, Reynolds and Prandtl number respectively. C , m , n and p are constants, and ϕ is the ratio of the finned area to that of a plane tube.

$C = 0.3$ for aligned tubes and 0.45 for staggered
 $m = 0.625$, $n = 0.333$, $p = 0.375$

Internal surface of a pipe [204],

$$\text{Dittus-Boelter equation} \quad Nu_D = 0.023(Re_D)^{4/5}(Pr)^n \quad \dots 7.23)$$

where n is 0.4 for heating and 0.3 for cooling, and the equation is valid for $0.7 < Pr < 160$, $Re > 10000$.

It is usual to express performance of cross-counterflow heat exchangers in terms of pure counter flow exchangers by introducing a correction factor, or efficiency. The correction is close to unity in an exhaust HRSG in it's normal range of operation [206]. This is usually assumed in boiler design generally. This means the heat exchanger can be assumed to behave like a pure counter flow exchanger.

The economiser and superheater are forms of heat exchanger with varying temperature on both the fluid sides. The governing equation is easily derived or obtained from a reference text such as [204];

$$\begin{aligned} \text{counter flow HEX} \\ \text{varying temperature both fluids} \quad \epsilon = \frac{1 - \exp[(-(1-E)/E) NTU]}{1 - (1/E)\exp[(-(1-E)/E) NTU]} \quad \dots 7.24) \end{aligned}$$

where E is the ratio of flow heat capacities, equation 7.11,
 and NTU is the Number of Transfer Units or,

$$NTU = (hA)/(\dot{m}_{HOT}C_{HOT}) \quad \dots 7.25)$$

where h , A , and C_{HOT} are the heat transfer coefficient, area of heat exchange, and heat capacity of the hotter fluid respectively.

For a heat exchanger with one fluid of constant temperature as in the evaporator the governing equation collapses into the following form;

$$\text{cold fluid constant temperature} \quad \epsilon = 1 - \exp[- NTU] \quad \dots 7.26)$$

The heat transfer coefficient is derived from the coefficients of the heat transfer path, figure 7.35. The overall coefficient is obtained as follows;

$$1/h_{TOT} = \sum (1/h)_i \quad \dots 7.27)$$

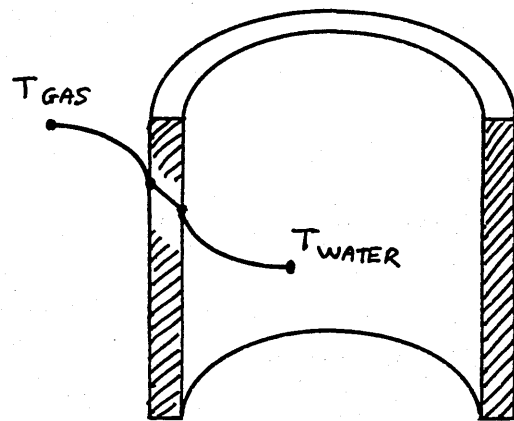
where h_1 is the heat transfer coefficients of the following;

- 1) convection from gas to pipe
- 2) surface fouling of outside surface
- 3) pipe wall conduction
- 4) surface fouling of the inside surface
- 5) convection from pipe to water or steam

h for boiling is very large, and thus $1/h$ is negligible.

Figure 7.35 Heat Transfer Resistances

To calculate results some assumptions had to be made. The tubes were assumed to be aligned (figure 7.9) and of dimensions in figure 7.36. This gives a fin area factor ϕ on the outside of 2.5 and a fin efficiency of approximately 0.8 (reference 205 page 621). The minimum duct area was assumed to be 5.0 m^2 for the gas turbine in section 3.4, which has 153 kg/s of exhaust flow at design point. The number of tubes in parallel was chosen to give realistic Reynolds numbers. It was satisfactory to guess the geometry of the HRSG as the trends of the off-design behaviour were more important than the numerical results. The fin pitch chosen was later found to be too large but the effect on the overall results is minimal and did not warrant recalculation. In real cases the geometry would be known and would differ for each HRSG. Calculations can be seen in appendix A.



HEAT TRANSFER RESISTANCES

- $1/h$ GAS CONVECTION
- $1/h$ FOULING EXTERNAL
- $1/h$ WALL CONDUCTION
- $1/h$ FOULING INTERNAL
- $1/h$ STEAM/WATER CONVECTION

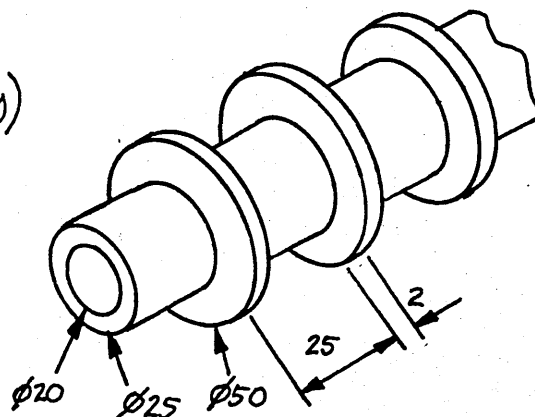
Figure 7.36 Assumed Fin and Tube Dimensions

$$\eta_{FIN} = fn(h, k, \text{geometry})$$

$$\approx 0.85$$

$$\phi_a = \frac{AREA_{ACTUAL}}{AREA_{PLAIN TUBE}}$$

$$= 2.5$$



The off-design calculations were handled by using the heat exchanger correlation in equation 7.22, substituting the definitions of Re, Nu and Pr and rearranging it in the form;

$$h = \Gamma \beta (\dot{m})^m \quad \dots 7.28)$$

where Γ is a number representing all the geometric factors and β represents all the gas property factors.
 m is the constant in equation 7.22.

The resulting equation 7.28 allows the variation in h to be related directly to mass flow. Γ is a geometric property and apart from variations in fin efficiency does not change at part load. Fin efficiency changes are small and ignored. The only other variable apart from mass flow is the property constant β , which varies weakly with fluid temperature but for the purposes of this thesis is assumed constant. This is only approximately correct but will introduce only small errors of smaller magnitude than the accuracy of the correlations. This method also allows comparison of different HRSG designs by changing only the number Γ .

The off design model in appendix A uses the design point calculation of the combined cycle performance to calculate the areas and the product $\Gamma\beta$. The product $\Gamma\beta$ is used in the off-design calculations to recalculate the heat transfer coefficient for different mass flows. The new heat transfer coefficients are combined using equation 7.27 and include fouling heat transfer resistances. The overall heat transfer coefficient and the areas are used to calculate the NTU, which in the heat exchanger equations 7.24 and 7.26 gives the temperature changes. If the temperatures indicate boiling in the economiser the areas are adjusted by reducing the economiser and increasing the evaporator areas until temperatures match (figure 7.34).

The method is summarised as follows;

$$\begin{aligned} h_1 & \text{ from eq. 7.28} & h &= \Gamma\beta(\dot{m})^m \text{ for each surface} \\ h_{TOT} & \text{ from eq. 7.27} & 1/h_{TOT} &= \sum 1/h_i \\ A_{ECON} & \text{ from} & A_{ECON} &= A'_{ECON}(1-i) & \dots 7.29) \\ A_{EVAP} & \text{ from} & A_{EVAP} &= A'_{EVAP} + A'_{ECON}(i) & \dots 7.30) \end{aligned}$$

where i is the fraction of the economiser area used in evaporation
 if $T_2 > T_3$ then i increased till $T_2 = T_3$
 if $T_2 < T_3$ then $i = 0$

$$\begin{aligned} NTU & \text{ from eq. 7.25} & NTU &= h_{TOT}A/(\dot{m}_G C_{pG}) \\ E & \text{ from eq. 7.11} & E &= \dot{m}_G C_{pG} / \dot{m}_G C_{pG} \\ \epsilon & \text{ from} & \epsilon &= fn(NTU \text{ and } E), \text{ either eq. 7.24 or 7.26} \end{aligned}$$

temperatures from the definition of ϵ equation 7.12.

The process is iterative, with the mass flow of steam and area of evaporation in the economiser guessed. The temperatures are calculated, then checked for energy balance. A new amount of feed heating in the evaporator and new steam flow is estimated. If the feed heating in the evaporator is negative the area of the economiser is reduced and the area in the evaporator increased correspondingly.

7.9.2 Off Design HRSG Results

The mathematical model of the HRSG produces the trends expected. It is based on the method of reference [206] and so has some experiential background, but is extended from this work to include a superheater, fouling possibilities and ability to vary the correlations used. To study the off-design performance the input temperature and mass flow of the hot gas were varied separately. The results are plotted in figures 7.37 and 7.38. The HRSG was assumed clean with zero fouling losses.

The study results show that the performance of the heat recovery falls rapidly with falling exhaust temperature, until the gas turbine exhaust temperature reaches the boiling temperature in the steam drum where no more steam can be raised. The effect of changing the exhaust mass flow is to change the steam mass flow in direct proportion. Therefore a drop in exhaust mass flow of 10% reduces the steam mass flow by 10% and leaves the maximum steam temperature nearly constant. This leads to the approximate combination of the two graphs into one performance curve with the ratio of the mass flows μ plotted against exhaust temperature. This is shown in figure 7.39. The single curve shows the performance of the HRSG for one steam drum pressure only, and thus there would be a family of curves for different steam pressures in the same HRSG.

The reducing feed heating in the evaporator (approach point), and eventual evaporation in the economiser at part load is shown in figure 7.40. Here it is seen the feed heating in the evaporator reduces to zero, and then evaporation occurs in the economiser. The pinch points also reduce with part load.

The effect of fouling was calculated by varying the degree of fouling heat transfer coefficients from zero up to 100% of some typical values. The values are from reference [204] and are given in table 7.4.

Table 7.4 Representative Fouling Factors

Fouling Factor	1/h ($\text{m}^2\text{K/W}$)
External air to metal	0.0002
Steam (non-oil bearing)	0.0001
Treated boiler feedwater ($>50^\circ\text{C}$)	0.0002

Effects of increased pressure loss due to fouling were not included. The results are shown in figure 7.41. It can be seen that the reductions in steam turbine power and bottoming efficiency for dirty boilers are the same and in the order of 5% for representative fouling from reference [204]. In reality even greater reductions in performance are possible, up to 25% was reported to the author from one plant with natural gas engines. It should also be noted that the HRSG may not be available for cleaning for extended periods on a base-load plant.

Comparative performance of several HRSG's must be viewed in light of the aims and assumptions made. Performance is effected by design, and compromises such as the size of the approach point to improve part load. Also properties may be assumed, for instance the specific heat of 'flue' gas can vary by 6 to 8 % depending on the moisture content in the gas, which is dictated by fuel type and inlet humidity.

Figure 7.37 HRSG Off Design - Varying GT Exhaust Temperature

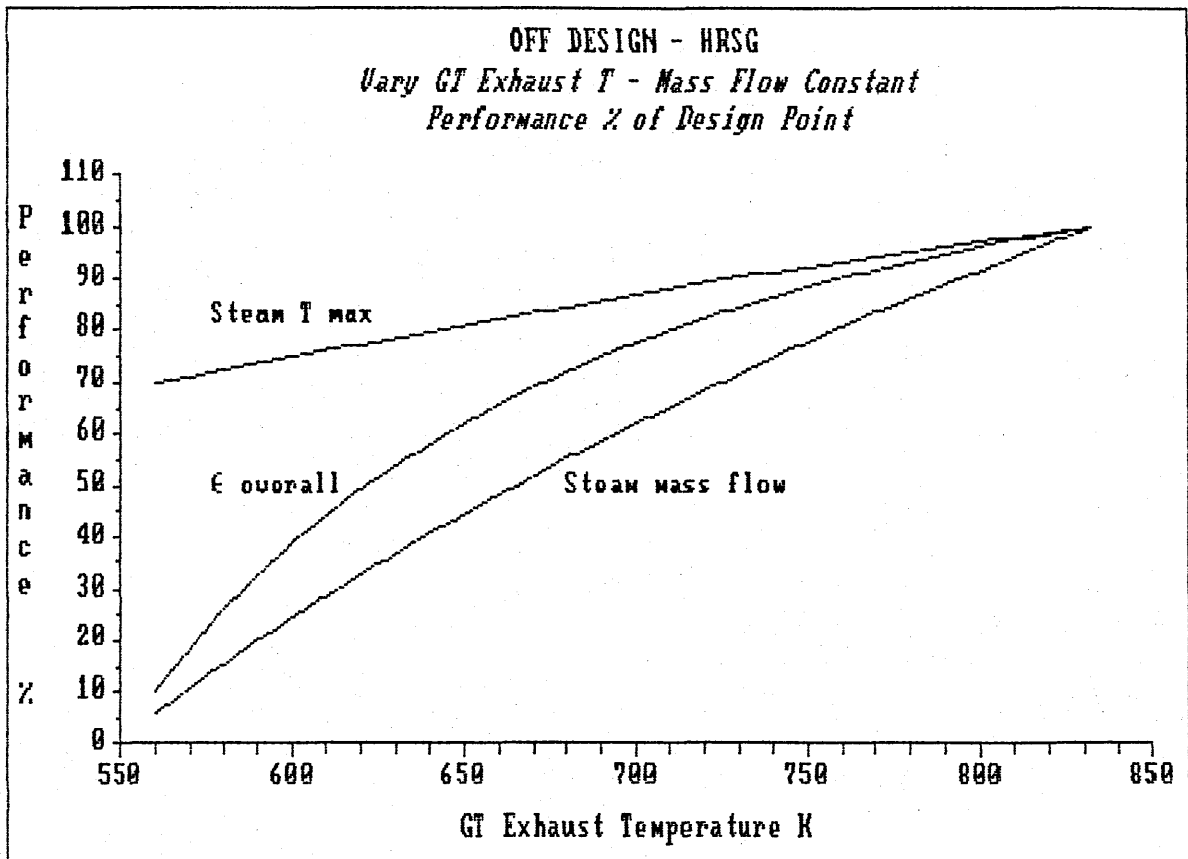


Figure 7.38 HRSG Off Design - Varying GT Exhaust Mass Flow

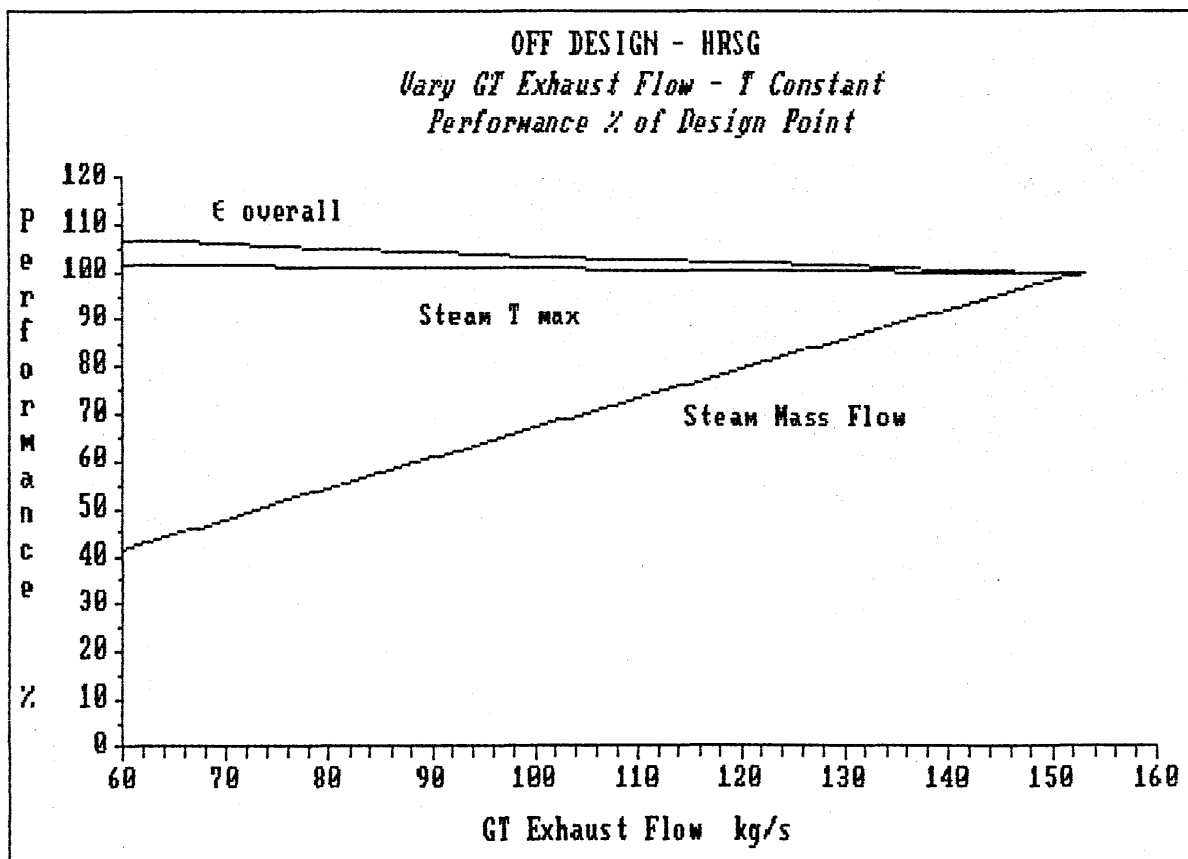


Figure 7.39 HRSG Off Design - Performance Characteristics

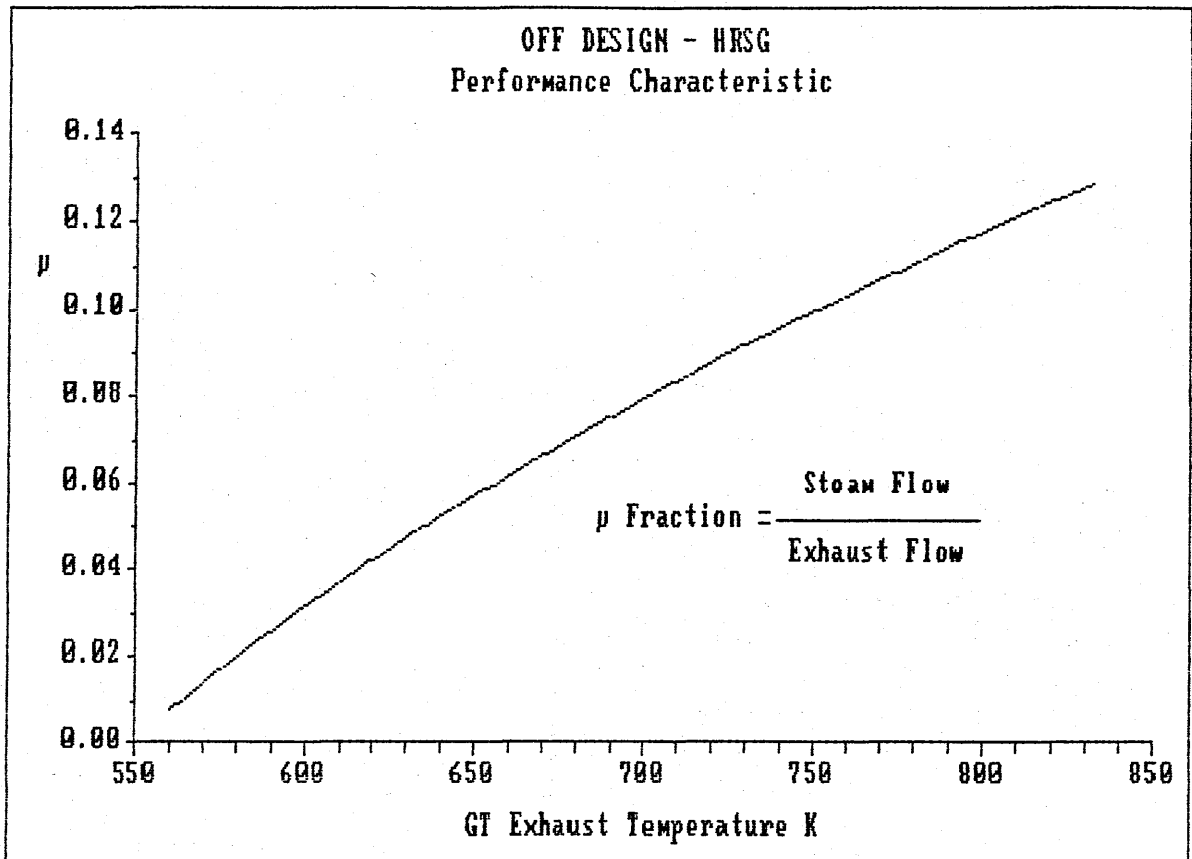


Figure 7.40 HRSG Off Design - Internal Characteristics

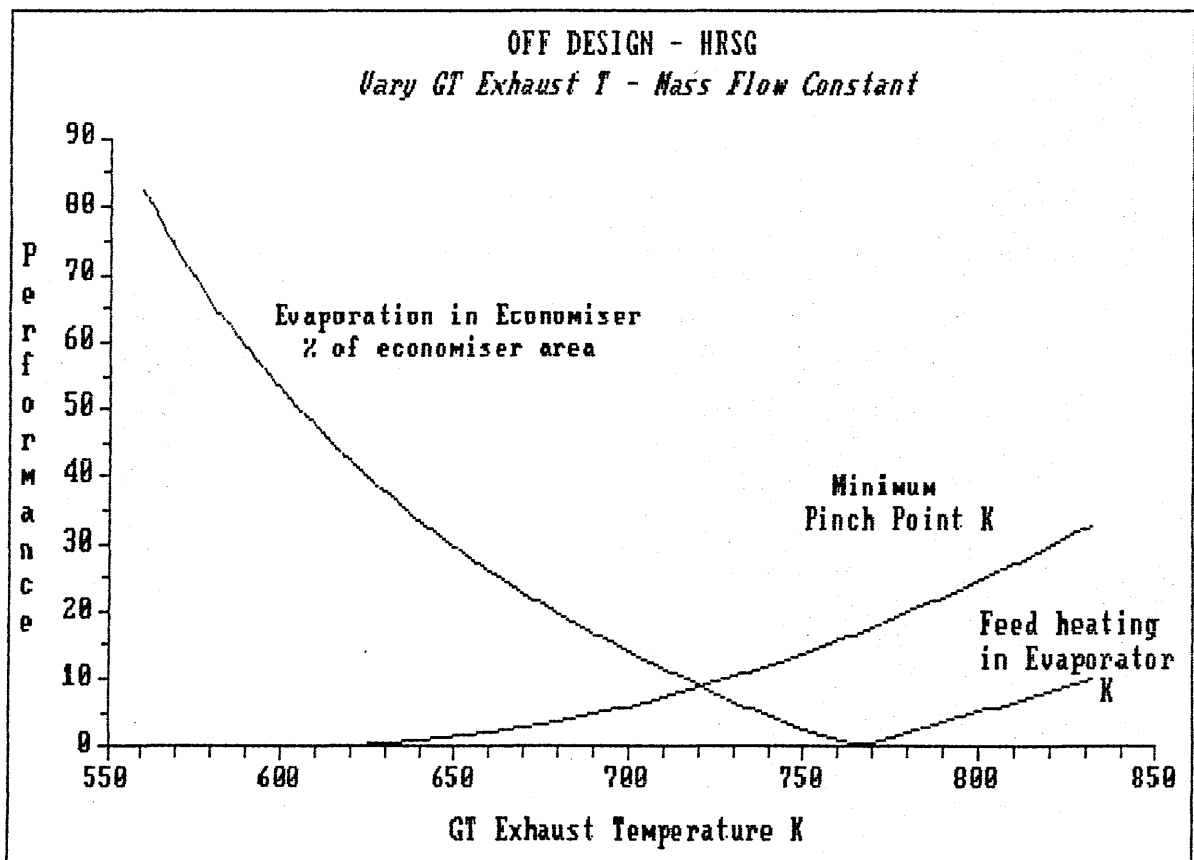
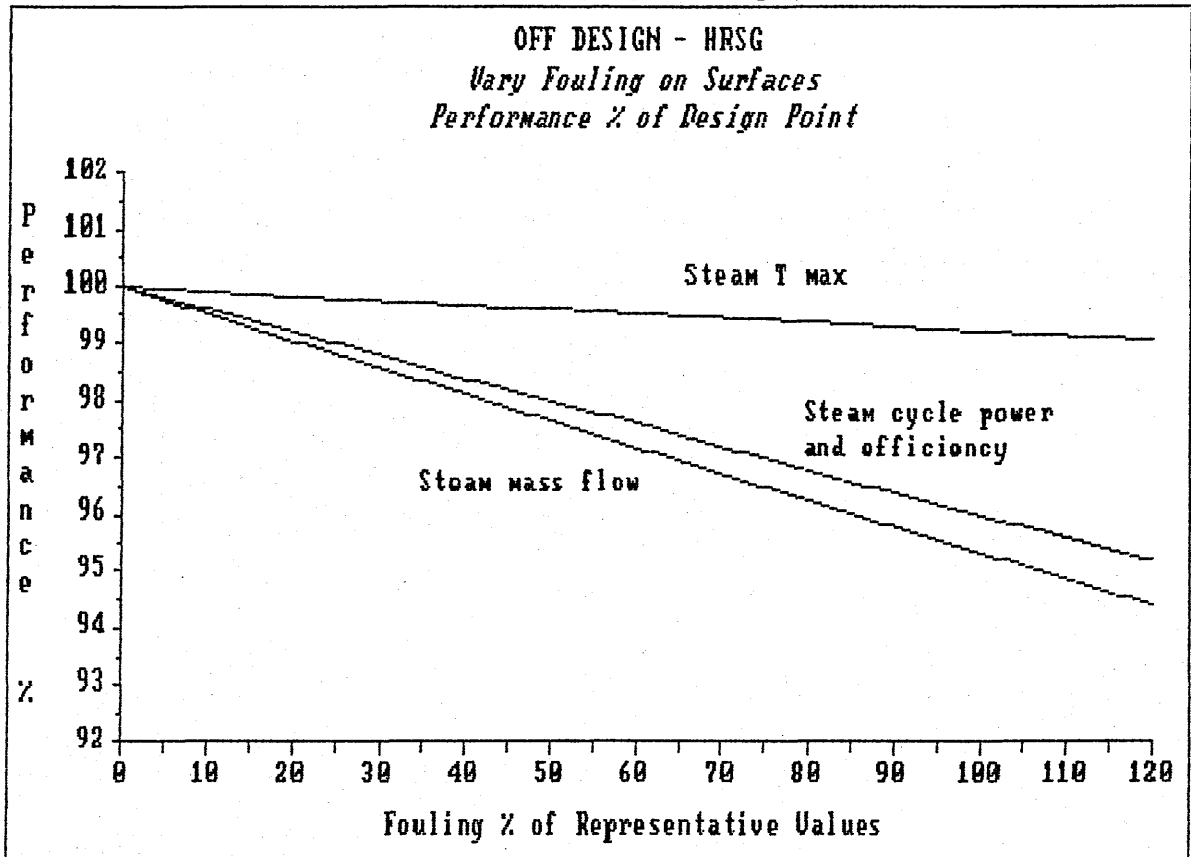


Figure 7.41 HRSG Off Design - Varying Fouling



7.10 PART LOAD COMBINED CYCLE

The part load efficiencies of the combined cycle require the part load efficiencies of the steam turbine which are as discussed in section 6.4. The corrections used are old [161] and are expected to be pessimistic as new blade shapes and profiles should have improved part load behaviour. They are used in spite of the possible improvements as they have real trends in a usable form, and in this case the trends are more important than the numerical results.

The off design combined cycle calculations were carried out on the spreadsheet model in appendix A. The part load behaviour of the gas turbine was calculated as in section 3.4.5, and the part load behaviour of the HRSG was calculated as in the last section. The steam turbine part load efficiency corrections were used from section 6.4. The results are plotted in figures 7.42 and 7.43. These figures are comparable to figure 3.25 for the gas turbine. In this case using a single shaft GT the rapid lowering exhaust temperature of the gas turbine reduces below the steam drum boiling point before the gas turbine is idling, thus the steam turbine work output falls to zero first.

Figure 7.42 CC Part Load - Power vs GT TET, Constant Steam Pressure

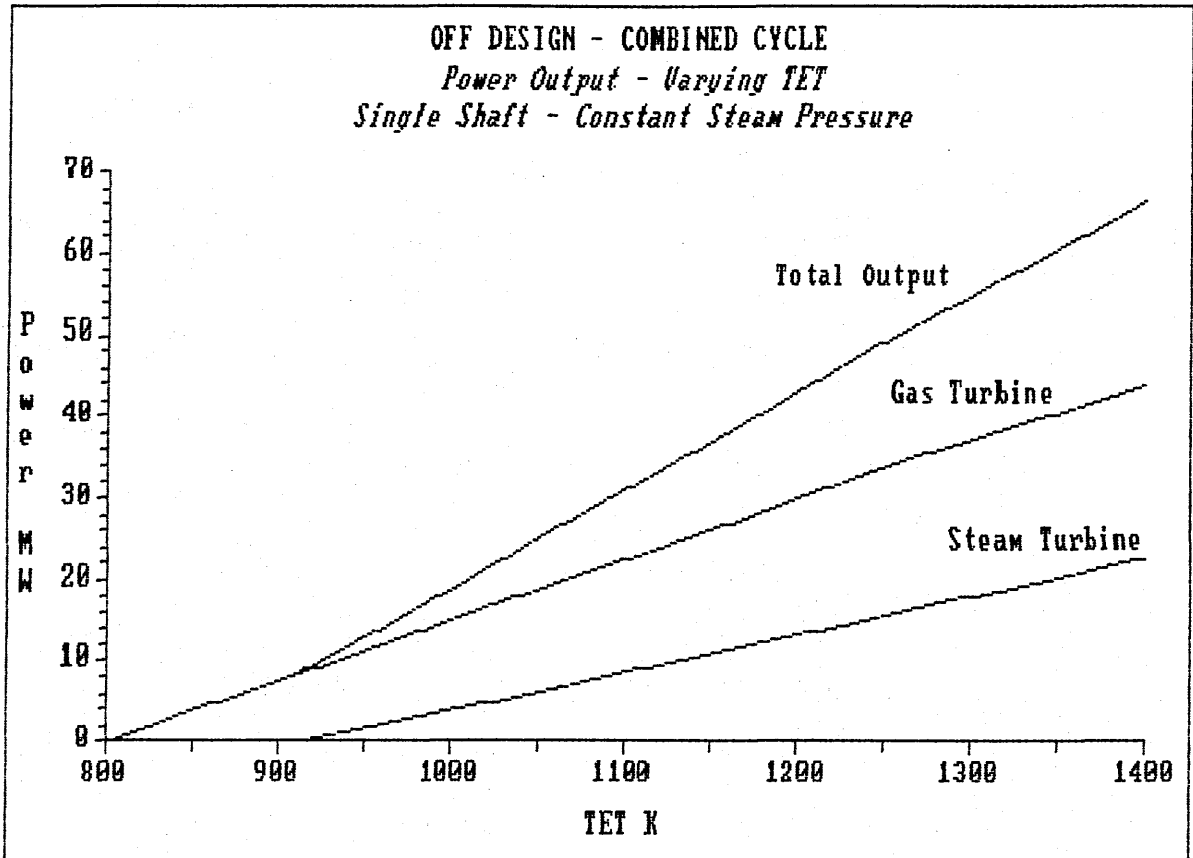
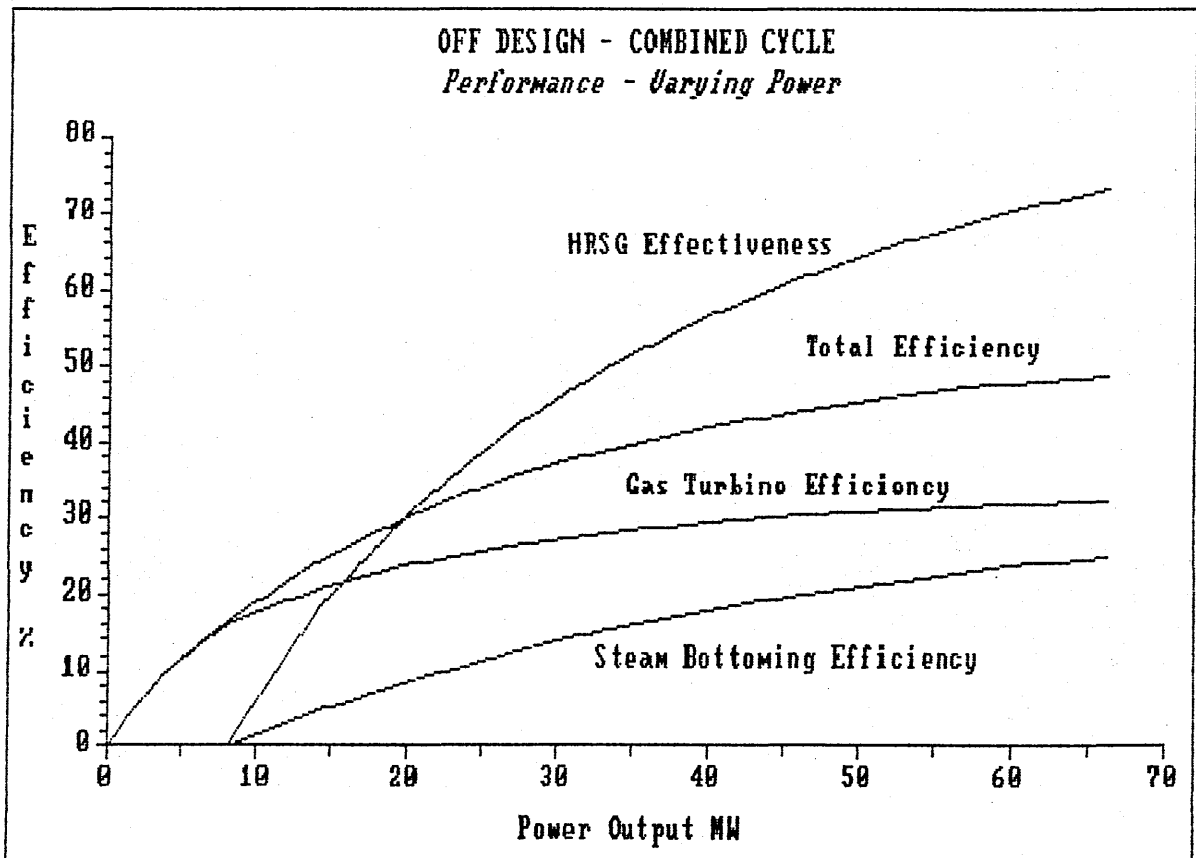
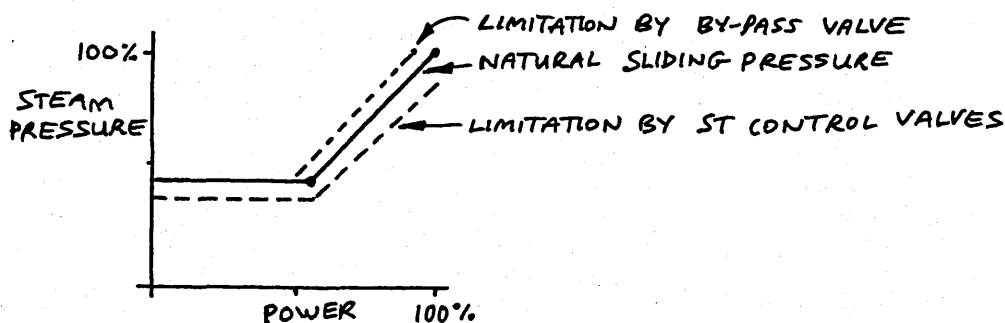


Figure 7.43 CC Part Load - Efficiency vs Power, Constant Steam Pressure



To combat the rapid fall in heat recovery efficiency as the exhaust temperature drops, and approaches the boiling point in the steam drum, the pressure in the steam drum can be scheduled to lower with the exhaust temperature. This is done in practice, and is called 'sliding pressure'. A typical pressure schedule is shown in figure 7.44, reference [248]. Using this schedule the off design of the combined cycle was recalculated and the results plotted in figure 7.45. It can be seen the improvement in part load performance of the bottoming cycle is dramatic. Sliding pressure is controlled by feed pressure in response to steam generator load, and the throttle valve only intervenes to limit transients or sudden pressure drops. It also causes least stress on the steam turbine and reduces throttling losses.

Figure 7.44 Typical Sliding Pressure Schedule



The analysis so far has been for a single shaft gas turbine. The different configurations of the gas turbine, single or two shaft, and single or two spool, were shown in section 3.4.5 to have different effects on the part load exhaust temperature and flow. These were seen in figures 3.29 and 3.30. The two shaft engine has a rapidly reducing mass flow at low load but maintains its exhaust temperature better than a single shaft engine. These configurations were also modelled as combined cycles and the results shown in figure 3.46. It can be seen the two shaft engines have a much improved part load performance. This is because the improved GT exhaust temperature at part load is more important than the reduced mass flow. The higher temperature maintains a better boiler temperature profile at low load so gives improved heat recovery and also has an improved steam exhaust quality due to more superheat. This increases the steam turbine low pressure stage efficiency and life at part load operation.

In practice the gas turbines IGV's are also closed at low load to reduce the mass flow and further increase the exhaust temperature, improving part load efficiency. The benefits of IGV control may be as high as 30% more steam flow [148], and a drop of part load efficiency of only 1% over the range of 100 to 80% load. Some two shaft gas turbines also have variable nozzle guide vanes on the outlet of the gas generator, such as the MS5002B. These are used to allow part load control with constant compressor speed, varying the power by varying the mass flow. This is unlike the above mathematical example which controlled power by varying the TET.

Multi-engine installations allow successive shut down of gas turbines for reducing load to regain near design point efficiency. The main loss in efficiency would then be due to part load effects on the steam turbine.

Figure 7.45 Bottoming Cycle Efficiency, Sliding and Nonsliding Pressure

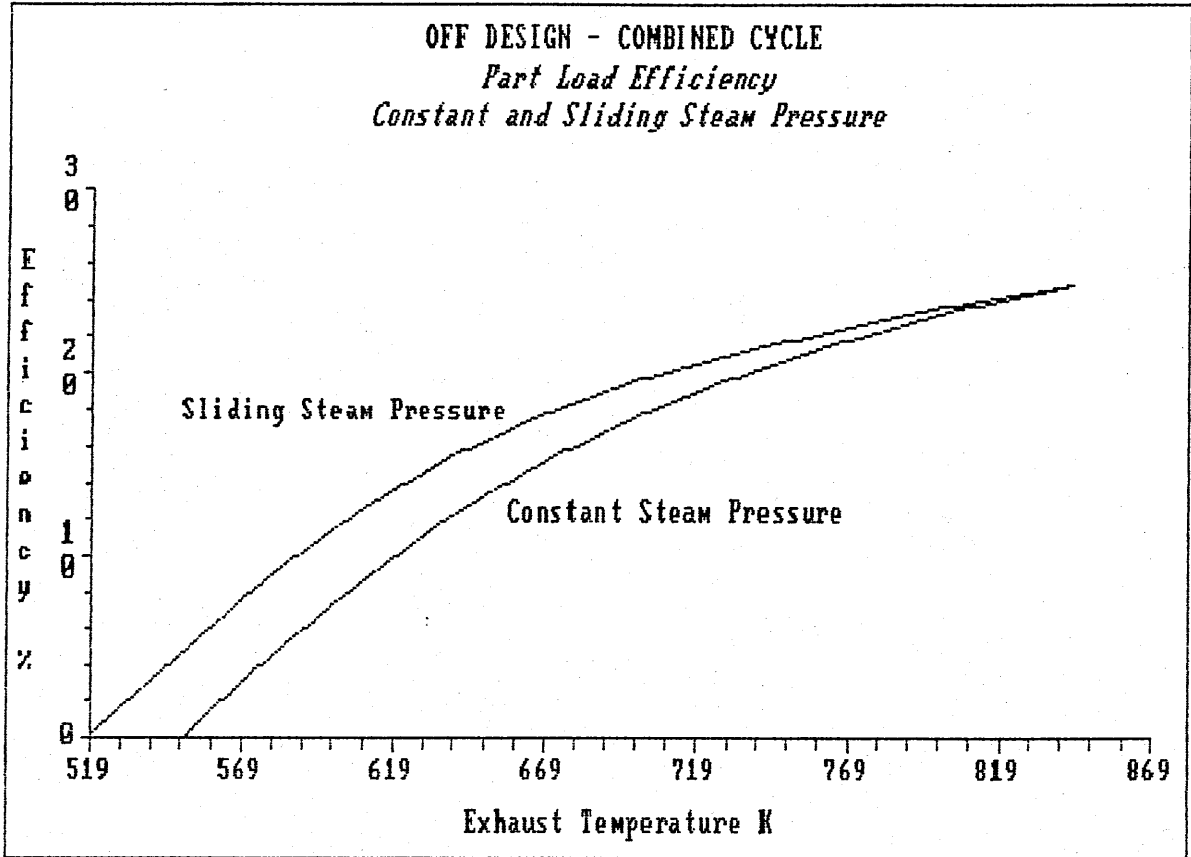
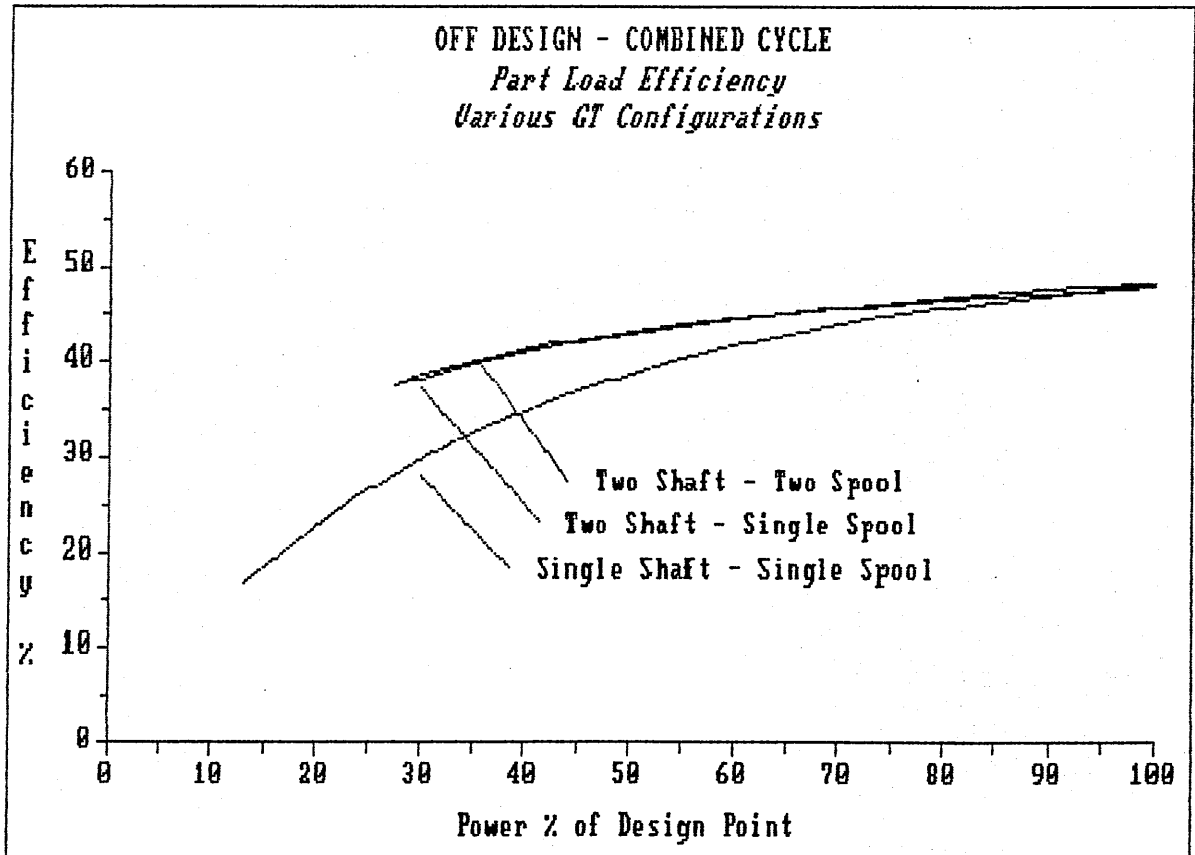


Figure 7.46 CC Part Load - Efficiency, Various GT Configurations



7.11 TRANSIENTS AND STARTING

Response times of combined cycles are longer than in gas turbines. The control system should be able to regulate power using the gas turbine and the steam turbine generates the complementary power corresponding to the steam conditions. The gas turbine is then the first to respond followed by the steam turbine after the HRSG response time. Typical large load change rates are in the order of 6-8% per minute in the 50-100% load range [248].

Start-up times are from the order of 40 minutes after an overnight shutdown to say 120 minutes from cold. If a bypass stack is provided for simple cycle operation, the gas turbine can be started and brought up to temperature while exhausting through the bypass stack. This gives about 2/3 of the power in GT starting times. Then the damper can be switched over in stages to progressively warm the boiler to avoid undue stress. The condenser is usually designed to take the full steam load, and allows bypass of the steam turbine during start up or after a plant trip. The steam turbine is brought up to speed after sufficient steam conditions have been reached. The steam turbine starting time depends on its temperature and may require holding at part speed during this phase. If no bypass stack is provided the HRSG must be designed to take the larger rate of temperature change, and the condenser must take the full steam load while the boiler is brought up to operating conditions.

7.12 MASSIVE STEAM INJECTION

Massive steam injection for power augmentation, as opposed to injection for NO_x control, is possible. It uses a gas turbine with exhaust heat recovery in a HRSG, as in a combined cycle, but the steam raised is injected into the gas turbine to increase the turbine mass flow and increase power. It is essentially a combined cycle without a steam turbine, but with the steam sharing the same turbine. This is cheaper in capital as the steam turbine and generator are not needed.

Steam injected engines have an increase in power of 25 to 60%, and an increase in efficiency of 10 to 20%. They suffer from fundamental mixing efficiencies which separate turbines do not have. Also they lose the injected water, so require a continuous large supply of high quality treated water. Water needs to be typically treated to remove particles down to 20 microns and for Na and K from all source (fuel/air/steam) to less than 200 ppb in the hottest section of the turbine [148]. These problems mean that although steam injection has a niche in the market it is unlikely to dominate power production in the future.

8 DEVELOPMENTS PAST AND FUTURE

8.1 INTRODUCTION

The first section gives an overview of the past technology trends, then the second section gives an overview of what can be considered as the current state-of-the-art. The last sections give an indication of the potential for improvement in the future. It is possible to make some estimates about the future trends in gas turbine and combined cycle performance, but the nature of the exercise means the boundary between predictions and speculation is somewhat hazy.

8.2 HISTORY AND TECHNOLOGICAL TRENDS

8.2.1 History of Development

The gas turbine industry is still very much developing and progressing. Unlike steam technology it is not yet sufficiently mature to have slowed the high rate of improvements. Currently with a rapid increase in sales and with a predicted boom in new electrical generation capacity the industry is preparing for growth. The number of companies has been reduced by mergers and joint ventures but this is to pool resources as the technology becomes more demanding, and on a whole the industry is positioning itself for improved engine performance and competition.

In the 1970's the gas turbine industry was at a peak, and several large engines were under development such as the FT50, Frame 10 and W1501, all rated at over 100 MW. After the oil shock, the market collapsed and much development was halted. But in time development did proceed again, with a new 'third' generation of large machines. These include the ABB 148 MW type 13E (1983), the GE 150 MW frame 7F (1987), the GE 200 MW frame 9F (1987), the Westinghouse 145 MW W501F (1989), and the KWU 155 MW V84.4. Several manufacturers are also developing direct flange to flange replacements of newer engines for the existing older low performance machines.

The current uprating and improvement are building on existing engines and are results of development work over the a number of previous years. Industrial TRIT is up to 1533 K on some models and higher pressure ratios are being used. The new engines reflect more aero-derived technology, which in turn reflects the large investment in military engines over the last few decades. The US for instance have been funding military aero-engine R & D by over \$400 million per year for the last 12 to 15 years. Over the last 15 years large improvements have also been made in industrial engine reliability, availability and maintainability, largely due to work by EPRI.

The third generation of engines currently entering the market operate on a simple cycle efficiency of 34 to 38% electrical efficiency, and lower maintenance costs are also predicted. An example of these engines is the GE industrial frame 9F currently (1990) at 212 MW and 34% electrical efficiency. It has film and impingement cooled nozzles, ceramic thermal barrier coatings on the combustor, the first two compressor stages are transonic and has a TRIT of 1533 K. The exhaust temperature is at 856 K and thus offers a potential bottoming cycle with an efficiency of about 30% (based on a conventional optimised two pressure steam turbine). The combined cycle efficiency is currently quoted at 51.6%.

Modern compressors have less stages and fewer blades than earlier engines. Each stage has to do about 1.5 to 2.0 times the work of early designs. This technology is assisted by computer models, computer aided design and advanced manufacturing procedures for complex aerofoil shapes.

Turbine development has largely centred on increasing temperatures. The higher temperatures are achieved by improved materials, cooling techniques, development of corrosion resistance in materials and by coatings. Material properties have increased steadily over the years, and development centres on creep-rupture, hot corrosion and oxidation resistance. After an initial rapid rate of gains the advancement in metal capability slowed in the 1970's due to hot corrosion becoming more limiting than strength. Also air cooling advancements separated the metal temperature limitation from the firing temperature. Increased use of poor fuels has added pressure for corrosion resistance. For the period 1960 to 1980 the blade temperature for constant creep rate on industrial blades has improved on average by 3.5 °C/yr [11]. Major advancements in aero-engines have been directionally solidified and single crystal blades which are only now being used in industrial engines.

The 1980's has seen emphasis on processing and air cooling techniques to improve TET without reducing corrosion resistance. As an example the history of cooling in the company GE can be followed. GE introduced air cooling in first stage nozzles in the early 1960's, improving the TRIT from 1090 K to 1180 K by the late 1960's. Moving blade cooling required a second air circuit internal to the engine, and for GE was introduced in 1972 in the MS7001B using a TRIT of 1280 K. Currently the latest GE engines have film, impingement and convection cooling in the first nozzles, serpentine passages in the first blades, impingement and convection in the second nozzles, drilled radial holes in the second blade, and convection cooling in the third nozzles. Other manufacturers will have similar development patterns. Since 1950 GE industrial turbine rotor inlet temperature has advanced 450 °C, or on average 11 °C/yr [148]. But there were two quantum changes, and average steady increases are more like 9 °C/yr.

Development in military marine intercooled and regenerative engines was started by the US navy. The result includes the marine ICR Spey giving 22 MW at 43% efficiency with a near flat part load efficiency down to about 50% load. This work may have spin off benefits in the Utility market.

8.2.2 Past Engine Performance Trends

Past trends in engine performance are presented in tables 8.1, 8.2 and 8.3. This data is from [22] and back copies of this handbook.

It is usual that engine performance is improved in a series of steps over the lifetime of the engine. Typically heavy industrial engines have had a large increase in power output and improvement in efficiency during their life. An example is the GE frame 7 which entered service in 1970 at 47 MW, 28.6 % efficiency, and grew to 147 MW, 34.3% efficiency in 1988 [148 and 11]. 65% of this increase is a result of higher TET, while the remainder is due to increased air flow and pressure ratio from

compressor developments. Also the Westinghouse W501 started life in the late 1960's at 45 MW and a TET of 1155 K, base-load, giving 27% efficiency. Today it is rated at 100 MW and 34% efficiency with a TET of 1360 K [11]. Trends are shown for two typical engines, over a 10 year period, in table 8.3. GE compressor trends are shown in figure 8.1.

Table 8.1 Aero-derivative Engine Trends (current performance)

	Year introduced	Isentropic gas power MW	Electrical Power MW	Electrical Efficiency %	Compressor stages	Pressure ratio
Olympus	1962	33.11	28.58		12	11.0
Avon	1964	17.75	15.56	28.4	17	9.2
LM2500	1970	25.13	21.23	34.8	16	18.7
RB211	1974	29.08	26.10	35.6	13	20.0
Spey	1976	14.54	12.55	34.4	16	19.0
LM5000	1977	39.60	33.06	35.1	19	30.0
LM1600	1988	15.47	13.40	36.1	10	21.5

Table 8.2 Heavy Industrial Engine Trends

	Year introduced	Electrical power MW	Current electrical efficiency	Original Power MW	Original efficiency %	Pressure ratio
PG3002	1949	10.20	25.28	10.20*	25.20*	7.2
PG5001P	1958	25.28	27.75	18.90*	25.47*	8.1
W171G	1960			16.05*	25.13*	7.0
MW191	1963	17.79	25.39	17.79*	25.39*	7.0
GT35	1968	16.53	31.75			12.0
GT11	1971	74.00	32.02	72.50*	31.70*	11.0
V94	1974	149.25	33.67	111.78*	30.87*	10.8
PG7111EA	1976	83.50	32.57	72.90*	31.63*	12.4
PG6001B	1978	37.40	31.42	31.75*	30.25*	12.4
MW501	1980	104.57	33.26	99.87	32.60	14.0
MW701	1981	130.55	33.89	121.80	32.00	14.0
GT13E	1983	145.80	34.04	142.90	33.71	14.3
GT8	1984	46.90	32.75	46.40	32.02	16.3
V84	1985	102.53	33.63	91.50	32.32	10.6
PG9281F	1987	212.20	34.15	202.30	34.06	13.5
MW501F	1989	145.00	34.12	145.00	34.12	14.2
* performance in 1979/80.						

Aero-derivative engines have tended to have less performance growth over their life time than heavy industrials. The LM2500 was introduced in the early 1970's at 23.11 MW isentropic gas power base-load, and currently is rated at 25.13 MW isentropic gas power. Although aero-derivatives have less improvement in their life, the development is often funded by the aero-engine development.

It is normal for new engines or models to be conservatively rated until

after testing. Then the next 'uprate' is to the level of performance that is only what was originally expected. This is to cover the design engineers and the manufacturing company. Also even after reaching a mature design stage a margin for unit to unit variation is included. For example the highly mature LM2500 engine still has a 3% margin on guaranteed performance for unit to unit variation. Consulting engineers may also add a margin on the manufacturers margin to further cover themselves.

Table 8.3 Typical 10 Year Performance Improvements

	GT13 Power MW	Effy. %	W501D Power MW	Effy. %
1978	77.20	29.20	89.86	30.73
1979	79.40	30.00	92.24	30.82
1980	81.00	31.40	95.25	31.60
1981	81.00	31.40	95.25	31.60
1982	88.90	31.80	96.44	32.02
1983	88.90	31.80	96.44	32.02
1984	88.90	31.80	96.44	32.02
1985	91.10	32.61	104.40	33.17
1986	91.10	32.61	104.40	33.17
1987	100.50	32.83	104.40	33.17
1988	100.50	32.83	104.40	33.17
1989	100.50	32.86	104.40	33.17

Figure 8.1 Industrial GT Compressor Trends - GE

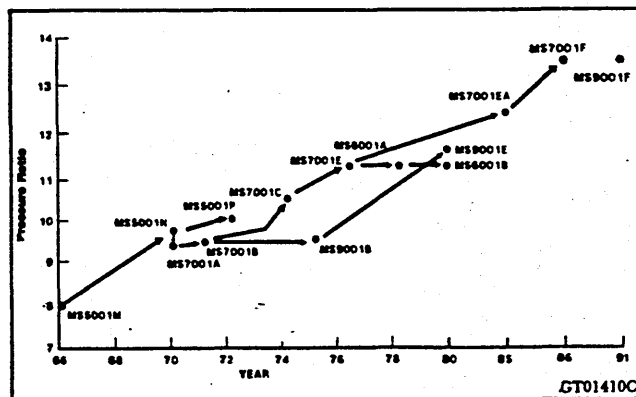
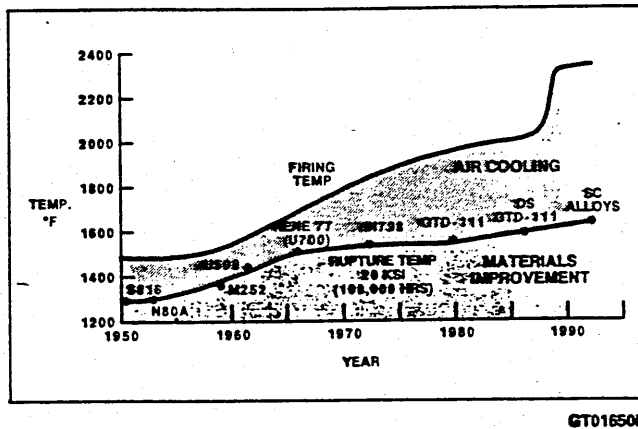


Figure Growth in compressor pressure ratio (ISO conditions)



8.3 CURRENT GAS TURBINE TECHNOLOGY

8.3.1 Components

Compressor

The loading of the compressor or turbine stage effects it's efficiency. Modern industrial compressors are routinely designed for a polytropic efficiency of 0.87 to 0.90 depending on loading [11]. Pitch line axial to blade velocity ratios are typically 0.4 to 0.6, and hub radius ratios are rarely less than 0.45 [11]. Aero-engines have tended to use higher speeds and have fewer stages than heavy industrial engines to reduce weight. But modern industrial engines are starting to use transonic inlet stages even with the slight penalty of performance with supersonic tip speed. Modern engines have also tended to have fewer more sizable blades and vanes. Aspect ratios are 2.0 to 3.5 which is lower than in early engines. Compressor blades are typically made from a stainless steel, and coated for the first few stages against moisture induced corrosion.

Turbines and Cooling

For modern large sized turbines the polytropic efficiency decreases from about 93% to 88% as the pitch line loading parameter goes from 1.2 to 2.0 [11]. This is for hub ratios above 0.8. Typical heavy industrial loading parameters are 1.3 to 1.75. High loading is often used as the reduction in efficiency is offset by reducing the number of stages and the cooling requirement.

Nearly all turbines use some form of cooling from compressor discharge

air. Blades and vanes are typically hollow investment castings using internal cooling passages. In aero-engines the cooling tends to include internal impingement on the leading edge, internal fins and external film cooling. The cooling air is discharged through the tip, trailing edge and film cooling holes. Industrial cooling has traditionally been straight passages in blades without film cooling, but modern heavy industrial engines are now using all the aero-engine technology. The discharge of cooling air through the tip helps to block tip leakage flow. Aircraft engines use good fuels and do not have the same problem of cooling hole blockage as has occurred in some industrial situations, so some highly developed new industrial engines will be unable to run on heavy fuels. In addition to vane and blade cooling, minor flows are used to seal casing, structures and wheel spaces.

The performance of the GT is a compromise between a high TET and low cooling flow. Traditional simple heavy industrial engine cooling schemes can require up to twice the air for cooling for given temperatures than aero-engine cooling technology, but heavy engine have tended to run at lower temperatures. A look at table 3.1 shows that the average TET for aero-derivatives is over 100 °C higher than for the heavy industrials, and the cooling flow is in reality about 20% in aero-derivatives compared to about 10% in heavy industrials. The temperature difference between aero-derivatives and heavy engines is being eroded with the introduction of third generation high temperature engines. When considering performance, work is extracted in the moving blades and the first stage nozzle cooling flow is 'nonchargeable' and behaves in a similar manner to the combustor dilution flow.

The first stage moving blade is usually the critical limiting component, due to stress and temperature. High temperature rotating blades are typically nickel based superalloys such as IN-738 and subsequent improved alloys, strengthened by solution and precipitation hardening. Cobalt based superalloys have less strength but more corrosion/oxidation resistance and are typically used in hot stationary components such as nozzles. Aero-engines use clean fuels and tend towards strength rather than corrosion resistance, hence use alloys like Rene 80. High temperature materials are expensive relative to other materials but only represent about 2% of the total cost of a combined cycle plant [148], yet dictate it's performance. Hot corrosion resistant coatings are usually used.

Combustors

Combustor design has developed in a largely empirical fashion. The main design requirements are flame stability in a high speed stream over a wide fuel flow, from ignition to full power. Other problems are emissions and exit temperature profiles, and further the combustor must be simple, rugged, maintainable yet small enough to be cooled with available air flow. Neither mathematical simulation nor scale model testing has proved successful, and the combustor has traditionally been a set of 'cans' where an individual liner can be tested full sized. The liners are typically 200 to 400 mm in diameter so they are easy to build, test and replace, and the number is varied for different engine sizes. ABB and KWU both use large 'silo' combustors, and aero-derivatives often use single annular combustors.

Fuel is discharged into the combustor via fuel nozzles. Liquids are

often pressure atomised or, if smoke is a problem, air atomised. 'Can' type combustors require transition pieces to duct the air to the turbine, and these pieces also require cooling. The combustor has traditionally been cooled by a film of air introduced by louvres. Slots techniques are now widely used to improve the cooling efficiency. The combustor exit temperature pattern is usually designed to give lower temperatures to the base of the blades where stress is highest. Aero-engines have moved largely into single annular combustors to eliminate the transition pieces and improve the emissions. Typical materials include Hastalloy, HS-188 and/or Nimonic 263. Ceramic thermal barrier coatings are being used in the latest combustors.

As the TET increases more air is needed in the combustion primary zone, and less air is available for cooling. This has to be offset by more efficient cooling, or by reducing the surface area of the combustor. Also higher temperatures require more intense heat release, and thus more vigorous mixing, resulting in higher combustor pressure drops.

Current development is concentrating on emissions. Smoke emissions have been largely eliminated, and water and steam injection are established techniques for reducing NO_x . Dry low NO_x combustors are the new development with other techniques like two stage combustion chambers. Multiple nozzles reduce the NO_x formation and have been found to reduce noise and wear increasing combustion inspection intervals. Medium and low CV coal gas are inherently more favourable towards low NO_x and future coal gasification is not expected to produce problems for limits down to 50 ppm [11].

8.4 POTENTIAL PERFORMANCE IMPROVEMENT

8.4.1 Cycle Inefficiencies

For the simple cycle gas turbine any further improvements in performance need to come from improved component efficiencies and from pressure and temperature increases. The main sources of inefficiency are the cooling flow, compressor efficiency, turbine efficiency and combustor pressure loss. Other losses include the inlet and exhaust pressure drop, and the generator efficiencies. The cooling flow reduces efficiency by mixing, diluting and cooling the gas flow when it is discharged from the blade/vane and also from disturbing the flow path. Compressor and turbine efficiencies include friction, tip leakage and blade flow effects. The combustor pressure loss reduces the pressure available for the turbine expansion.

Full performance evaluation requires information on the component efficiencies, blade cooling, leakage, heat losses, gear and generator losses. These are hidden among the data publicly available and are usually considered propriety. Information that is available is often inconsistent, and may include uncertainty margins for guarantees. Accurate evaluation of an engine can only be done by the manufacturer who can resort to exhaustive measurements to calibrate computer programs. Even if the data was available it would only refer to single engines and would not be easily generalised to all machine sizes and technology. But to give an indication of the effects from small changes to temperatures and efficiencies a mathematical model for design point performance has been developed.

Appendix A contains a mathematical model based on the frame 9F heavy industrial engine. The values known are;

Pressure ratio	13.5
turbine rotor inlet temperature	1533 K
efficiency	34.1%
exhaust flow	613 kg/s
exhaust temperature	856 K

The unknown data was adjusted within reasonable bounds to achieve the above values using the same theory as in section 3.4.2 but with polytropic efficiency definitions. The assumed geometry and cooling is shown in figure 8.3. The assumptions are;

compressor polytropic efficiency	87%
turbine polytropic efficiency	88%
total cooling flow	15%
split 1) 1st nozzle	40%
2) Effectively before turbine	40%
3) Effectively after turbine	20%
combustor outlet temperature	1568 K
combustor pressure loss	5%
shaft and heat losses	0.7%
generator and auxiliaries efficiency	98%
inlet and exhaust losses,	
effect design point efficiency	2%

These assumptions are realistic and are give the appropriate efficiency, exhaust temperature and TRIT.

Figure 8.3 Cooling Geometry Frame 9F

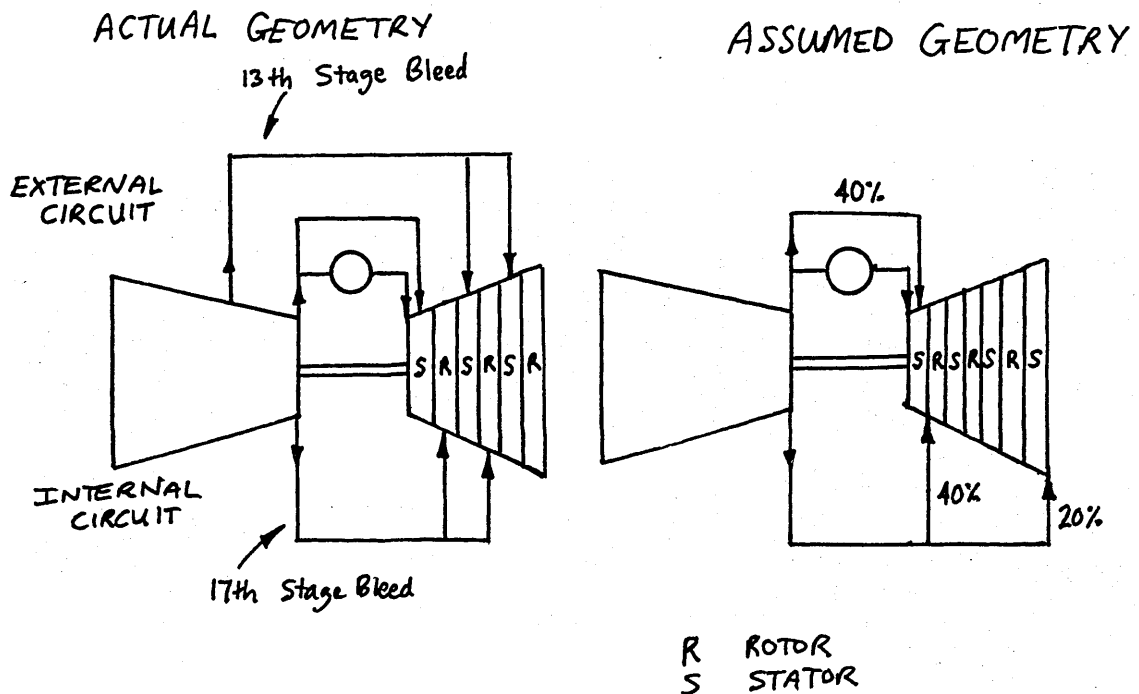


Figure 8.4 Effect of Inefficiencies - GT Efficiency vs Pressure Ratio

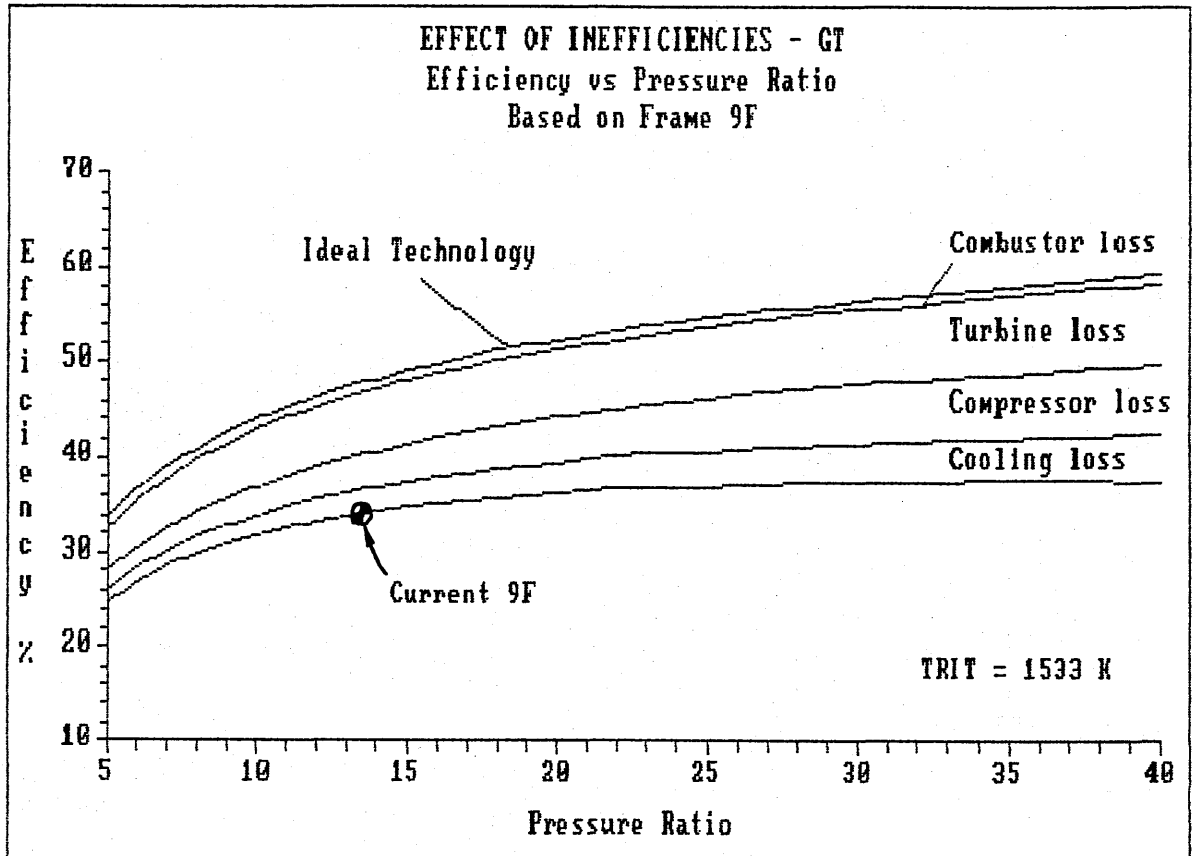


Figure 8.5 Effect of Inefficiencies - GT Efficiency vs TET

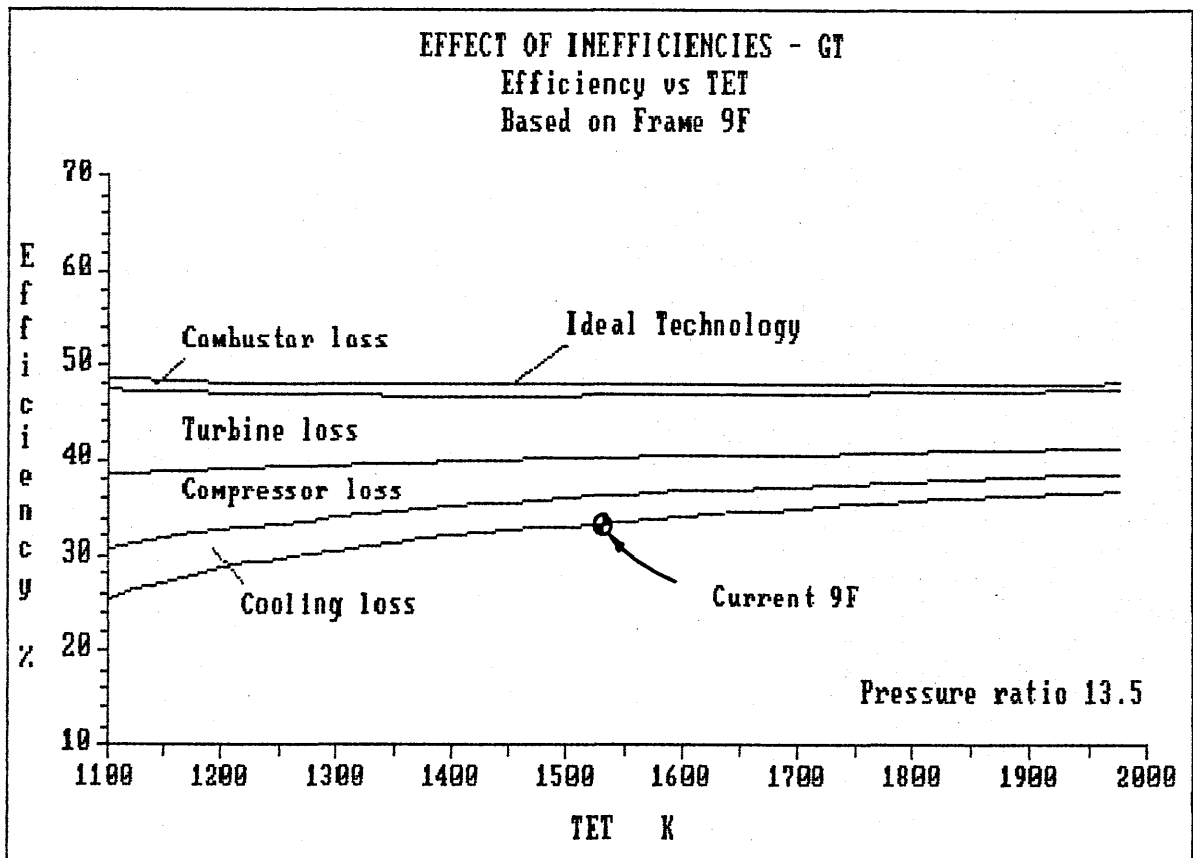


Figure 8.6 Effect of Inefficiencies - CC Efficiency vs Pressure Ratio

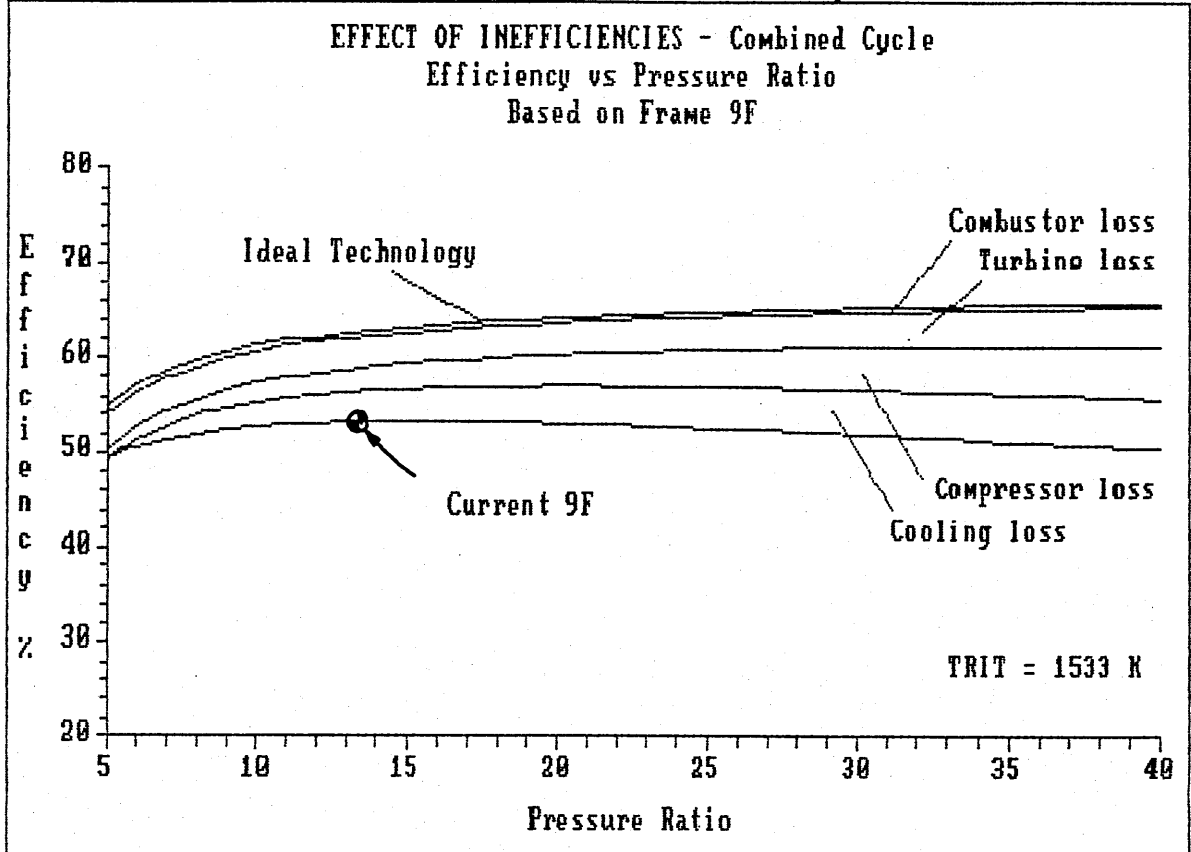


Figure 8.7 Effect of Inefficiencies - CC Efficiency vs TET

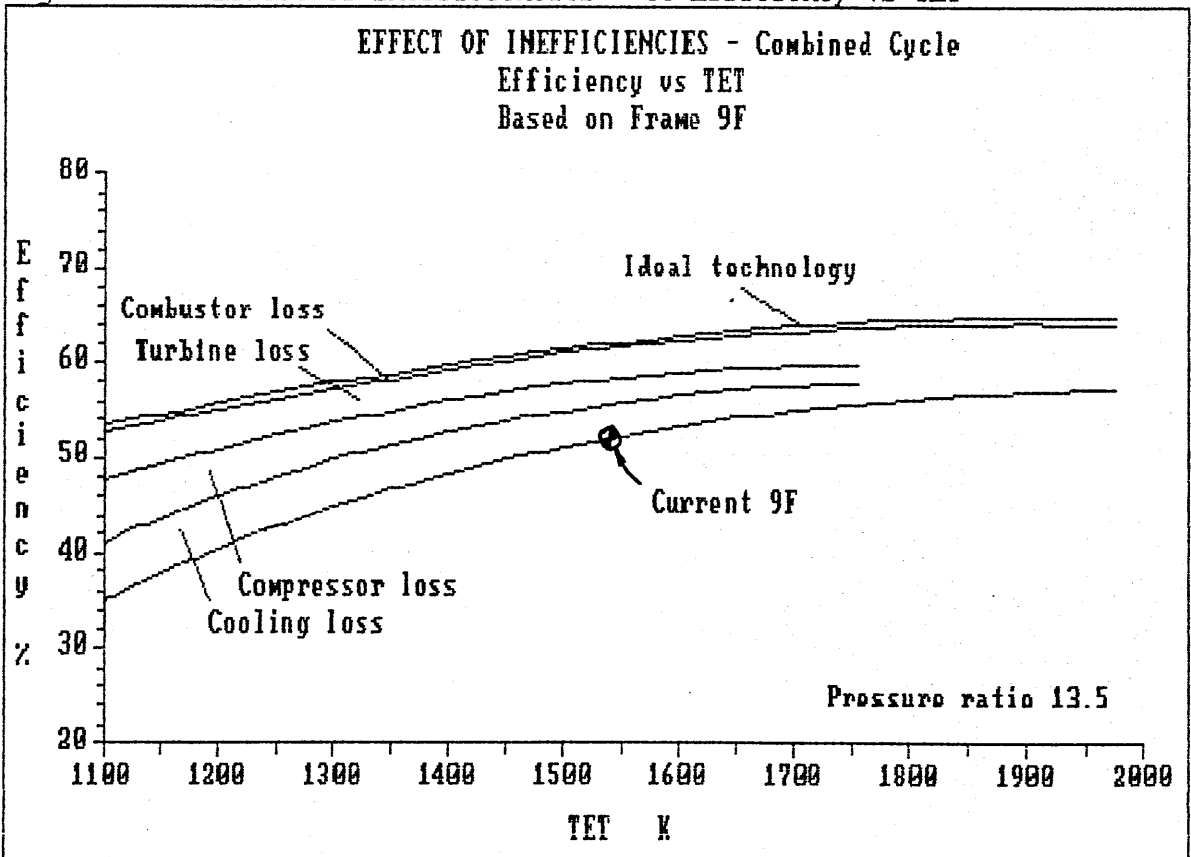


Figure 8.8 Gas Turbine Efficiency - Varying Pressure Ratio and TET

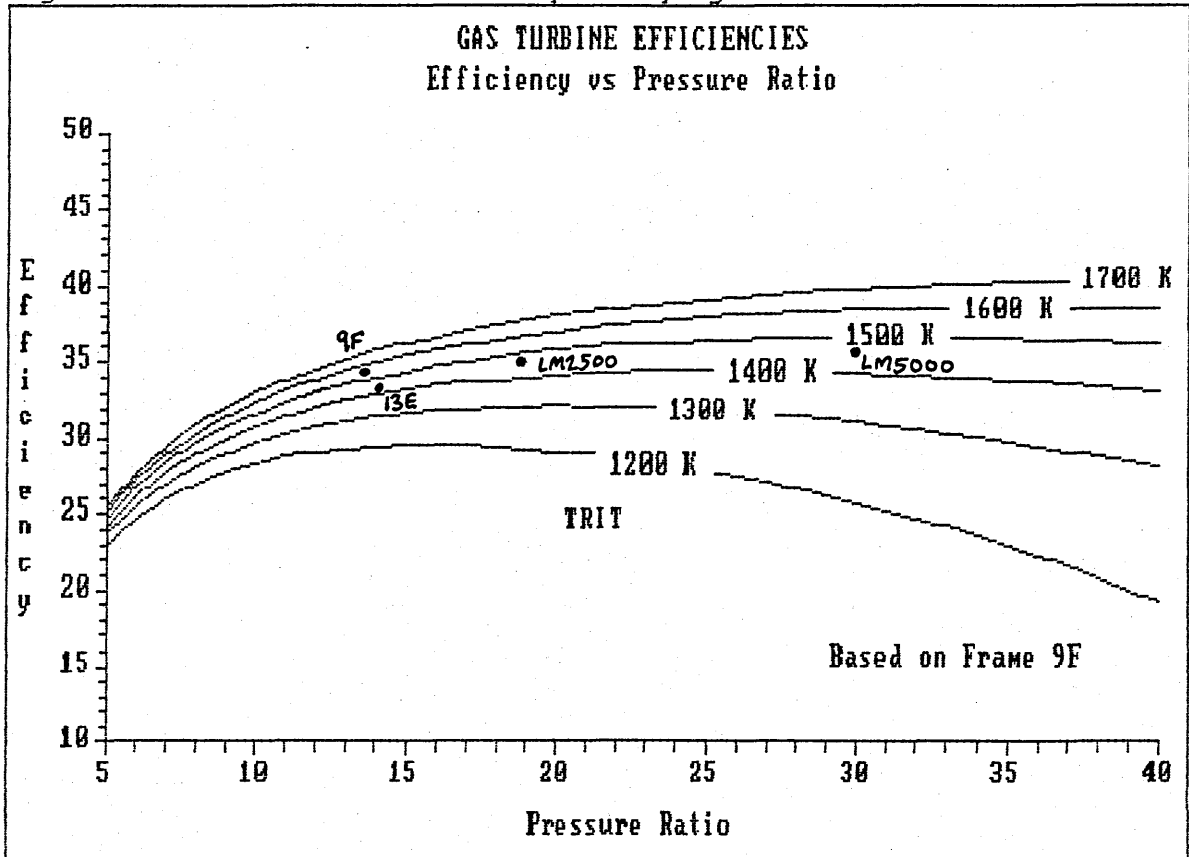
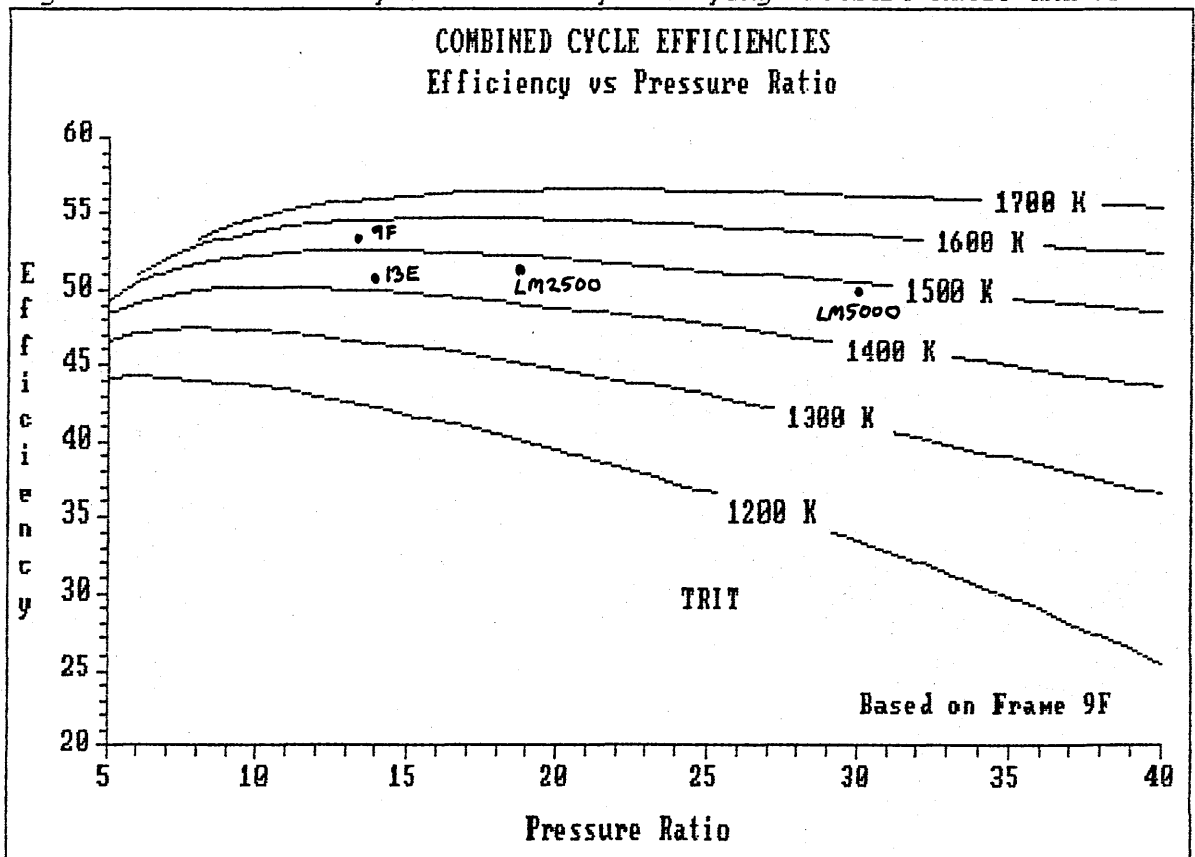


Figure 8.9 Combined Cycle Efficiency - Varying Pressure Ratio and TET



The trends in performance of the current gas turbine technology, based on the frame 9F are shown as the pressure ratio and TET is altered in figures 8.4 and 8.5. The losses were also reduced to zero to provide curves indicating the order of magnitude of the four major losses and the resulting ideal curves for no losses. As is to be expected the ideal cycle efficiency without losses is nearly independent of TET. The reason it is not fully independent is that the air does not have constant ideal properties and that the fuel flow is added to the exhaust gasses increasing power. From the figures it can be seen that for a simple cycle gas turbine, the efficiency is still to be improved by increasing the TET and the pressure ratio. As the component efficiency and TET improves the pressure ratio should also increase to maximise the effect.

To find the effects of the inefficiencies, pressure ratio and TET on the combined cycle, equation 7.17 is used. Here the bottoming efficiency is an optimised cycle using conventional technology of either single pressure reheat or two pressure steam cycle. The results for varying pressure ratio and gas turbine TET are shown in figures 8.6 and 8.7. The gas turbine component efficiencies are also made ideal and the resultant ideal gas turbine with a conventional bottoming cycle is shown.

Figures 8.8 and 8.9 are similar to figures 3.11 and 7.15 but referenced to the frame 9F and current technology. The combined cycle behaves differently for improved TET and gas turbine pressure ratio than the simple cycle gas turbine. The effects were discussed fully in the previous sections, and the main points to note are that the higher pressure ratio gas turbines are the most efficient simple cycles, such as the LM5000, but for maximum combined cycle efficiency the gas turbine needs modest pressure ratio and high temperatures.

These generalised predictions on combined cycle efficiency must be viewed with the realisation that steam cycle efficiency is highly dependant on the configuration, and figure 8.9 is for an optimal two pressure steam cycle. Real combined cycles may be less than this potential maximum efficiency. Also figures 8.8 and 8.9 match best engines similar to the frame 9F in component efficiency and cooling.

8.4.2 Component Developments

Compressors efficiency

Figure 8.9 shows there is little to be gained from large pressure ratios in conventional combined cycles. This means future improvements in are more likely to concentrate on increasing component efficiencies by low loading of compressors and complex three dimensional blades. Current development in industrial compressors continues in mechanical design and materials for higher blade speeds, aerodynamic refinement for higher efficiency and use of variable geometry such as tip clearance control. Industrial engines may see more use of multiple stages of variable stator vanes.

Turbines temperatures

Figures 8.8 and 8.9 indicate that there is even more gains from high temperatures in combined cycles than in simple cycle gas turbines, and large industrial engines are likely to be developed for increasing temperatures. Current developments in materials give promises of a 140

°C gain in metal temperature over a standard Rene 80 casting, including single crystal RAM-DS and DS-Eutectic [11] (DS = Directionally Solidified). Although aero-engines have been using directionally controlled and single crystal blades, much of this technology has yet to be used in heavy industrial engines. Addition processing and new coating techniques will also increase performance. An example is hot isostatic pressing to increase casting strength. Also ceramic core technology developments allow more complex cooling passages and it is projected that blades so produced will reduce bulk metal temperatures by about 20 °C [11].

New materials for blades or nozzles can take 3 to 5 years to be tested. These tests include creep, fatigue and corrosion. And even after laboratory testing, field tests are needed, therefore the evolution time is slow.

Turbine temperatures cannot increase much beyond 1700 to 1800 K if NO_x emissions are to be controlled. This can be seen from figure 4.2. Selective catalytic reduction as it currently stands involves a consumable, ammonia, which is itself toxic. This asks questions as to the likelihood of this method being used for NO_x control. Even if temperatures stagnate between 1700 and 1800 K improved cooling technology will reduce cooling air requirements.

Thermal barrier coating

Thermal barrier coatings give the potential for a quantum leap in improvement of turbine temperatures. Development work is being carried out, and the technology is now applied to combustors, transition pieces and turbine vane platforms, but mainly at present to increase life of the component rather than to improve the TET.

Work by Pratt and Whitney [71] claims that a 0.25 mm layer of zirconia can reduce metal temperatures by as much as 170 K. This is equivalent to 20 years of cooling technology improvement. But these benefits are only of practical use when all technological barriers are overcome. The barriers are large and the development slow.

The largest problem is loss of the brittle ceramic coat to thermal fatigue, and the need to be sure of reliability, and predictability of the coat before it is relied on in service. The loss of the coating is driven by thermal cyclic strains from expansion differences, and improvements in processing have dramatically increased this life. The other problems are, life prediction, surface roughness, maintaining cooling hole geometry and blade aerodynamics, and the increased blade stress due to the reduced cross sectional area.

Coating techniques are plasma spray and more recently Electron Beam Physical Vapour Deposition (EB-PVD), of which P & W [71] claims that laboratory tests have shown durability of EB-PVD to be about 8 times that of a plasma spray. Field tests showed durability of at least 3 times plasma spray. But the key is not only a better average life but an 'assured' life better than the life of the blade. It has been shown that the predictability of the coat is comparable to the predictability of other materials properties that have cyclic and/or time dependant failure, such as creep and fatigue. P & W expect to have purpose designed blades providing a benefit from thermal barrier coatings in

aircraft in the early 1990's. Parallel work to the application technique is to develop better ceramics at same time. This is also expected by P & W to be available in the early 1990's.

Ceramics Components

Research is continuing towards ceramic components. In small turbines ceramics have been used experimentally in rotors, nozzles, combustor liners, regenerators, and ducting. In large engines the principal use to date has only been in coatings.

8.4.3 Cycle Developments

Cooling

There some but limited scope to improve temperatures by cooling the compressor bleed flow prior to cooling the turbine [48]. But this is being done by ABB in their 13E engine. The heat is used to preheat fuel and feedwater [256]. The most important improvements can be obtained by dispensing with compressor air for cooling, and replacing it with say steam from the bottoming cycle. A US study (RP2620-1) has investigated cooling and indicated open circuit steam cooling may even reduce efficiency, but substantial improvements in efficiency are possible from closed circuit steam cooling.

The advantages of closed circuit cooling are no mixing and dilution losses. The high pressure steam also has a high density and high specific heat capacity and thus has a higher heat transfer potential. The heat transferred is then recovered in the steam turbine.

The EPRI study RP2620-1 showed the improvement with closed circuit steam cooling in a combined cycle to be 6% (+3%) on efficiency and +8% on specific work [48]. The reference example study was a gas turbine of 300 MW, pressure ratio 16 and TET of 1644 K. The four stage turbine had all nozzles and blades closed circuit steam cooled, and air bleed used for wheel spaces. This gave an efficiency of 54.4%. At a TET of 1866 K the efficiency rises to 56.6% and the ultimate efficiency of no cooling is suggested as 57%. An interim proposal was for a 200 MW turbine at 1622 K with a 3 stage turbine. It has an air cooled 1st nozzle, uncooled last stage and steam cooled between. The expected efficiency is 53% and at 10% less cost in dollar per kW than a conventional combined cycle. It is seen as the near term future as it does not have to wait for new materials and component efficiencies, and from the description is probably based on the GE frame 9F. Overall efficiency is not expected to be sensitive to the leakage of steam into the GT.

The main problem with closed circuit cooling is that it is untested, although ideas exist, and would require an expensive development program. A possible operational problem is slow starting.

Intercooling

Intercooling does not always improve combined cycle efficiency, but intercooling may improve other factors. It allows a higher pressure ratio up to 50 or 60, the intercooling heat has possibilities of use within the steam cycle. It also means the compressor discharge can arrive hundreds of degrees cooler, which reduces the turbine cooling

air requirement.

Intercooled and steam injected

Intercooled and steam injected engines have been suggested, which give a near combined cycle efficiency but only requiring a HRSG and not the expense of the steam turbine. The most likely engine candidate is the LM5000, which currently with steam injection gives 52.5 MW at 44.4% efficiency nonintercooled. This engine is twin spooled and high pressure ratio, and has sufficient room between the compressors for an intercooler connection. GE claims an advanced intercooled design of the LM5000 could potentially give 115 MW at 52% efficiency on natural gas. The main problem is the development costs. It is reported to have taken \$3-4 million to develop the steam injected LM5000, but an advanced intercooled engine could take 3-4 years and over \$100 million.

Gas Turbine Reheat combustor

Reheat in a combined cycle gives potentially large performance benefits. It improves efficiency, gives less dependence and sensitivity to the peak TET and it also improves part load efficiency. The higher exhaust temperature also allows a more complex steam cycle, improving efficiency again. Reheat is discussed more fully in section 7.7.2.

A major problem is cooling with performance for reheat sensitive to the cooling requirement. Gains are smaller for less advanced cooling techniques. Using advanced cooling technology such as closed circuit steam then has considerable benefits. The commentator in reference [191] suggests the optimum pressure ratio is about 12 to 25 without intercooling, and that use of advanced cooling technology like liquid metal thermosyphons could make a thermal efficiency of over 60%.

Reheat combustors are not an established technology and pose a significant development problem. One suggested solution is using an aircraft reheat combustor (after burner) using vee gutter flame holders. The limited stability of these combustors may be satisfactory as they are not needed for starting and reheat would tend to be used in base-load combined cycles that operate in a narrow load band. Aero-engine reheat chambers have a very limited life under actual operation, hence ceramic liners and possible steam cooling would be required. Other suggested solutions are more conventional combustion chambers but using steam cooling which is then a part of the steam cycle superheating.

The present LM5000 has a pressure ratio of 30 and gives a pressure ratio over the power turbine of 3.75, which is a little low for reheat. It would make a large reheat combustor and limit the reheat temperature to about 1280 K, based on the temperature of the exhaust being a maximum of about 980 K for exhaust duct materials. A pressure ratio of 38 would be better which according to reference [186] with a TET of 1590 K gives a power turbine ratio of 5.3, and with steam cooling and injection the ratio would be an even more acceptable 6.5 allowing a reheat to 1420 K. This commentator quotes a potential LM5000 reheat and steam cooling/injection combined cycle efficiency of 56%.

The problems in reheat is likely to limit it to a single reheat stage in the foreseeable future. Possible problems are also expected with

increased NO_x formation. As the aero-derivative gas generator TET and pressure ratio increases the benefits of reheat improve, and so does the likelihood of an engine being developed. The most important work on reheat combustors is that being carried out by the moonlight project, a consortium of Japanese companies discussed in the next section.

Moonlight Project

The Japanese national 'Moonlight' project for advanced power plant research, started in 1978 to developing a combined cycle aiming at 55% efficiency. The development engine called the AGTJ100A, is a very high pressure ratio intercooled gas turbine with reheat. The gas turbine has been built and tested at reduced power, with the overall proposed plan for a 6+1 combined cycle block rated at 1000 MW. Work has proceeded slowly and meanwhile conventional cycles are gradually approaching the targeted efficiencies of the advanced cycle. Figure 8.11 gives the intended layout.

The gas turbine uses two separate shafts, the LP shaft is synchronised while the HP is free. Intercooling is applied by water spray injection at a pressure ratio of 4.8. The advantages of water injection as opposed to a heat exchanger are improved performance and reduced NO_x emissions, but it also requires a supply of demineralised water. The intercooler drops the temperature from 199 °C to 78 °C by increasing the humidity from a design point of 60% to 90%. The overall pressure ratio is 55 and the first TET is intended to be 1570 K. The reheat is at a pressure ratio of 8.8 to a TET of 1444 K, and gives an exhaust temperature of 880 K. This exhaust temperature is sufficiently high to allow a steam TET of 840 K and steam reheat. Using natural gas and a two pressure HRSG allows reduction of the stack temperatures to only 95 °C. The next engine, AGTJ100B intends to have a TET of 1670 K. Steam reheat allows higher steam pressures and improved steam efficiency.

The gas turbine configuration of intercooled and reheat exhibits excellent part load performance in particular by maintaining near constant exhaust temperature at lower load. But in a 6+1 block this may not be a useful feature. References are [259] and [260].

Bottoming Cycle

For an optimum bottoming cycle the cycle needs a triangular shaped T-s curve. The two main improvements to the Rankine cycle are the two pressure cycle giving a stepped T-s curve and the supercritical cycle, but which has a very high pressure with associated low efficiency turbine. A third possibility is to use a mixture of fluids, but this also gives a variable condensation temperature offsetting the advantage.

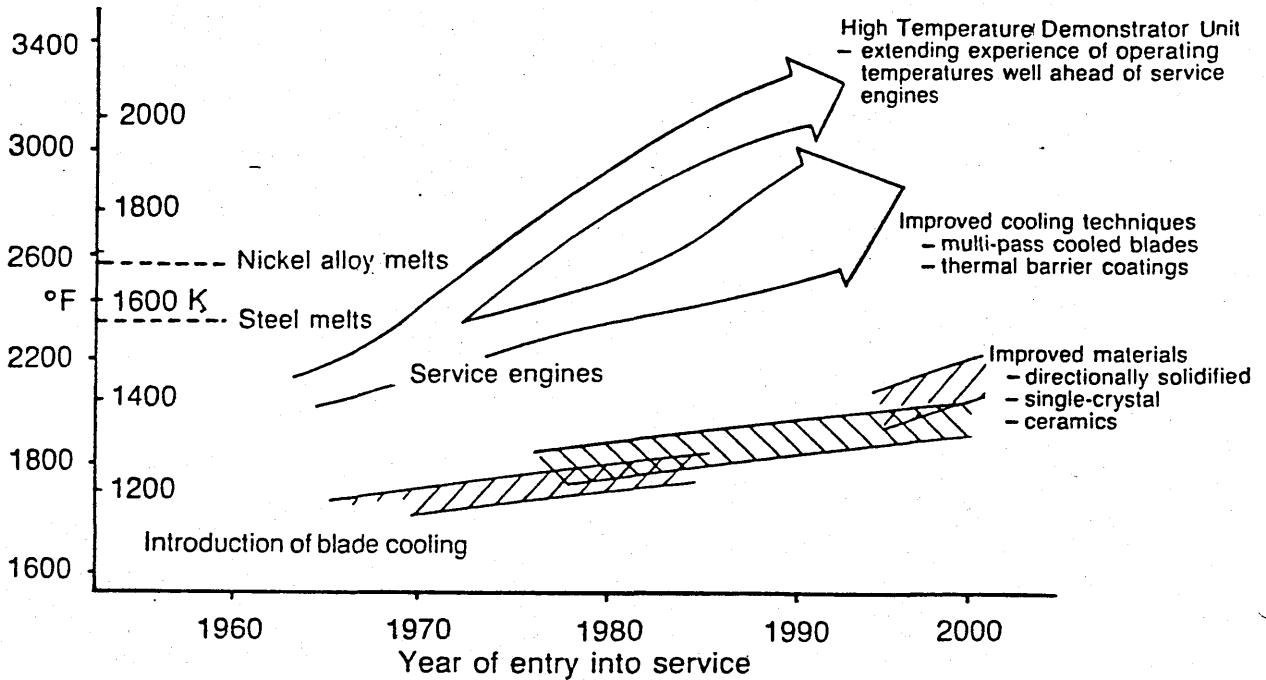
The Kalina cycle is a non-Rankine cycle using a multicomponent fluid (water/ammonia) and has variable boiling temperature. The low temp from turbine outlet is used for distillation rather than evaporation, which is better thermodynamically and the quantity of heat rejected is less than in Rankine cycle. The cycle is not of high cost, but is unused as yet commercially. A demonstration engine has been developed. Further reference can be found in [243].

Figure 8.10 Aero-engine Trends - Rolls Royce



Advanced turbine technology

Turbine entry temperature



Polytropic efficiency for multistage LP turbines

Polytropic efficiency %

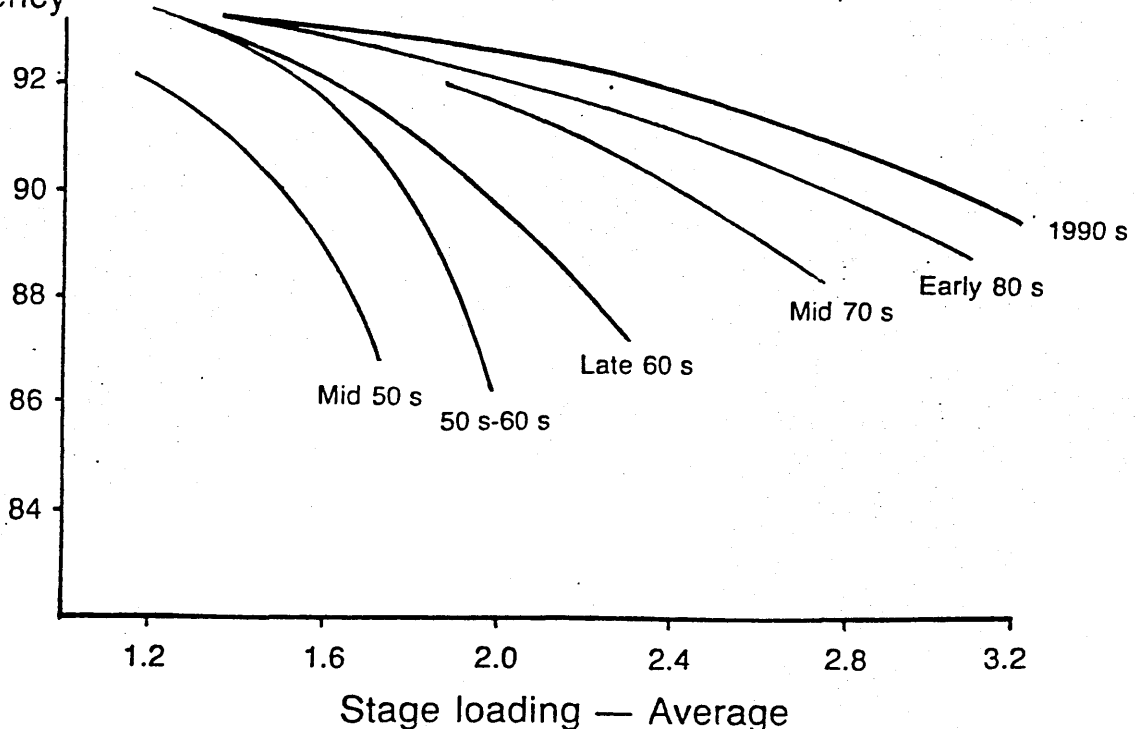
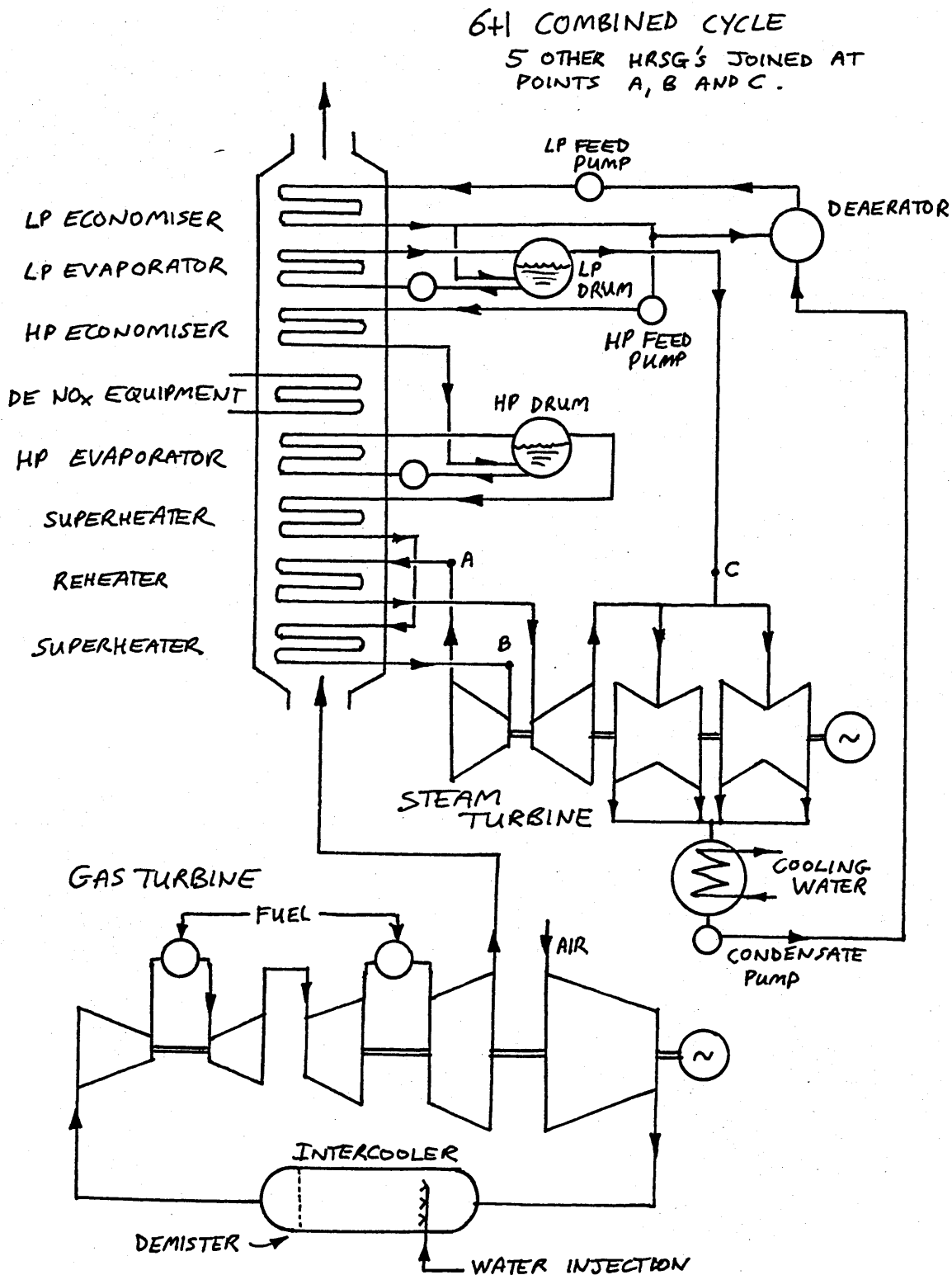


Figure 8.11 Advanced Gas Turbine Cycle - AGTJ100A Moonlight Project



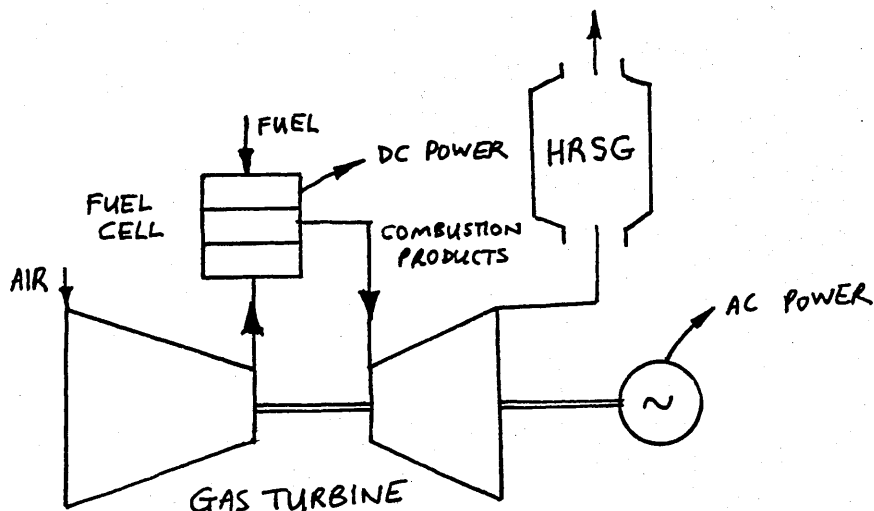
It was noted in section 7.7.3 that going from a single pressure steam cycle to a dual pressure cycle improves efficiency by about 3%, and reduces the entropy losses of the heat transfer by about half, therefore further improvements to heat transfer efficiency are only likely to improve combined cycle efficiency by about 1%.

Steam cycles of the future are likely to include more extensive use of reheat and small amounts of regeneration as exhaust temperatures increase. Steam cycles are unlikely to have more than one reheat and three regenerative feedheaters.

Fuel Cell

The fuel cell technology has been developing in laboratories for many years. It promises high efficiencies in a cycle that is not a heat engine and therefore not subject to the limitations of temperature. Currently two main technologies are being researched, the molten carbonate and solid oxide methods [11]. The solid oxide method is projected to be used in place of the combustor in a gas turbine of a combined cycle, figure 8.12. The fuel cell combustion products gives a gas turbine TET of about 1475 K. This would be used with a low pressure ratio GT so as to allow a reheat steam cycle. This offers a potential efficiency on natural gas of about 60%. On coal gasification the potential efficiency is more like 48%.

Figure 8.12 Fuel Cell Integrated Combined Cycle



8.6 FUTURE PREDICTIONS

Long term predictions about new as yet undiscovered technology is possible if viewed with appropriate caution. It is best followed up by suggestions of how the goal might be achieved.

Rolls Royce suggest in less than 20 years a composite metal aero-engine with stoichiometric TET (about 2500 K) and pressure ratio of 30 will be available [48]. They predict the improvement in performance to come 50% by materials and 50% by component efficiency improvement. It is

suggested that compressors in fighters will use fibre reinforced titanium aluminides operating up to 900 °C. Turbine disks are suggested to be of nickel aluminides or nickel matrix composites for up to 800 °C. Turbine blades will be using thermal barrier coatings. No mention is made of emission problems.

Gas turbines have not reached a plateau, and significant improvements can be expected in the future. But the introduction of new technology is a slow process, requiring extensive testing and validation before introduction into service. Engine development is very expensive and has a high level of risk, so is only justified by significant market forces. The engine must then go through a trial period of loss making with launch customers before significant sales can be expected. And new development must be funded from existing engine sales putting a limitation on funds. Recently a third generation of engines has been launched and these engines will be the basis of the next decade or two of power generation.

Figure 8.9 indicated the future direction of combined cycles is for modest pressure ratios, and high temperatures. This suits use of the current engine designs. Industrial engines will continue to use improvements from the aero-engine field. Within the current engines improvement may include more use of directionally controlled casting techniques, processes like hot isostatic pressing, and thermal barrier coatings. Major improvement within existing engine frames may include closed circuit steam cooling in combined cycles.

There are a number of difficulties in predicting the expected changes due to unknown proprietary information for the current engines. But using the simple model developed in section 8.4.1 the effect of some changes are listed in table 8.4.

Table 8.4 Effect of Parameter Changes

Change from reference (frame 9F, TRIT 1533 K)	Gas Turbine efficiency %	Potential Max. Combined Cycle efficiency %
Reference	34.1	53.3
TRIT up 50 K up 100 K	34.6 35.0	54.2 55.1
Polytropic efficiencies of comp/turb up 1%	35.2	53.8
10 % reduction in cooling	34.4	53.5
Reduce cooling to zero	36.2	55.6
TET up 100 K and 1% on polytropic efficiencies	36.0	55.6
TET up 200 K, no cooling, 1% on comp. and 2% on turb. efficiency	38.9	58.6

In table 8.4 the simple cycle gas turbine efficiencies are not so useful for generalisation as they are for a fixed pressure ratio and some current engines give higher efficiencies for similar temperatures due to higher pressure ratios, but in a combined cycle the efficiency is nearly independent of pressure ratio. The combined cycle efficiencies are for idealised and optimised dual pressure bottoming cycles and in reality the actual efficiency might be lower due to compromises in HRSG cost.

Using the past trend of TET increasing 9 to 11 °C/yr the improvement over the next decade should be about 100 K. As the temperature increases the problem with NO_x will be more severe and temperatures may stagnate near 1700 K.

Using a temperature improvement of 100 K and component polytropic efficiency improvement of 1%, a combined cycle based on the frame 9F has a potential for efficiency about 55 %. With steam cooling and a temperature of near 1700 K the combined cycle efficiency may be near 58%.

Even without all these improvements efficiencies on natural gas should reach about 60% when the fuel cell is established, and with coal gasification fuel cell technology should reach about 48% efficiency. The current projected time for commercialisation of the fuel cell is the late 1990's.

9 PROCUREMENT

9.1 INTRODUCTION

Design features in construction of power plants, such as component redundancy and layout, can increase reliability, availability and maintainability (RAM). Often the gas turbine is a 100% engineered product by the manufacturer and the manufacturer resists changes/options not in his model planning. But standardised plant will not necessarily be the best plant for the customer. As maintenance and availability costs are likely to be more than the purchase cost over the lifetime of the plant it is important to get the procurement right. The topic is too large for this thesis except to give some introduction, and the emphasis here is more on selection of the type of plant than subsequent purchase details.

9.2 SELECTION CONSIDERATIONS

The operating environment will effect the selection process such as fuel price, operating hours, part load usage, number of starts and ambient conditions. The following is a list of the basic equipment and systems that will need consideration.

Gas Turbine Basic equipment	Bottoming cycle basic equipment
Gas Turbine	Heat Recovery Steam Boiler
Possible Gear Box	Steam turbine
Generator	Generator
Transformer	Transformer
Filter system	Condenser
Soundproofing	Cooling water system
Exhaust ducting and stack	Deaerator
Fuel system	Feed water & condensate pumps
Oil system	Water treatment system
Cooling system for generator and oil	Oil, cooling etc. as in GT
Firefighting	Control and Monitoring
Control & monitoring system	
Start-up equipment	
Lifting equipment	
& other maintenance systems	

Variations on these equipment are available, some of the major ones are listed as follows;

Gas Turbine

- Aero-derivative or heavy industrial
- Outdoor or indoor
- Fuel type
 - gas - gas regulating station, heating, filtering and pressure reduction
 - oil - light and heavy distillates, crudes and residuals, heating, water washing and additive systems
 - dual fuel - fuel switch-over
 - coal - gasification or fluidised bed
- Dry, steam or water injection
- 50 or 60 Hz

- Possible synchronous condenser operation
- Filtering and Inlet
 - Snow hoods and anti-icing
 - Inertial filters
 - High efficiency filters, with or without prefilters
 - Self cleaning filters
- Water or air cooling for oil and generator
- Black-start generator or mains powered auxiliary transformer starter

Combined Cycle

- Exhaust bypass stack for simple cycle operation
- Fired or unfired HRSG
- If supplementary fired, then for peak-load or base-load, and for same fuel or a different fuel
- Reheat or non-reheat
- Natural or forced circulation
- Single or dual pressure HRSG
- Triple pressure HRSG with steam injection
- Feedheating bleed steam from turbine or HRSG
- Open circuit water, wet cooling tower or air cooled condensers

Fuels;

The fuel type is very important. Natural gas is ideal, it is clean, requires little treatment or handling, has the lowest maintenance costs and highest efficiency. Heavy oils are hot burning, contain contaminants and require treatment. They can reduce availability due to contaminants and will require derating. Heavier oils like No. 6 produces ash and needs soot blowers in the HRSG for base-load use. Coal gasification, requires large additional gasification equipment and the overall fuel to electricity efficiency is lower.

Reheat vs non-reheat;

Reheat maximises the steam cycle efficiency. It is not common as only some GT's produce sufficiency exhaust temperature without supplementary firing, but it can give an improved efficiency of about 1 to 1.5% and a longer life for the low pressure steam turbine blades.

Supplementary firing;

This only gives gains for low exhaust temperature turbines, but can burn poor fuels and allow reheat steam cycles where they would otherwise be impractical. This is more likely to be used in boosting power in peaking installations, matching repowering situations or varying power to heat ratio in CHP situations.

Single vs multiple pressure Steam Turbine;

Dual pressure maximises efficiency by allowing a much reduced stack temperature, giving about 3% more efficiency. It also allows the deaerator steam to be sourced from boiler and not the turbine. It involves much added complexity of pumps and level controls, and much increased HRSG heat transfer surface size and therefore cost. Single pressure combined cycles are quicker starting and simpler to maintain in hot stand-by condition.

High vs Low Pressure Steam;

Higher pressures improve efficiency in dual pressure cycles, but pressures may need optimisation in single pressure cycles. Generally higher dual pressure cycles are used in base-load.

Cooling Water;

The cooling method effects the back pressure in the condenser hence the cycle efficiency. The best method is using river or sea water, but wet cooling towers and dry air coolers are possible.

Ambient Conditions;

The range of ambient conditions will effect performance greatly. It will effect the sizing of the steam turbine relative to the gas turbine, limited by cold days, and the allowable wetness in the LP stages of the steam turbine on hot days.

HRSG;

The boiler is likely to be custom designed for the specific conditions. Pressure loss, pinch and approach points in the boiler will need optimisation for the operating conditions and load cycle.

The type of arrangement likely to be used is summarised in table 9.1.

Table 9.1 Generalised Plant Configuration

	Peaking	Base-load
Steam pressure	Intermediate	High
No. of pressures	Single	Dual or Triple
Steam reheat	No	Yes and No
Fuel	Natural gas or light distillates	Natural gas to heavy residual

The number of engines in the plant will effect availability and part load efficiency greatly. A two engine installation can regain good efficiency at half load, and mean higher availability of power. A common arrangement is a 2+1 block with two gas turbines and one steam turbine.

9.3 AERO-DERIVED vs HEAVY INDUSTRIAL

The aero-derived and heavy industrial engines each have there own advantages and disadvantages. They are compared here with generalisations on their features.

Aero-derived engines;

- Smaller unit sizes, up to 36 MW
- Smaller and lighter per unit output, easier to transport and possible air freight of modules.
- Small enclosure with less crane and maintenance facilities
- More expensive in \$/kW for basic engine
- Quicker starting, in the order of two minutes fast start and five minutes normal start
- Modular or semimodular construction possible, with replacement by spare/refurbished modules or rental engines
- Maintenance actions based on on-condition and allowing module replacement in short times and off-site overhaul
- Higher potential availability, over 98% with spare engines
- Higher pressure ratio, thus higher fuel pressures needed
- Often higher TET and more sophisticated cooling (often 20% of flow)
- More maintenance implied by sophistication and high TET

- Benefits from aircraft development funded separately
- Higher efficiency in gas turbine requires a relatively smaller steam turbine but lower exhaust temperature may require more HRSG surface area to obtain competitive efficiency
- Free turbine and twin spool available, giving better part load efficiency, but tighter control over frequency and speed required
- Rolling contact bearings
- Thin casings and hollow shafts
- Less robust
- Less tolerant to heavier fuels, and unable to easily burn residuals or crudes due to smoke formation
- Low starting power
- Low start-up losses, less than 80 MJ/MW

Heavy Industrial;

- Heavy and large per unit output
- Available in large unit sizes up to 200 MW
- Less expensive in \$/kW for basic engine
- Slower starting, typically 10 to 20 minutes
- Start-up losses larger, about 800 MJ/MW
- Lower pressure ratio, thus lower fuel pressure required
- Lower efficiency in gas turbine requires a relatively larger steam turbine, but higher exhaust temperatures may require less HRSG surface area or even a simpler cycle
- More tolerant to heavy fuels
- Combustor generally larger and less likely to smoke
- Often lower TET and less cooling flow (typically 10%)
- Emission control easier
- Single shaft configuration normal thus more stable frequency control
- Small sized engines with single shaft configuration may need a gearbox reduction to the generator with associated losses
- Larger starting power required
- Longer time between maintenance actions
- Maintenance often on site, and involves extended downtime
- Potential availability lower

9.4 SINGLE SHAFT vs FREE TURBINE

The aero-derivatives are inevitably free turbine engines, while larger heavy industrials are single shaft and designed for power production. Small industrials usually come in both configurations, as a single shaft for power production and two shaft for pumping and mechanical drive applications.

A free turbine engine obtains its power variation by varying the gas generator speed. This means that power changes are inherently slow and dependant on rotor inertia, surge margin and over temperature limits. On the other hand a single shaft engine has a high rate of load change possible once it is synchronised, as there is no change of inertia, and it has a higher resistance to speed change as the inertia includes the compressor, extra shafting and extra turbines. Therefore a single shaft GT is more speed stable easier to regulate. But at part load a two shaft engine gives much improved performance.

A single shaft engine requires more starting power as the entire engine and generator must be rotated, as opposed to a two shaft engine where only the gas generator needs powering. A twin spool engine needs even

less power to start as only the HP spool is powered. This is advantageous for black start purposes.

A two shaft turbine allows the power turbine to rotate at lower speed, hence even a small aero-derivative engine can run the power turbine at 3000 or 3600 rpm without gearing. Industrial single shaft engines in smaller size ranges need gearing for these low generator speeds, with associated gear box losses.

9.5 SPECIFICATION AND TENDERING

Manufacturers are generally immovable in the fundamental engine design, but pressure from the potential customer can normally cause a departure from the standard designs in auxiliaries and layout. Without detailed specification the customer will get a standard package based on minimised piping, wiring and satisfactory performance until the end of say a 12 month guarantee. But with appropriate specification writing a better package can be obtained, improving maintainability and guarantees can be extended on specific components to periods more in keeping with the life claims.

Details of specification writing cannot be dealt with in a thesis of this size. Details range from the small points like no flood lighting near intakes to requirements for minimum space around engines for access, and redundancy for on-line maintenance of auxiliaries. Requirements must be formalised to enable checking, and methods of measuring agreed before hand.

Many references give guidelines on procurement. Also an EPRI report "Guidelines for procuring high availability utility gas turbine systems" was due out in 1988, project RP2915-2. Other EPRI references of interest are [274] and [275], and some procurement standards are [130] and [131].

9.6 CONSTRUCTION

Construction times of the gas turbine depends on the engine. Some manufacturers keep engines 'on the shelf' and can give very short delivery times for the gas turbine. Steam turbines are generally made to order and have a longer delivery. Typical construction times were given in section 2.5. A typical construction schedule is given in figure 9.1. Aero-derivatives are light and smaller, and require less foundation work, and so are easier to transport and erect. Heavy industrials may weight hundreds of tonnes, and require shipment in pieces.

9.7 PRICES

The growing electricity demand, especially in the US, has put an upward pressure on gas turbine prices. The prices vary widely depending on how much the manufacturer wants the contract. In some instance a manufacturer will accept a loss to establish a presence in a new area or with a new engine.

Example equipment prices for the basic gas turbine only, for June 1988 in US dollars are as follows [22]. These are based on immediate shipping for a single unit, and the cost escalates at about 5% per year for delayed shipment.

Large industrials;	PG9281F	202.3 MW	\$154/kW
	MW701	126.9 MW	\$169/kW
	GT 13	100.5 MW	\$174/kW
	GT 8	48.8 MW	\$240/kW
Aero-derivatives;	Tornado	6.2 MW	\$323/kW
	LM5000PC	33.1 MW	\$339/kW
	RB211	26.1 MW	\$337/kW
	LM2500PE	21.2 MW	\$367/kW

It can be seen the larger the plant the cheaper in terms of \$/kW. Aero-derivatives are more expensive than heavy industrials, but may require less complex civil engineering and cheaper transport. The basic gas turbine cost represents less than 50% of the total installation cost. Turnkey installations also include costs for design, transport, erection, commissioning and training. Installation may also include buildings, roads, fuel storage, maintenance facilities and other extras. An approximate rule of thumb for a simple cycle generator package is the costs are 50% gas turbine, and 50% for the electrical generator and it's auxiliaries.

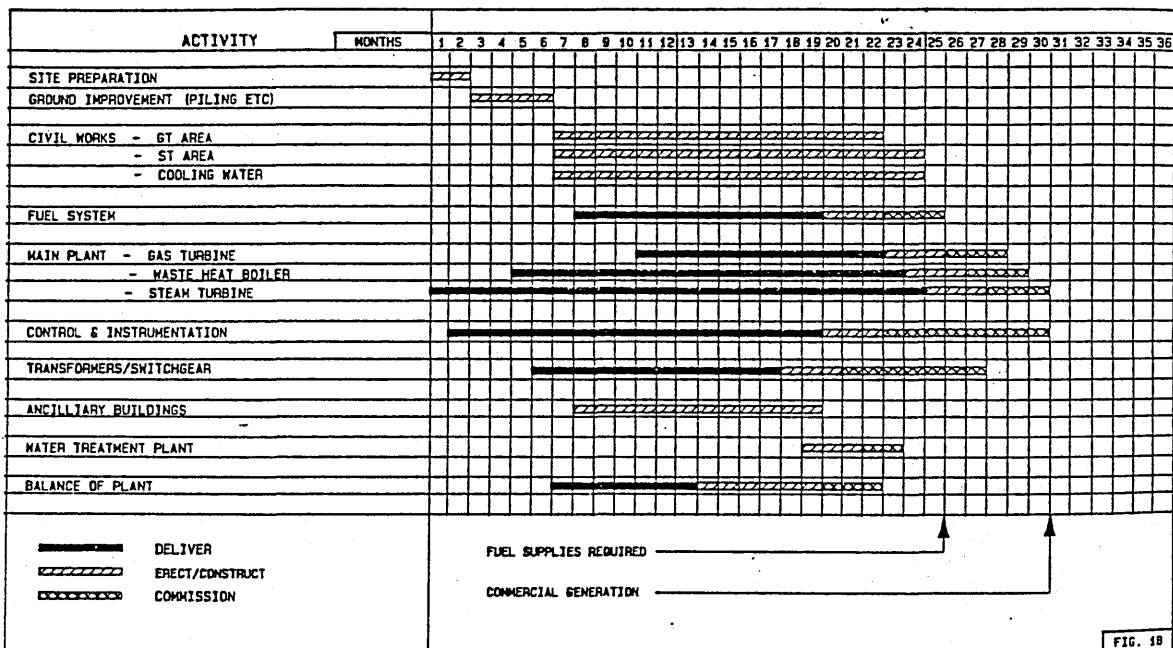
An example breakdown for a 20 MW Cogeneration system is as follows [22];

- Gas turbine, generator, backup fuel storage and auxiliaries:	7 800 000	(47%)
- HRSG, water treatment and pumps:	2 300 000	(14%)
- Transformer, switch gear and interconnection:	800 000	(5%)
- Services and installation:	4 400 000	(26%)
- Contingency 10%	1 500 000	

Total \$US 16 800 000

Examples like this have limited usefulness as each situation has different costs, but it does serve to give an indication of the relative size of areas of cost.

Figure 9.1 Typical Construction Schedule



10 CONCLUSIONS AND SUMMARY

Combined cycles have been investigated to some depth, but details have often been brief because of space and time limitations. Emphasis is more on gas turbines and heat recovery boilers than the steam turbine as the gas turbine dominates the combined cycle and is a less mature technology. The HRSG is discussed in depth as it dominates the steam cycle and dictates the bottoming efficiency.

Gas turbines are an established form of power production, and are predicted to form more than 50% of future new plants. After the oil shock the market slumped and much development was slowed, but currently a third generation of engines are being produced. Reliability problems were addressed in the 1970's and currently high availability is possible of over 90% with forced outage rates of 2 to 4%.

Utility objectives are to produce lowest cost power at an acceptable level of reliability. This means low production costs such as efficiency, fuel type and maintenance, but also low transmission costs such as small unit size, flexibility in location, and high availability and reliability. Due to the high cost of capital and the uncertainties of predicting future demand small unit size, short construction periods and staged construction are attractive features. Selection features also include electrical performance, load change rates, starting times, part load efficiency, safety and life.

Gas turbines and combined cycles have advantages of small unit sizes, high efficiency, short starting times, prepackaged, short construction times, minimised on site construction, low water requirements, low emissions and can use a wide variety of fuels. A gas turbine efficiency is typically between 30 and 36%, while a combined cycle efficiency is typically between 45 and 51%. But these efficiencies are while new and average life efficiencies could be slightly lower. Conventional steam cycles are about 36 to 38% efficiency.

Gas turbines come in a variety of configurations. The simple cycle is a cycle that sequentially compresses, heats and expands the working fluid. Most engines operate on air. Simple cycles can be two and three shaft, giving different part load and transient characteristics. The cycle wants the coldest temperatures for compression and the hottest temperatures for expansion. The peak temperature for work removal is the turbine rotor inlet temperature, and is limited by material strength. Cooling flow bleed from the compressor is used to increase the peak temperatures, but then a compromise is made between reduced performance due to cooling flow and increased performance due to temperature.

A reheat cycle uses a second combustor part way through the expansion to increase the temperatures of work extraction. This improves the specific power but without exhaust heat recovery reduces efficiency. In a combined cycle reheat would improve power and efficiency, including part load efficiency, but is as yet to be developed commercially.

Intercooling is reducing the temperature of the working fluid part way through compression. It increases efficiency of gas turbines, but in combined cycles it has to be carefully optimised to prevent reduction of efficiency. It does however allow a greatly increased pressure ratio and reduced cooling flow. Combined with gas turbine reheat, intercooling

offers excellent part load performance in a combined cycle.

Closed cycles are used but they have not been widely established. They can have near flat efficiency characteristics down to about 30% power output, and a variable ratio of power to heat in CHP applications. But due to the limited peak temperature of the heat input heat exchanger they have a low design point efficiency. The cost is also excessive at about twice conventional steam cycles.

The gas turbine performance is dependent on the peak temperature, or TET, and the pressure ratio. Ideally the efficiency is only a function of the pressure ratio, but in reality the efficiency is reduced by component inefficiencies, combustor pressure loss and cooling flows. Therefore the efficiency improves with TET and an optimum pressure ratio exists for maximum efficiency. The optimum for maximum specific power is at a lower pressure ratio than that for maximum efficiency. In a combined cycle the efficiency is a much reduced function of pressure ratio and nearly independent over a wide range. Low pressure ratio gas turbines can allow higher combined cycle efficiency due to more complex bottoming cycles.

Real engines have a compromise between maximum temperature, engine life, cooling flow, cost, pressure ratio and size. Pressure ratios are higher on aero-derivatives where they are 15 to 30, whereas heavy industrials use pressure ratios of 10 to 15. Current turbine maximum metal temperatures are about 1090 K. Turbine rotor inlet temperatures are mostly 1200 to 1500 K. Traditionally the aero-derivatives use higher temperatures, but recently the new third generation of heavy industrial engines are using complex cooling and temperatures comparable with aero-derivatives.

Increases in ambient temperature reduces an engines performance significantly. The actual rated engine performance will be some average appropriate for the life and ambient temperatures expected. The engine also reduces in power for reduced ambient pressure and is de-rated for operation at altitude. Inlet and exhaust pressure losses also reduce the engine performance.

The gas turbine is rated for the expected conditions and adjusted for losses and auxiliary power consumptions. To give the ability to compare engines they are quoted at ISO ambient conditions, and often for ISO loading cycles. But still confusion is possible due to what is included in the output, whether it includes inlet/outlet losses, auxiliaries, generator losses and effects of fuel type.

The fuel type has a big impact on engine performance and maintenance. Most fuels are hydrocarbons. The best fuel is sulphur free natural gas. Light distillates are good fuels but as the carbon content and density increases the quality also decreases. Heavy distillates and residuals can be burnt but require cleaning, and treatment. Many of the problems with contaminants occur mainly in heavy fuels.

Hot corrosion and fouling are common problems with heavy fuels. The usual way to combat these problems is to water wash the fuel to remove water soluble contaminants such as sodium. The fuel is then treated with additives such as magnesium sulphate for vanadium contaminants and with silicon dioxide for reducing fouling.

Emissions are currently a very important issue. The requirements for low CO and unburnt fuel are opposite to requirements for low NO_x, so some compromise is made. Water injection is the established method for NO_x control but currently several manufacturers are using dry low NO_x combustors.

Coal can be utilised as a fuel, by gasification or by integrating combustion into the cycle, but it is currently not widely used. It is a proven technology, not awaiting any technological advancement or even acceptance by the utility community, but awaiting increases to natural gas prices. Overall plant efficiency from coal pile to buss bar are expected to be in the range of 38 to 42%. A German development is aiming at coal efficiencies of 47%. This is as or more efficient than conventional steam plant, producing less emissions and including flue gas desulphurization cheaper than convention steam plant.

Maintenance of gas turbines is a very important issue for the utility. The annual expenditure for routine maintenance could be 10 to 20% of the turbines initial price. But maintenance practices vary so widely between companies that it is not possible to state a standard practice. The aim of maintenance is to minimise the long term costs for the utility which translates essentially to reducing downtime and maintaining performance. It is usual that the cost of downtime, including stand-by system capacity is the overriding cost and minimising the outage time is the first priority. This can be achieved from simple ways through proper training and tools, and through attention to detail at procurement.

Typically few turbines achieve their full design potential due to erosion, damage, corrosion and fouling. And many of the problems that can occur are directly effected by the level of air filtration or fuel specification. It is vitally important that appropriate filtration is used if good availability and low maintenance costs are required.

Approximately 70% of performance loss is due to compressor blade fouling. This can be recovered by water washes, either at reduced load or off-line. Off-line soak washes are the most effective. But some losses are permanent and washing does not usually return the engine to original condition. Washing intervals vary but might be say 250 hours for on-line wash to 1000 hours for off-line.

Corrosion problems can be controlled with modern coating technology, and there should be no reason for scheduled turbine overhaul based on corrosion life. But no coating is totally resistant to corrosion and the coat degrades.

Ideal availability for a heavy industrial gas turbine, only stopped for routine inspection, is about 98%. The routine outages range from 1 day at intervals of about 4000 hours to about 15 days at intervals of about 40000 hours. Forced outage factors can be as low as about 2 to 4%, but vary considerably. An outage due to turbine blade damage can be in excess of 80 days. As the figures can be defined in different ways, and could cover periods without major maintenance, availability figures are only useful when viewed in light of other data. Aero-derivatives can have very high availabilities as they use a modular construction with possible replacement of an entire module with a refurbished or rental module and later repair. Availability and reliability can be significantly improved by careful specification and planning.

Combined cycles achieve a high efficiency because the gas turbine starts to convert heat to work at relatively high temperatures while the steam turbine can continue the process to near ambient temperatures. The main problem is to match the cycles and build a boiler to minimise temperature degradation without being too large and complex. To transfer the heat from the exhaust gas to the steam requires a temperature difference and this implies a reduction in the quality of the energy. Typically combined cycles have low steam conditions relative to conventional steam cycles as the gas turbine exhaust gas is not hot enough to sustain advanced steam cycles.

Combined cycles can be produced from repowering an existing steam plant, by replacing aging boilers with gas turbines and HRSG's. Typically the older steam cycles with low steam conditions are suitable for repowering, the boilers may be aging while the steam turbines are still in good condition, and as a rule the space created by the removal of existing boilers is sufficient for the gas turbines and new HRSG's. The high steam conditions of more recent steam plant might not be so suitable for repowering without supplementary firing.

Cogeneration is where both heat and power are produced in the same plant to improve the fuel utilisation efficiency. The power is usually electricity and the heat steam. To be more economic than separate production of power by the electricity utility and steam by a fired boiler the ratio of power to steam must be maximised. This may require some export of power from the industry to the national grid.

A single shaft combined cycle has the steam turbine and gas turbine on the same shaft using the same generator. It has the highest efficiency of the various configurations, mainly because the electricity generator efficiency improves with size. As the generator represents about 50% of the costs of a GT generator set it is very much cheaper if only one generator is used. The control is simpler as the steam turbine speed is controlled by the gas turbine.

Power plant with multiple engines on separate shafts have very much improved flexibility of operation as a gas turbine can be stopped and yet allow the others to continue supplying power and steam for the steam turbine. The part load efficiency is also greatly improved, for instance with a 2 + 1 block near design point efficiency is regained at 50% load when a gas turbine is shutdown. Multi-engine installations also allow staged construction as the utilities power requirements grow.

The combined cycle performance can be improved in many ways. Most of the losses in the bottoming cycle are due to the temperature differences in the HRSG, thus most improvements are related to improving the temperature profile of the steam in the boiler. For gas turbines with low exhaust temperature supplementary firing of fuel in the gas turbine exhaust can increase the efficiency slightly, especially if it includes a more complex steam cycle. But it is more normal to use a two pressure steam cycle. A two pressure cycle increases the efficiency of the combined cycle by typically 3% by reducing heat transfer losses by half, but at significantly more capital cost. Reheat on the steam cycle is only possible with high GT exhaust temperature, or with supplementary firing and can add about 1% to the efficiency. Steam cycle regeneration only improves the combined cycle efficiency for high exhaust temperatures and then only by small amounts.

Part load behaviour of plain combined cycles is similar or even worse than gas turbines. The part load efficiency is improved by using a two shaft gas turbine, but most combined cycles are large single shaft engines. Part load improvements can be achieved by using variable inlet guide vanes and even variable turbine nozzles to reduce the mass flow and increase the temperatures. Also the pressure of the steam turbine is reduced at part load to improve heat recovery, which is called 'sliding pressure'. With these improvements the efficiency may only reduce by 1% over the 100 to 80% range, but then with rapid fall off.

Improvements in combined cycle efficiency will come more from improved temperatures and reduced cooling than from increased pressure ratio. With current blade efficiencies a 100 °C improvement in TRIT without increase in cooling will increase combined cycle efficiencies by about 4% (up 2%). It is reasonable to assume improvements in the next 10 years will give cycle efficiencies of 55% to 60%. The turbine entry temperature increases may stagnate at 1700 to 1800 K due to NO_x emissions.

Large gains in performance are expected with use of gas turbine reheat in a combined cycle, but as yet this is not available commercially. Other potentially large gains are with closed circuit steam cooling of the gas turbine. It is suggested that using closed circuit steam cooling with current temperatures and component efficiencies the combined cycle efficiency could be 53% and 10% less capital cost. With a 200 K improvement in temperatures the efficiency could be 57%. The combination of a fuel cell in place of a gas turbine combustor in a combined cycle engine has the potential of 60% efficiency on natural gas with a TET of only 1475 K. The main problems with each of these proposals are the tremendous development costs and development time.

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APPENDIX A — SPREADSHEET MODELS

The spreadsheet used is the "SMART" system from Innovative Software. It uses the r14c3 format for row 14 column 3, unlike the Lotus 123 method of C14. The spreadsheet also recalculates in the correct sequence even though some formulae are placed in convenient groupings and not calculation order.

A.1 Design Point Calculations

The calculation is purposely broken into small steps, rather than collecting several calculations into one equation. This is the most flexible approach and allows all the results to be observed. All calculations are arranged into one column so that the column can be "copied right" for multiple calculations, to observe trends.

Spreadsheet Layout

	Col.= 1	Col.= 2	Col.= 3	Col.= 4	Col.= 5
Row = 1	Design Point Calculations - Single Pressure Steam Cycle				
Row = 2	INPUTS				
Row = 3					
Row = 4					
Row = 5	GAS TURBINE	Ambient	Temperature	°C	15.0
Row = 6			Pressure	kPa abs	101.3
Row = 7		Intake	P losses	%	1.0%
Row = 8			Air flow	kg/s	150.0
Row = 9		Compressor	P ratio	-	12.0
Row = 10			Isen. efficiency	%	86.0%
Row = 11		Combustion	P losses	%	5.0%
Row = 12			Fuel LCV	kJ/kg	42000
Row = 13			Efficiency	%	100.0%
Row = 14		Turbine	TET	K	1400.0
Row = 15			Isen. efficiency	%	88.0%
Row = 16			Cooling flow	%	5.0%
Row = 17			Cooling split	%	66.0%
Row = 18		Shaft	Efficiency	%	99.5%
Row = 19	SUPPLEMENTARY FIRING		Fuel Ratio	-	0.0
Row = 20	HEAT EXCHANGE	Exhaust	P losses	kPa	2.0
Row = 21		Steam drum	Pressure	MPa abs	6.0
Row = 22		Pinch point	T drop min.	°C	33.0
Row = 23		Exhaust	T minimum	°C	150.0
Row = 24		Evaporator	Feed heating	°C	10.0
Row = 25	STEAM TURBINE	Throttle	P losses	%	5.0%
Row = 26			T max.	°C	530.0
Row = 27		Turbine	Isen. efficiency	%	85.0%
Row = 28		Condenser	Pressure	kPa abs	8.5
Row = 29	FEED	Pump	Isen. efficiency	%	80.0%
Row = 30	GENERATOR	Generator	Efficiency	%	98.7%
Row = 31	CONSTANTS	Constants	Cp cold air	J/kg K	1005.0
Row = 32			R air	J/kg K	287.0
Row = 33	RESULTS				
Row = 34					
Row = 35					
Row = 36	Gas Turbine	Fuel mass flow		kg/s	3.19
Row = 37		Power output		MW	45.6

	Col.= 1	Col.= 2	Col.= 3	Col.= 4	Col.= 5
Row = 38		Efficiency	%		34.0
Row = 39		Exhaust temperature	°C		559
Row = 40	Supplementary	Firing Temperature	°C		559
Row = 41		Fuel mass flow	kg/s		0.00
Row = 42	HRSG	Effectiveness	%		72.1
Row = 43		Heat recovery efficiency	%		68.5
Row = 44		Steam flow rate	kg/s		19.7
Row = 45		Exhaust Stack Temperature	°C		186
Row = 46	Steam turbine	Power output	MW		21.5
Row = 47		Bottoming cycle efficiency	%		24.3
Row = 48		Steam Cycle efficiency	%		33.0
Row = 49		Quality of exit steam	%		92.4
Row = 50	Condenser	Heat rejection rate	MW		43.7
Row = 51		Feed temperature	°C		42.4
Row = 52	Overall	Power export	MW		66.2
Row = 53		Combined Cycle Efficiency	%		50.0
Row = 54	Check	Correlation error messages =>	Inlet		OK
Row = 55			Exit		OK

CALCULATIONS

Row = 59	GAS TURBINE	Intake	T outlet	K	288.150
Row = 60			P outlet	kPa	100.312
Row = 61		Compressor	P outlet	kPa	1203.741
Row = 62			Exp (k-1)/k	-	0.286
Row = 63			T ratio ideal	K	2.033
Row = 64			T rise ideal	K	297.722
Row = 65			T rise actual	K	346.188
Row = 66			Power required	MW	52.188
Row = 67			T outlet	K	634.338
Row = 68		Bleed	Cooling flow	kg/s	7.500
Row = 69		Combustor	Mass flow inlet	kg/s	142.500
Row = 70			T rise required	K	765.662
Row = 71			Fuel air ratio	-	0.022
Row = 72			Fuel flow	kg/s	3.192
Row = 73			P outlet	kPa	1143.554
Row = 74		Fixed Turb.	Mass flow in	kg/s	150.642
Row = 75			T inlet	K	1373.403
Row = 76			Work required	MW	52.450
Row = 77			Cp gas	J/kg K	1238.415
Row = 78			Exp (k/k-1)	-	4.315
Row = 79			T drop actual	K	281.147
Row = 80			T drop ideal	K	319.485
Row = 81			P ratio	-	3.135
Row = 82			P outlet	kPa	364.809
Row = 83			T outlet	K	1092.256
Row = 84		Power Turb.	Mass flow inlet	kg/s	153.192
Row = 85			T inlet	K	1084.505
Row = 86			P outlet	kPa	103.325
Row = 87			P ratio	-	3.531
Row = 88			Cp gas	J/kg K	1177.746
Row = 89			Exp (k-1)/k	-	0.244
Row = 90			T ratio ideal	K	1.360
Row = 91			T outlet ideal	K	797.491
Row = 92			T drop ideal	K	287.014

Col.= 1	Col.= 2	Col.= 3	Col.= 4	Col.= 5	
Row = 93		T drop actual	K	252.572	
Row = 94		T outlet actual	K	831.933	
Row = 95		Power output	MW	45.569	
Row = 96	SUPPLEMENTARY FIRING	Fuel flow	kg/s	0.000	
Row = 97		Cp gas	J/kg K	1124.706	
Row = 98		T rise	K	0.000	
Row = 99		Mass flow outlet	kg/s	153.192	
Row =100		T outlet	K	831.933	
Row =101	STEAM CYCLE	Feed pump	h rise ideal	kJ/kg	6.021
Row =102			h rise actual	kJ/kg	7.527
Row =103			h inlet	kJ/kg	185.243
Row =104			T feed inlet	K	315.582
Row =105		HRSG	T steam drum	K	548.790
Row =106			T steam out	K	798.933
Row =107			Cp water sat.	J/kgK	5238.381
Row =108			hfg sat.	kJ/kg	1568.219
Row =109			h outlet	J/kg	3496.136
Row =110		Case 1, Evaporator limiting			
Row =111			T gas av	K	706.861
Row =112			Cp gas av	J/kgK	1098.441
Row =113			T steam av	K	673.861
Row =114			Cp steam av	J/kgK	2066.313
Row =115			Flow steam (1)	kg/s	19.692
Row =116			T gas av	K	502.470
Row =117			Cp gas av	J/kgK	1055.519
Row =118			Cp water av	J/kgK	4500.000
Row =119			T gas outlet (1)	K	459.464
Row =120		Case 2, Exhaust T limiting			
Row =121			T gas outlet (2)	K	423.150
Row =122			T gas av	K	627.541
Row =123			Cp gas av	J/kgK	1081.784
Row =124			Q from gas	MW	67.744
Row =125			Flow steam (2)	kg/s	20.461
Row =126		Choice, case 1 or 2			
Row =127			Flow steam act.	kg/s	19.692
Row =128			T outlet actual	K	459.464
Row =129			€ HRSG	-	0.721
Row =130		Throttle	P outlet	MPa	5.700
Row =131			s outlet	kJ/kgK	7.017
Row =132		Steam Turb.	hf outlet P	kJ/kg	177.717
Row =133			hg outlet P	kJ/kg	2579.168
Row =134			sf outlet P	kJ/kgK	0.609
Row =135			sg outlet P	kJ/kgK	8.206
Row =136			x outlet ideal	-	0.844
Row =137			h outlet ideal	kJ/kg	2203.445
Row =138			h drop ideal	kJ/kg	1292.691
Row =139			h drop actual	kJ/kg	1098.787
Row =140			Power output	MW	21.638
Row =141			h outlet	kJ/kg	2397.349
Row =142			x outlet actual	-	0.924
Row =143		Condenser	h drop	kJ/kg	2219.633
Row =144			Q rejected	MW	43.710
Row =145		Pump work	Power input	MW	0.148
Row =146					

Formula

Row	Column	Formula
36	5	r72c5
37	5	r95c5
38	5	r37c5/r36c5/r12c5*100000
39	5	r94c5-273.15
40	5	r100c5-273.15
41	5	r96c5
42	5	r129c5*100
43	5	(r94c5-r128c5)/(r94c5-r5c5-273.15)
44	5	r127c5
45	5	r128c5-273.15
46	5	r140c5-r145c5
47	5	r46c5*1000/((r36c5+r41c5)*r12c5-r37c5*1000)*100
48	5	r46c5*1000/(r44c5*(r109c5-r103c5))*100
49	5	r142c5*100
50	5	r144c5
51	5	r104c5-273.15
52	5	(r37c5+r46c5)*r30c5
53	5	(r37c5+r46c5)*1000/((r36c5+r41c5)*r12c5)*100
54	5	if r21c5>8 or r21c5<3 or r106c5>973 or r106c5<573 then "Throttle conditions out of range" else "OK"
55	5	if r28c5>15 or r28c5<3 then "P condenser out of range" else "OK"
59	5	r5c5+273.15
60	5	r6c5*(1-r7c5)
61	5	r60c5*r9c5
62	5	r32c5/r31c5
63	5	r9c5^r62c5
64	5	r63c5*r59c5-r59c5
65	5	r64c5/r10c5
66	5	r65c5*r31c5*r8c5/1000000
67	5	r59c5+r65c5
68	5	r8c5*r16c5
69	5	r8c5-r68c5
70	5	r14c5-r67c5
71	5	1.1*(r70c5-50)*(1+r67c5/3250)/r12c5/r13c5
72	5	r71c5*r69c5
73	5	r61c5*(1-r11c5)
74	5	r69c5+r68c5*r17c5+r72c5
75	5	r14c5-r17c5*r68c5/r69c5*(r14c5-r67c5)
76	5	r66c5/r18c5
77	5	950+0.21*r75c5
78	5	r77c5/r32c5
79	5	r76c5*1000000/r74c5/r77c5
80	5	r79c5/r15c5
81	5	(r75c5/(r75c5-r80c5))^r78c5
82	5	r73c5/r81c5
83	5	r75c5-r79c5
84	5	r8c5+r72c5
85	5	r83c5-(1-r17c5)*r68c5/r74c5*(r83c5-r67c5)
86	5	r6c5+r20c5
87	5	r82c5/r86c5
88	5	950+0.21*r85c5
89	5	r32c5/r88c5

Row	Column	Formula
90	5	$r87c5 \cdot r89c5$
91	5	$r85c5 / r90c5$
92	5	$r85c5 - r91c5$
93	5	$r92c5 \cdot r15c5$
94	5	$r85c5 - r93c5$
95	5	$r93c5 \cdot r88c5 \cdot r99c5 / 1000000$
96	5	$r19c5 \cdot r72c5$
97	5	$950 + 0.21 \cdot r94c5$
98	5	$r96c5 \cdot r12c5 \cdot 1000 / r99c5 / r97c5$
99	5	$r96c5 + r84c5$
100	5	$r98c5 + r94c5$
101	5	$0.001005 \cdot (r21c5 \cdot 1000 - r28c5)$
102	5	$r101c5 / r29c5$
103	5	$r102c5 + r132c5$
104	5	$288.33 + 4.017 \cdot r28c5 - 0.0954 \cdot r28c5 \cdot r28c5$
105	5	$455.61 + 19.895 \cdot r21c5 - 0.7275 \cdot r21c5 \cdot r21c5$
106	5	if $r100c5 - r22c5 > r26c5 + 273.15$ then $r26c5 + 273.15$ else $r100c5 - r22c5$
107	5	$41582 - 151.946 \cdot r105c5 + 0.1562 \cdot r105c5 \cdot r105c5$
108	5	$(14461.67 \cdot r105c5 - 671200 - 18.9162 \cdot r105c5 \cdot r105c5) / 1000$
109	5	$632.679 + 5.10687 \cdot r106c5 - 0.00174667 \cdot r106c5 \cdot r106c5 -$ $15.05 \cdot r21c5 - 0.316667 \cdot r21c5 \cdot r21c5$
111	5	$(r100c5 + r105c5 + r22c5) / 2$
112	5	$950 + 0.21 \cdot r111c5$
113	5	$(r100c5 - r22c5 + r105c5) / 2$
114	5	$(7.9406 - 10.188 \cdot (r113c5 / 100) \cdot 0.25 + 4.5934 \cdot (r113c5 /$ $100) \cdot 0.5 - 0.20532 \cdot (r113c5 / 100)) \cdot 1000$
115	5	$r99c5 \cdot r112c5 \cdot (r100c5 - r22c5 - r105c5) / (r114c5 \cdot (r106c5 -$ $r105c5) + r108c5 \cdot 1000 + r107c5 \cdot r24c5)$
116	5	$(r105c5 + r22c5 + r121c5) / 2$
117	5	$950 + 0.21 \cdot r116c5$
119	5	$r105c5 + r22c5 - r115c5 \cdot r118c5 / r99c5 / r117c5 \cdot$ $(r105c5 - r24c5 - r104c5)$
122	5	$(r100c5 + r121c5) / 2$
123	5	$950 + 0.21 \cdot r122c5$
124	5	$r99c5 \cdot r123c5 \cdot (r100c5 - r121c5) / 1000000$
125	5	$r124c5 \cdot 1000 / (r109c5 - r103c5)$
127	5	if $r119c5 > r121c5$ then $r115c5$ else $r125c5$
128	5	if $r119c5 > r121c5$ then $r119c5$ else $r121c5$
129	5	$(r100c5 - r128c5) / (r100c5 - r104c5)$
130	5	$r21c5 \cdot (1 - r25c5)$
131	5	$2.18287 + 0.0113789 \cdot r106c5 - 0.00000533167 \cdot r106c5 \cdot$ $r106c5 - 0.1893 \cdot r130c5 + 0.00694444 \cdot r130c5 \cdot r130c5$
132	5	$63.91 + 16.772 \cdot r28c5 - 0.398 \cdot r28c5 \cdot r28c5$
133	5	$2524.7 + 8.72 \cdot r28c5 - 0.272 \cdot r28c5 \cdot r28c5$
134	5	$0.1951 + 0.0671 \cdot r28c5 - 0.002168 \cdot r28c5 \cdot r28c5$
135	5	$8.8092 - 0.09974 \cdot r28c5 + 0.003384 \cdot r28c5 \cdot r28c5$
136	5	$(r131c5 - r134c5) / (r135c5 - r134c5)$
137	5	$r132c5 + r136c5 \cdot (r133c5 - r132c5)$
138	5	$r109c5 - r137c5$
139	5	$r138c5 \cdot r27c5$
140	5	$r139c5 \cdot r127c5 / 1000$
141	5	$r109c5 - r139c5$
142	5	$(r141c5 - r132c5) / (r133c5 - r132c5)$
143	5	$r141c5 - r132c5$

Row	Column	Formula
144	5	r143c5*r127c5/1000
145	5	r102c5*r127c5/1000

A.2 HRSG Design Point

The above calculations produce design point results but do not give any indication of the physical area of the HRSG heat transfer surfaces or it's off design behaviour. The following is a block of spreadsheet added to the right of the above spreadsheet. It is a recalculation of the above design point but calculating the heat transfer coefficients from the correlations mentioned in section 7.9.1, and thus the areas of the heat exchangers. The physical size and shape of the HRSG is then collected into a number, Γ , and the fluid properties into a number, β . The product $\Gamma\beta$ is used in the later spreadsheet section to calculate off design performance of the HRSG and thus of the steam cycle.

Spreadsheet

Row	Col=7	Col=8	Col=9	Col=11	Col=12	Col=13	Col=14
1	HEAT TRANSFER COEFFICIENTS						
2	RESULTS FROM DESIGN POINT						
3	DATA FOR ECONOMISER CALCULATIONS						
4							
5	mg	153.19 kg/s	INTERNAL	Water at T	427.19 K		
6	θ_1	831.93 K		density	910.00 kg/m3		
7	θ_4	459.46 K		k	0.685 W/mK		
8	mf	19.69 kg/s		μ	1.73E-04 Ns/m2		
9	T1	315.58 K		Pr	1.090		
10	T2	538.79 K		N	20		
11	T3	548.79 K		do	0.025 m		
12	T4	798.93 K		di	0.020 m		
13	P3	6.00 MPa		C	0.023		
14	θ_1 -T4	33.00 K		m	0.800		
15	T3-T2	10.00 K		n	0.400		
16							
17	HEAT TRANSFER RESULTS						
18							
19							
20							
21	h superheater	465.50 W/m2K		$\Gamma\beta$	1683.76 J/m2Kkg		
22	h evaporator	611.71 W/m2K	WALL	k	33.00 W/mK		
23	h economiser	530.76 W/m2K		b	0.0025 m		
24	A superheater	220.11 m2		1/h	0.000076 m2K/W		
25	A evaporator	525.17 m2	EXTERNAL	Gas at T =	520.85 K		
26	A economiser	445.35 m2		Gas const.	287.00 J/kg K		
27	$\Gamma\beta$	Internal		μ	2.7E-05 Ns/m2		
28	Econ	1683.76 J/m2Kkg		Pr stream	0.684		
29	Evap	Large J/m2Kkg		k	0.041 W/mK		
30	Supr	159.90 J/m2Kkg		A duct min.	5.000 m2		
31	$\Gamma\beta$	External		A fin facto	2.500		
32	Econ	24.56 J/m2Kkg		fin b/p	0.080		
33	Evap	27.63 J/m2Kkg		fin effy.	0.800		
34	Supr	28.79 J/m2Kkg		C	0.300		

Col=7	Col=8	Col=9	Col=11	Col=12	Col=13	Col=14
35				m	0.625	
36	CALCULATIONS			n	0.333	
37				Density	0.676 kg/m ³	
38				V	45.35 m/s	
39	SUPER HEATER			Re	28369	
40				Nu	160.39	
41	θ ₁₃ av ≈	706.86 K		h plain	261.11 W/m ² K	
42	C _{pg}	1098.44 J/kgK		fin factor	2.18	
43	T ₃₄ av	673.86 K		h	570.26 W/m ² K	
44	C _{pf} av	2066.31 J/kgK		1/h	0.001754 m ² /W	
45	θ ₂	771.44 K		Γβ	24.56 J/m ² Kkg	
46	C _{pf} sat	5238.38 J/kgK		DATA FOR EVAPORATOR CALCULATIONS		
47	E	0.2418		INTERNAL	1/h	Small
48	ε	0.2136		WALL	k	33.00 W/mK
49	hA/mgC _{pg}	0.6089			b	0.0025 m
50	1/hf	0.00058 m ² /W			1/h	0.000076 m ² /W
51	Wall	0.00008 m ² /W		EXTERNAL	Gas at T =	676.62 K
52	1/hg	0.00150 m ² /W			μ	3.4E-05 Ns/m ²
53	h	465.50 W/m ² K			Pr stream	0.695
54	A	220.11 m ²			k	0.052 W/mK
55					A duct min.	5.000 m ²
56	EVAPORATOR				A fin facto	2.500
57	θ ₃	581.79 K			fin b/p	0.080
58	Check	0.05 K			fin effy.	0.800
59	ε	0.8518			C	0.300
60	hA/mgC _{pg}	1.9091			m	0.625
61	1/hf	0.000000 m ² /W			n	0.333
62	Wall	0.000076 m ² /W			Density	0.520 kg/m ³
63	1/hg	0.001559 m ² /W			V	58.907 m/s
64	h	611.71 W/m ² K			Re	22662
65	A	525.17 m ²			Nu	140.12
66					h plain	293.70 W/m ² K
67					fin factor	2.18
68					h	641.43 W/m ² K
69	ECONOMISER				1/h	0.001559 m ² /W
70	C _{pf}	4500.00 J/kgK			Γβ	27.63 J/m ² Kkg
71	θ ₃₄ av	520.63 K		DATA FOR SUPERHEATER CALCULATIONS		
72	C _{pg}	1059.33 J/kgK		INTERNAL	Steam T =	673.86 K
73	θ ₄	459.90 K			density	21.10 kg/m ³
74	Check	0.44 K			k	0.042 W/mK
75	E	0.5461			μ	2.06E-05 Ns/m ²
76	ε	0.4579			Pr	1.140
77	hA/mgC _{pg}	1.4566			N	100
78	1/hf	0.00005 m ² /W			do	0.025 m
79	Wall	0.00008 m ² /W			di	0.020 m
80	1/hg	0.00175 m ² /W			C	0.023
81	h	530.76 W/m ² K			m	0.800
82	A	445.35 m ²			n	0.400
83					V	29.707 m/s
84					Re	608554
85					Nu	1027.85
86						
87						
88						
89						

Col=7	Col=8	Col=9	Col=11	Col=12	Col=13	Col=14
90				h	1735.01	W/m2K
91				1/h	0.000576	m2K/W
92				$\Gamma\beta$	159.90	J/m2Kkg
93			WALL	k	33.00	W/mK
94				b	0.0025	m
95				1/h	0.000076	
96			EXTERNAL	Gas T =	801.69	K
97				R	287.00	J/kg K
98				μ	3.7E-05	Ns/m2
99				Pr stream	0.709	
100				k	0.057	W/mK
101				A duct min.	5.000	m2
102				A fin facto	2.500	
103				fin b/p	0.080	
104				fin effy.	0.800	
105				C	0.300	
106				m	0.625	
107				n	0.333	
108				Density	0.439	kg/m3
109				V	69.80	m/s
110				Re	20758	
111				Nu	133.53	
112				h plain	306.05	W/m2K
113				fin factor	2.18	
114				h	668.41	W/m2K
115				1/h	0.001496	m2K/W
116				$\Gamma\beta$	28.79	J/m2Kkg
117						

Formula

Row	Column	Formula
5	8	r99c5
5	13	(r9c8+r10c8)/2
6	8	r100c5
7	8	r128c5
8	8	r127c5
9	8	r104c5
10	8	r11c7-r24c5
11	8	r105c5
12	8	r106c5
13	8	r21c5
14	8	r22c5
15	8	r24c5
16	13	r8c8/r10c13/r6c13*1.2732/r12c13/r12c13
17	13	r6c13*r16c13*r12c13/r8c13
18	13	r13c13*r17c13*r14c13*r9c13*r15c13
19	13	r18c13*r7c13/r11c13
20	13	1/r19c13
21	8	r53c8
21	13	r19c13/r8c8*r14c13
22	8	r66c8
23	8	r83c8

Row	Column	Formula
23	13	$(r11c13-r12c13)/2$
24	8	$r54c8$
24	13	$r23c13/r22c13$
25	8	$r67c8$
25	13	$(r75c8+r59c8)/2$
26	8	$r84c8$
28	8	$r21c13$
30	8	$r92c13$
32	8	$r45c13$
33	8	$r72c13$
34	8	$r116c13$
37	13	$101000/r26c13/r25c13$
38	13	$r5c8/r37c13/r30c13$
39	13	$r37c13*r38c13*r11c13/r27c13$
40	13	$r34c13*r39c13*r35c13*r28c13*r36c13$
41	8	$(r6c8+r11c8+r14c8)/2$
41	13	$r40c13*r29c13/r11c13$
42	8	$950+0.21*r41c8$
42	13	$r33c13*r31c13+(1-r33c13)*(1-r32c13)$
43	8	$(r12c8+r11c8)/2$
43	13	$r42c13*r41c13$
44	8	$(7.9406-10.188*(r43c8/100)^{0.25}+4.5934*(r43c8/100)^{0.5}-0.20532*(r43c8/100))*1000$
44	13	$1/r43c13$
45	8	$r6c8-r8c8*r44c8/r5c8/r42c8*(r12c8-r11c8)$
45	13	$r43c13/r5c8*r35c13$
46	8	$41582-151.946*r11c8+0.1562*r11c8*r11c8$
47	8	$r8c8*r44c8/r5c8/r42c8$
48	8	$(r6c8-r45c8)/(r6c8-r11c8)$
49	8	$r47c8/(1-r47c8)*\ln((r48c8-1)/(r48c8/r47c8-1))$
50	8	$r91c13$
51	8	$r95c13$
51	13	$r23c13$
52	8	$r115c13$
52	13	$r51c13/r50c13$
53	8	$1/(r50c8+r51c8+r52c8)$
53	13	$(r59c8+r45c8)/2$
54	8	$r49c8*r5c8*r42c8/r53c8$
59	8	$r45c8-r8c8/r5c8/r42c8*(r108c5*1000+r46c8*r15c8)$
60	8	$(r59c8-r11c8)/r43c8$
61	8	$(r45c8-r59c8)/(r45c8-r11c8)$
62	8	$-\ln(1-r61c8)$
64	8	$r52c13$
64	13	$101000/r26c13/r53c13$
65	8	$r71c13$
65	13	$r5c8/r64c13/r57c13$
66	8	$1/(r63c8+r64c8+r65c8)$
66	13	$r64c13*r65c13*r11c13/r54c13$
67	8	$r62c8*r5c8*r42c8/r66c8$
67	13	$r61c13*r66c13*r62c13*r55c13*r63c13$
68	13	$r67c13*r56c13/r11c13$
69	13	$r60c13*r58c13+(1-r60c13)*(1-r59c13)$
70	13	$r69c13*r68c13$
71	13	$1/r70c13$
72	13	$r70c13/r5c8*r62c13$

Row	Column	Formula
73	8	$(r59c8+r7c8)/2$
74	8	$950+0.21*r73c8$
75	8	$r59c8-r8c8*r72c8/r5c8/r74c8*(r10c8-r9c8)$
76	8	$r75c8-r7c8$
76	13	$(r11c8+r12c8)/2$
77	8	$r8c8*r72c8/r5c8/r74c8$
78	8	$(r59c8-r75c8)/(r59c8-r9c8)$
79	8	$r77c8/(1-r77c8)*\ln((r78c8-1)/(r78c8/r77c8-1))$
80	8	$r20c13$
81	8	$r24c13$
82	8	$r44c13$
83	8	$1/(r80c8+r81c8+r82c8)$
84	8	$r79c8*r5c8*r74c8/r83c8$
87	13	$r8c8/r81c13/r77c13*1.2732/r83c13/r83c13$
88	13	$r77c13*r87c13*r83c13/r79c13$
89	13	$r84c13*r88c13*r85c13*r80c13*r86c13$
90	13	$r89c13*r78c13/r82c13$
91	13	$1/r90c13$
92	13	$r90c13/r8c8*r85c13$
94	13	$(r82c13-r83c13)/2$
95	13	$r94c13/r93c13$
96	13	$(r6c8+r45c8)/2$
108	13	$101000/r97c13/r96c13$
109	13	$r5c8/r108c13/r101c13$
110	13	$r108c13*r109c13*r82c13/r98c13$
111	13	$r105c13*r110c13*r106c13*r99c13*r107c13$
112	13	$r111c13*r100c13/r82c13$
113	13	$r104c13*r102c13+(1-r104c13)*1-r103c13$
114	13	$r113c13*r112c13$
115	13	$1/r114c13$
116	13	$r114c13/r5c8*r106c13$

A.3 Off Design of HRSG and Steam Turbine

The following is a block of spreadsheet added to the right of the above two blocks to calculate the off design performance of the steam bottoming cycle. The above block calculates the areas and parameters $\Gamma\beta$ of the HRSG. The following block uses these values to calculate the off design performance by guessing the steam mass flow and the amount of economiser area used in evaporation, then iterating and reguessing until the heat exchanger and energy conservation equations are satisfied. The spreadsheet requires multiple recalculation or using the iterate function of the SMART spreadsheet until convergence is indicated by either r23c20 or r24c30 being zero.

Spreadsheet

	col=16	col=17	col=18	col=19	col=20
1	Off Design Calculations - Single Pressure Steam Cycle				
2					
3	INPUTS				
4					
5	Ambient		Inlet	K	288.15
6	Gas Turbine	Exhaust	Mass flow	kg/s	153.19

	col=16	col=17	col=18	col=19	col=20
7			T exhaust	K	831.93
8		Steam	P drum	MPa	6.00
9		Throttle	P loss	%	5.00%
10		Condenser	P condenser	kPa	8.50
11		Turbine	Efficiency	%	85.00%
12	Fouling	1/h internal	Economiser	m2K/W	0.000000
13			Evaporator	m2K/W	0.000000
14			Superheater	m2K/W	0.000000
15		1/h external	Economiser	m2K/W	0.000000
16			Evaporator	m2K/W	0.000000
17			Superheater	m2K/W	0.000000
18					
19	RESULTS				
20					
21	HRSG	Effectiveness	%		72.1
22		Exhaust temperature	°C		187
23		Feed T rise in evaporator	°C		10.2
24		Evaporation in economiser	%		0.0
25	Steam turbine	Power output	MW		21.6
26		Bottoming cycle efficiency	%		24.3
27		Quality of exit steam	%		92.4
28	Condenser	Heat rejection rate	MW		43.7
29		Feed temperature	°C		42.4
30	Check	Correlation error messages =>	Inlet		OK
31			Exit		OK
32					
33	CALCULATIONS				
34					
35	DP RESULTS	Γβ internal	Economiser	J/m2Kkg	1683.765
36			Superheater	J/m2Kkg	159.905
37		Γβ external	Economiser	J/m2Kkg	24.564
38			Evaporator	J/m2Kkg	27.630
39			Superheater	J/m2Kkg	28.792
40		A actual	Economiser	m2	445.355
41			Evaporator	m2	525.173
42			Superheater	m2	220.105
43	ITERATION one	Initial guess	Steam flow	kg/s	19.694
44		1/h internal	Economiser	m2K/W	0.000055
45			Superheater	m2K/W	0.000576
46		1/h external	Economiser	m2K/W	0.001754
47			Evaporator	m2K/W	0.001559
48			Superheater	m2K/W	0.001496
49		1/h wall	Economiser	m2K/W	0.000076
50			Evaporator	m2K/W	0.000076
51			Superheater	m2K/W	0.000076
52		h total	Economiser	W/m2K	530.765
53			Evaporator	W/m2K	611.706
54			Superheater	W/m2K	465.513
55		Tav gas	Economiser	K	520.627
56			Evaporator	K	676.617
57			Superheater	K	801.689
58		Cpg	Economiser	J/kgK	1059.332
59			Evaporator	J/kgK	1092.090
60			Superheater	J/kgK	1118.355
61	ITERATION two	Initial guess	Evap in Econ	-	0.000

	col=16	col=17	col=18	col=19	col=20
62		A effective	Economiser	m2	445.355
63			Evaporator	m2	525.173
64		hA/mgCpg	Economiser	W/m2K	1.457
65			Evaporator	W/m2K	1.920
66			Superheater	W/m2K	0.598
67		Tf av	Economiser	K	427.186
68			Evaporator	K	548.790
69			Superheater	K	673.861
70		Cpf av	Economiser	J/kgK	4500.000
71			Evaporator	J/kgK	5238.381
72			Superheater	J/kgK	2066.313
73		hfg	Evaporator	J/kg	1568219
74		E	Economiser	-	0.546
75			Superheater	-	0.238
76		€	Economiser	-	0.458
77			Evaporator	-	0.853
78			Superheater	-	0.210
79		Temperatures	T1	K	315.582
80			T3	K	548.790
81			Ø2	K	772.468
82			Ø3	K	581.576
83			Ø4	K	459.781
84			T4	K	799.132
85			T2	K	538.601
86		Result	T3-T2	K	10.189
87	LOOP END two	New guess	Evap in Econ	-	0.000
88		Result	Flow steam	kg/s	19.694
89			Error	kg/s	0.000
90	LOOP END one	New guess	Flow steam	kg/s	19.694
91		Feed pump	h rise ideal	kJ/kg	6.021
92			h rise actual	kJ/kg	7.527
93		HRSG	h inlet	kJ/kg	185.243
94			h outlet	J/kg	3496.597
95			€ HRSG	-	0.721
96		Throttle	P outlet	MPa	5.700
97			s outlet	kJ/kgK	7.018
98		Steam Turb.	hf outlet P	kJ/kg	177.717
99			hg outlet P	kJ/kg	2579.168
100			sf outlet P	kJ/kgK	0.609
101			sg outlet P	kJ/kgK	8.206
102			x outlet ideal %		0.844
103			h outlet ideal	kJ/kg	2203.625
104			h drop ideal	kJ/kg	1292.972
105			h drop actual	kJ/kg	1099.026
106			Power output	MW	21.645
107			h outlet	kJ/kg	2397.571
108			x outlet actual%		0.924
109		Condenser	h drop	kJ/kg	2219.854
110			Q rejected	MW	43.719
111		Pump work	Power input	MW	0.148
112					

Formula

Row	Column	Formula
21	20	r95c20*100
22	20	r83c20-273.15
23	20	r86c20
24	20	r87c20*100
25	20	r106c20
26	20	r25c20*100000000/(r6c20*(950+0.21*(r7c20+r5c20)/2)*(r7c20-r5c20))
27	20	r108c20*100
28	20	r110c20
29	20	r79c20-273.15
30	20	if r8c20>8 or r8c20<3 or r84c20>973 or r84c20<573 then "throttle conditions out of range" else "OK"
31	20	if r10c20>15 or r10c20<3 then "condenser P out of range" else "OK"
35	20	r21c13
36	20	r92c13
37	20	r45c13
38	20	r72c13
39	20	r116c13
40	20	r84c8
41	20	r67c8
42	20	r54c8
43	20	r90c20
44	20	1/(r28c8*r43c20^r14c13)
45	20	1/(r30c8*r43c20^r85c13)
46	20	1/(r32c8*r6c20^r35c13)
47	20	1/(r33c8*r6c20^r62c13)
48	20	1/(r34c8*r6c20^r106c13)
49	20	r24c13
50	20	r52c13
51	20	r95c13
52	20	1/(r44c20+r12c20+r49c20+r15c20+r46c20)
53	20	1/(r13c20+r50c20+r16c20+r47c20)
54	20	1/(r45c20+r14c20+r51c20+r17c20+r48c20)
55	20	(r7c8+r59c8)/2
56	20	(r59c8+r45c8)/2
57	20	(r45c8+r6c8)/2
58	20	950+0.21*r55c20
59	20	950+0.21*r56c20
60	20	950+0.21*r57c20
61	20	r87c20
62	20	r40c20*(1-r61c20)
63	20	r41c20+r61c20*r40c20
64	20	r52c20*r62c20/r6c20/r58c20
65	20	r53c20*r63c20/r6c20/r59c20
66	20	r42c20*r54c20/r6c20/r60c20
67	20	(r9c8+r10c8)/2
68	20	r11c8
69	20	(r11c8+r12c8)/2
70	20	r72c8
71	20	41582-151.946*r80c20+0.1562*r80c20*r80c20
72	20	(7.9406-10.188*(r69c20/100)^0.25+4.5934*(r69c20/100)^0.5-0.20532*(r69c20/100))*1000

Row	Column	Formula
73	20	14461.67*r68c20-671200-18.9162*r68c20*r68c20
74	20	r43c20*r70c20/r6c20/r58c20
75	20	r43c20*r72c20/r6c20/r60c20
76	20	(1-exp((1-r74c20)/r74c20*r64c20))/(1-exp((1-r74c20)/r74c20*r64c20)/r74c20)
77	20	1-exp(-r65c20)
78	20	(1-exp((1-r75c20)/r75c20*r66c20))/(1-exp((1-r75c20)/r75c20*r66c20)/r75c20)
79	20	288.33+4.017*r10c20-0.0954*r10c20*r10c20
80	20	455.61+19.895*r8c20-0.7275*r8c20*r8c20
81	20	r7c20-r78c20*(r7c20-r80c20)
82	20	r81c20-r77c20*(r81c20-r80c20)
83	20	r82c20-r76c20*(r82c20-r79c20)
84	20	r80c20+(r7c20-r81c20)/r75c20
85	20	r79c20+(r82c20-r83c20)/r74c20
86	20	r80c20-r85c20
87	20	if r86c20>=0 then 0 else r61c20-r86c20/r80c20*2
88	20	r6c20*r59c20*(r81c20-r82c20)/(r73c20+r71c20*(r80c20-r85c20))
89	20	r88c20-r43c20
90	20	r88c20-r89c20*0.2
91	20	0.001005*(r8c20*1000-r10c20)
92	20	r91c20/r29c5
93	20	r92c20+r98c20
94	20	632.679+5.10687*r84c20-0.00174667*r84c20*r84c20-15.05*r8c20-0.316667*r8c20*r8c20
95	20	(r7c20-r83c20)/(r7c20-r79c20)
96	20	r8c20*(1-r9c20)
97	20	2.18287+0.0113789*r84c20-0.00000533167*r84c20*r84c20-0.1893*r96c20+0.00694444*r96c20*r96c20
98	20	63.91+16.772*r10c20-0.398*r10c20*r10c20
99	20	2524.7+8.72*r10c20-0.272*r10c20*r10c20
100	20	0.1951+0.0671*r10c20-0.002168*r10c20*r10c20
101	20	8.8092-0.09974*r10c20+0.003384*r10c20*r10c20
102	20	(r97c20-r100c20)/(r101c20-r100c20)
103	20	r98c20+r102c20*(r99c20-r98c20)
104	20	r94c20-r103c20
105	20	r104c20*r11c20
106	20	r105c20*r90c20/1000
107	20	r94c20-r105c20
108	20	(r107c20-r98c20)/(r99c20-r98c20)
109	20	r107c20-r98c20
110	20	r109c20*r90c20/1000
111	20	r92c20*r90c20/1000

A.4 Design Point of Frame 9F

The following spreadsheet is for the design point model of the Frame 9F heavy industrial gas turbine. It is based on the overall efficiency, TRIT and exhaust temperature of the quoted performance figures. The component efficiencies, combustor pressure drop and cooling flows are not given and are assumed and adjusted to fit the quoted values.

Col. = 1 Col. = 2

Col. = 3

Col. = Col. = 5

Row = 2	INPUTS		Units	Value
Row = 3	Compressor	Inlet air flow	kg/s	599.0
Row = 4		Pressure ratio	-	13.5
Row = 5	Cooling	Polytropic efficiency	%	87.0%
Row = 6		Total cooling bleed	%	15.0%
Row = 7		% to 1st nozzle	%	40.0%
Row = 8		% before turbine	%	40.0%
Row = 9	Combustor	% after turbine	%	20.0%
Row = 10		Pressure loss	%	5.0%
Row = 11		Fuel LCV	kJ/kg	42000.0
Row = 12	Turbine	Combustion efficiency	%	100.0%
Row = 13		Combustor outlet T	K	1597.0
Row = 14	Overall	Polytropic efficiency	%	88.0%
Row = 15		Shaft/radiation loss	%	0.7%
Row = 16		Generator efficiency	%	98.0%
Row = 17		Inlet/exhaust losses	%	2.0%
Row = 18		R air	J/kg K	287.0
Row = 19	RESULTS			
Row = 20	GAS TURBINE	Exhaust flow	kg/s	613
Row = 21		Turbine rotor inlet T	K	1533
Row = 22		Power output	MW	199
Row = 23		Efficiency	%	34.1
Row = 24		Exhaust temperature	K	856
Row = 25		Bottoming cycle efficiency	%	29.8
Row = 26	CALCULATIONS	Combined Cycle Efficiency	%	53.3
Row = 27				
Row = 28	GAS TURBINE	Inlet T	K	288.150
Row = 29		Inlet P	kPa	101.325
Row = 30	Compressor	T average	K	476.367
Row = 31		CP average	J/kgK	1027.412
Row = 32		$nC(k/(k-1))$	-	3.114
Row = 33		T out	K	664.584
Row = 34	Combustor	Bleed	kg/s	89.850
Row = 35		Flow passed on	kg/s	509.150
Row = 36		Total work	MW	231.665
Row = 37		Fuel air ratio	-	0.028
Row = 38	1st Nozzle	Fuel flow	kg/s	14.173
Row = 39		Cooling	kg/s	35.940
Row = 40	Turbine	Flow out	kg/s	559.263
Row = 41		T out	K	1532.965
Row = 42		Cooling before	kg/s	35.940
Row = 43		Effective flow	kg/s	595.203
Row = 44	Overall	Effective T in	K	1477.160
Row = 45		P out / P in	-	0.078
Row = 46		T average	K	1169.430
Row = 47		CP average	J/kgK	1195.580
Row = 48	Overall	$(k/(k-1))/nT$	-	4.734
Row = 49		T out	K	861.700
Row = 50		Power	MW	434.903
Row = 51		Net output	MW	203.238

	Col. = 1	Col. = 2	Col. = 3	Col. = 4	Col. = 5
Row = 57			Cooling after	kg/s	17.970
Row = 58			Exhaust T	K	855.749
Row = 59			GT efficiency	%	34.142%
Row = 60	COMBINED CYCLE		Corrected for losses	%	33.459%
Row = 61			Γ	-	2.970
Row = 62			BC efficiency	%	29.760%
Row = 63			CC efficiency	%	53.262%
Row = 64			Less generator loss	%	52.196%

Formula

Row	Column	Formula
23	5	$r4c5 + r43c5$
24	5	$r46c5$
25	5	$r56c5 * r17c5$
26	5	$r59c5 * 100$
27	5	$r58c5$
28	5	$r62c5 * 100$
29	5	$r63c5 * 100$
35	5	$(r33c5 + r38c5) / 2$
36	5	$991.24 - 0.0141 * r35c5 + 0.000189 * r35c5 * r35c5$
37	5	$r6c5 * r36c5 / r19c5$
38	5	$r33c5 * r5c5^{(1/r37c5)}$
39	5	$r7c5 * r4c5$
40	5	$r4c5 - r39c5$
41	5	$r4c5 * r36c5 * (r38c5 - r33c5) / 1000000$
42	5	$1.1 * (r14c5 - r38c5 - 50) * (1 + r38c5 / 3250) / r13c5 / r12c5$
43	5	$r42c5 * r40c5$
44	5	$r8c5 * r7c5 * r4c5$
45	5	$r44c5 + r43c5 + r40c5$
46	5	$r14c5 - r44c5 / (r40c5 + r43c5) * (r14c5 - r38c5)$
47	5	$r9c5 * r7c5 * r4c5$
48	5	$r47c5 + r45c5$
49	5	$r46c5 - r47c5 / r45c5 * (r46c5 - r38c5)$
50	5	$1 / (r5c5 * (1 - r11c5))$
51	5	$(r49c5 + r54c5) / 2$
52	5	$950 + 0.21 * r51c5$
53	5	$r52c5 / r19c5 / r15c5$
54	5	$r49c5 * r50c5^{(1/r53c5)}$
55	5	$r48c5 * r52c5 * (r49c5 - r54c5) / 1000000 * (1 - r16c5)$
56	5	$r55c5 - r41c5$
57	5	$r10c5 * r7c5 * r4c5$
58	5	$r54c5 - c57c5 / r48c5 * (r54c5 - r38c5)$
59	5	$r56c5 / r43c5 / r12c5 * 1000$
60	5	$r59c5 * (1 - r18c5)$
61	5	$r58c5 / r33c5$
62	5	$-0.7285 + 0.5855 * r61c5 - 0.08081 * r61c5 * r61c5$
63	5	$r60c5 + (1 - r60c5) * r62c5$
64	5	$r63c5 * r17c5$

APPENDIX B - PROPERTY CORRELATIONS

B.1 Steam

Many empirical equations are available to represent the properties of steam and water. Virtually all modern steam tables are based on one of five equations, IFC industrial, IFC scientific, MIT, Pollak or NBS/NRC [162]. These equations are primarily concerned about accuracy for generation of tables, and are far too cumbersome for this thesis. Even a "very simple equation" of state by J.B. Young [162], designed to simplify the calculation for finite differences models, involves in excess of 20 lengthy equations and some 162 operations. This method would still be cumbersome for this thesis, where trends and not accuracy is required. The main reason for these complex methods is to cover as large an area of the pressure-temperature range as possible. As the performance modelling in this thesis is only concerned with the inlet and outlet of the steam turbine, specific equations were developed for the pressure and temperature ranges involved. To obtain suitable equations for steam properties, formulae were fitted to data from the steam tables in the reference Van Wylen and Sonntag [1].

For the superheated range the following form was used;

$$H = c_1 + c_2T + c_3T^2 + c_4P + c_5P^2 \quad \dots \text{B.1)}$$

$$S = k_1 + k_2T + k_3T^2 + k_4P + k_5P^2 \quad \dots \text{B.2)}$$

The constants were calculated using the points; at temperatures 300°C, 400°C, and 600°C at 6 MPa, and at pressures 3, 6, and 9 MPa, at 400°C. The resulting constants are as follows;

$$\begin{array}{ll} c_1 = +6.32679\text{E}+02 & c_2 = +5.10687\text{E}+00 \\ c_3 = -1.74667\text{E}-03 & c_4 = -1.50500\text{E}+01 \\ c_5 = -3.16667\text{E}-01 & k_1 = +2.18287\text{E}+00 \\ k_2 = +1.13789\text{E}-02 & k_3 = -5.33167\text{E}-06 \\ k_4 = -1.89300\text{E}-01 & k_5 = +6.94444\text{E}-03 \end{array}$$

where P in MPa, T in K, H in kJ/kg, S in kJ/kgK.

The errors for the above formulae relative to the steam tables in reference [1] are in table B.1. They are used with an error message if outside the range 3-8 MPa and 573-973 K (300-700 °C) giving a maximum error of 2% at the extremes, and usually less than 1% within the region used in this thesis. The reference steam tables will contain errors which combine with these errors, but the total errors will not be significant to the results of this thesis.

Saturated value property equations were also developed similarly but in quadratic form. The formulae obtained are as follows;

$$H_f = 63.910 + 16.772 P - 0.398 P^2 \quad \dots \text{B.3)}$$

$$H_g = 2524.7 + 8.720 P - 0.272 P^2 \quad \dots \text{B.4)}$$

$$S_f = 0.1951 + 0.0671 P - 0.002168 P^2 \quad \dots \text{B.5)}$$

$$S_g = -8.8092 - 0.09974 P + 0.003384 P^2 \quad \dots \text{B.6)}$$

where P is in kPa absolute, H in kJ/kg and S in kJ/kgK. The errors for these formula relative to the steam tables in reference [1] are in table B.2. The range used for the error message is 3-15 kPa but in the normal

range of 5-10 kPa the maximum error is 1%.

The heat capacity of steam as used in the HRSG and is found from the correlation in reference [1] as follows;

$$C_p = 7.9406 - 10.188(\theta)^{0.25} + 4.5934(\theta)^{0.5} - 0.20532(\theta) \quad \dots B.7)$$

$\theta = T/100$, T in K and C_p in kJ/kgK and accurate to 0.43% of tables within 300-3500 K

Other correlations developed from the tables in reference [1] for saturated water are as follows;

Saturated water;

$$T_{SAT} = 455.61 + 19.895(P) - 0.7275(P)^2 \quad \dots B.8)$$

T in K and P in MPa accurate to within 0.4% of tables over 3-10 MPa

$$T_{SAT} = 288.33 + 4.017(P) - 0.09540(P)^2 \quad \dots B.9)$$

T in K and P in kPa and accurate within 0.8% of tables over 3-20 kPa absolute

$$(C_{p_f})_{SAT} = 41582 - 151.946(T_{SAT}) + 0.1562(T_{SAT})^2 \quad \dots B.10)$$

T in K and accurate to within 0.9% of tables from 500 - 580 K

$$h_{fg} = -671200 + 14461.67(T_{SAT}) - 18.9167(T_{SAT})^2 \quad \dots B.11)$$

T in K and accurate to within 0.3% of tables from 480 - 580 K

B.2 Air

A correlation of the heat capacity of dry air was calculated from the data in the tables of reference [1].

$$C_p = 991.24 - 0.0141 T + 0.000186 T^2 \quad \dots B.12)$$

T in K and accurate to within 0.3% of tables for range 280 - 800 K.

Table B.1 Correlation Errors for Superheated Steam Vapour

ENTHALPY - Error of formula to tables, in %							
Pressure MPa	Temperature °C						
	300	350	400	450	500	600	700
1	2.64	1.15	0.01		-1.24	-1.27	-0.25
2	2.28	1.01	0.00		-1.10	-1.05	0.01
3	1.86	0.85	0.00	-0.61	-0.94	-0.81	0.29
4	1.35	0.68	0.00	-0.50	-0.77	-0.56	0.59
5	0.74	0.48	0.00	-0.39	-0.58	-0.28	0.90
6	0.00	0.26	0.00	-0.27	-0.38	0.00	1.22
7	-0.94	0.00	0.00	-0.13	-0.17	0.30	1.57
8	-2.16	-0.29	0.00	0.01	0.06	0.62	1.93
9		-0.65	0.00	0.17	0.30	0.95	2.30
10		-1.07	-0.01	0.33	0.56	1.31	2.70
ENTROPY - Error of formula to tables, in %							
Pressure MPa	Temperature °C						
	300	350	400	450	500	600	700
1	4.95	3.85	2.97		1.96	1.97	3.01
2	2.44	1.51	0.73		-0.14	0.00	1.17
3	1.41	0.68	0.01	-0.48	-0.74	-0.52	0.72
4	0.87	0.39	-0.16	-0.58	-0.80	-0.52	0.78
5	0.47	0.31	-0.11	-0.46	-0.63	-0.29	1.05
6	0.01	0.27	0.01	-0.26	-0.39	0.01	1.38
7	-0.62	0.17	0.11	-0.08	-0.15	0.30	1.71
8	-1.58	-0.04	0.12	0.04	0.02	0.54	1.99
9		-0.42	0.01	0.05	0.09	0.68	2.17
10		-1.00	-0.25	-0.08	0.04	0.70	2.24

Table B.2 Correlation Errors for Saturated Steam

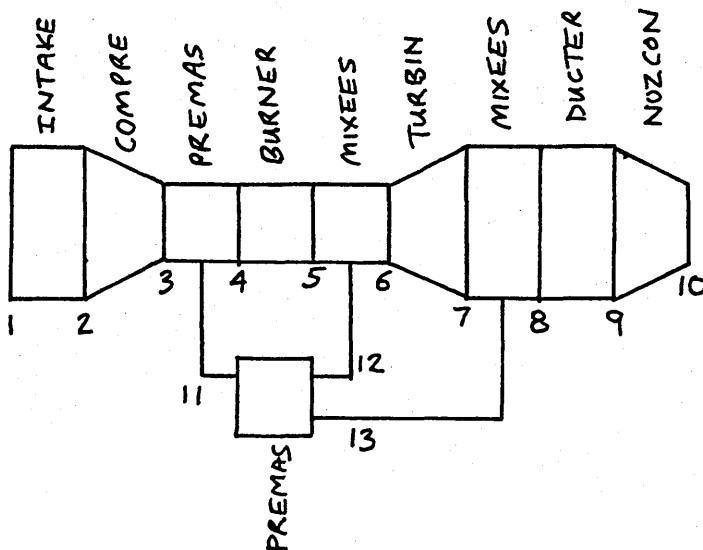
Pressure kPa	Hg Error %	Hf Error %	Sg Error %	Sf Error %
2.5	-0.19	-16.80	0.72	-11.96
3.0	-0.11	-9.49	0.43	-6.32
4.0	-0.03	-2.61	0.12	-1.47
5.0	0.00	0.00	0.00	0.00
7.5	0.00	0.88	0.00	0.00
10.0	0.00	0.00	0.00	0.00
15.0	0.18	0.00	-0.82	5.44
20.0	0.74	4.47	-3.28	19.48

APPENDIX C - TURBOMATCH PROGRAMS

The following programs are for the package TURBOMATCH, a design and off-design point calculation scheme for gas turbines. The first major block of each program is the program proper, consisting of the subroutines called during calculations to model each component or action. The next major block of data is the "Brick data" and gives the design point particulars for the Gas Turbine. A full explanation of TURBOMATCH is beyond the scope of this thesis and the programs are included for reference only.

C.1 Single Shaft Engine

Figure C.1 Single Shaft Layout



Program

Configuration SINGLE SHAFT GT ////

OD SI KE SP

-1

-1

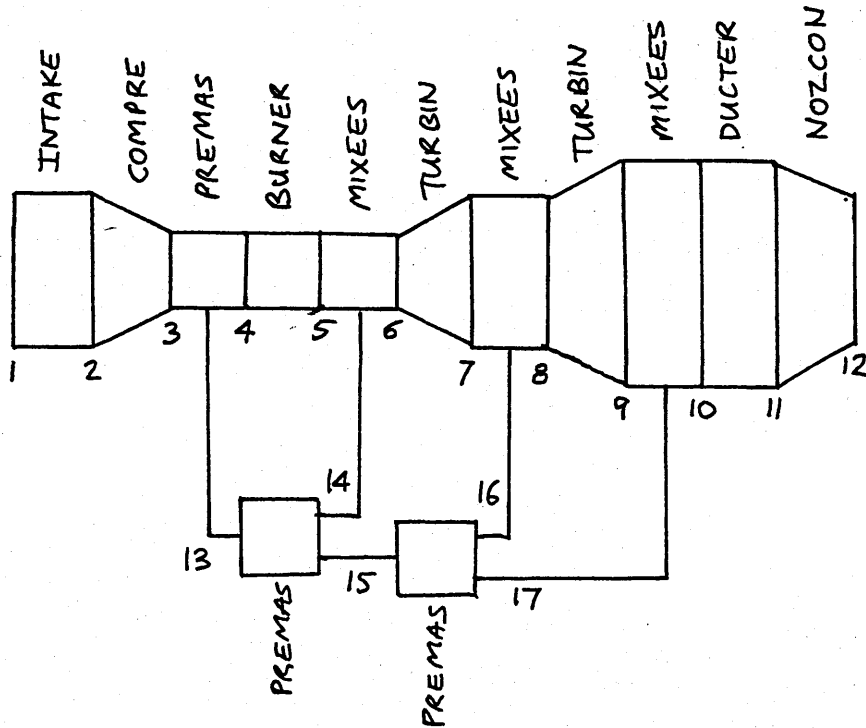
INTAKE	S1-2	D1-4	R70	
COMPRE	S2-3	D5-10	R71	V1,2,5
PREMAS	S3,11,4	D11-14		
PREMAS	S11,12,13	D15-18		
BURNER	S4-5	D19-21	R72	V2,1,5,6
MIXEES	S5,12,6			
TURBIN	S6-7	D22-29,71		V3,2,23
MIXEES	S7,13,8			
DUCTER	S8-9	D30-33	R73	
NOZCON	S9-10,1	D34	R74	
PERFOR	S1,0,0	D22,35-37,74,70,72		
CODEND				

Brick Data for SINGLE SHAFT GT ////

1	0.0	!	INTAKE	Altitude
2	0.0	!		I.S.A. deviation
3	0.0	!		Mach number
4	0.990	!		Pressure recovery
5	-1.0	!	COMPRES	Z Surge margin (default 0.85)
6	-1.0	!		PCN Speed (default 100%)
7	12.0	!		Pressure ratio
8	0.860	!		Isentropic efficiency
9	0.0	!		Error selection (off)
10	5.0	!		Map number
11	0.050	!	PREMAS	Cooling flow
12	0.0	!		Bleed
13	1.0	!		Pressure recovery
14	0.0	!		Pressure loss
15	0.667	!	PREMAS	Flow split
16	0.0	!		Bleed
17	1.0	!		Pressure recovery
18	0.0	!		Pressure loss
19	0.050	!	BURNER	Pressure loss
20	1.0	!		Combustion efficiency
21	-1.0	!		Fuel flow (stated later)
22	29000000	!	TURBIN	Auxiliary work
23	-1.0	!		TF Non-dimensional flow (default 0.8)
24	-1.0	!		CN Non-dimensional speed (default 0.6)
25	0.880	!		Isentropic efficiency
26	-1.0	!		PCN Speed
27	1.0	!		Compressor number
28	5.0	!		Map number
29	-1.0	!		Power law (constant)
30	0.0	!	DUCTER	Reheat switch (off)
31	0.020	!		Pressure loss
32	0.0	!		Combustion efficiency
33	1000000	!		Flow limit (n/a)
34	-1.0	!	NOZCON	Floating area switch (off)
35	1.0	!	PERFOR	Propellor efficiency
36	0.0	!		Scaling index
37	0.0	!		Required thrust
-1				
1 2	150.0	!		Mass Flow
5 6	1200.0	!		Combustor outlet temperature
9 7	50.0	!		Outlet velocity
-1				

C.2 Two Shaft Engine

Figure C.2 Two Shaft Layout



Program

Configuration TWIN SHAFT - SINGLE SPOOL GT ////

OD SI KE SP

-1

-1

INTAKE	S1-2	D1-4	R70	
COMPRE	S2-3	D5-10	R71	V1,2,5 W2,2,6
PREMAS	S3,13,4	D11-14		
PREMAS	S13,15,14	D15-18		
PREMAS	S15,17,16	D19-22		
BURNER	S4-5	D23-25	R72	V3,1,5,6
MIXEES	S5,14,6			
TURBIN	S6-7	D26-33,71		V4,2,27
MIXEES	S7,16,8			
TURBIN	S8-9	D34-42		V5,2,35
MIXEES	S9,17,10			
DUCTER	S10-11	D43-46	R73	
NOZCON	S11-12,1	D47	R74	
PERFOR	S1,0,0	D34,48-50,74,70,72		
CODEND				

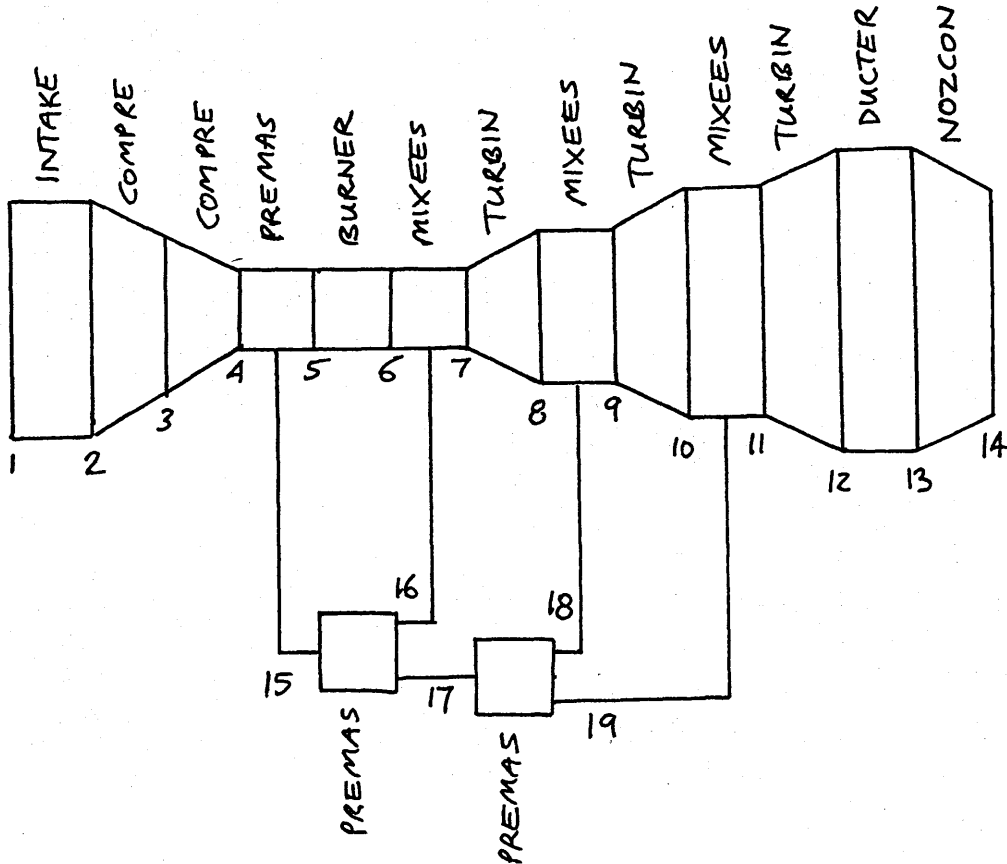
Brick Data for TWIN SHAFT - SINGLE SPOOL GT ////

1	0.0	!	INTAKE	Altitude
2	0.0	!		I.S.A. deviation
3	0.0	!		Mach number
4	0.990	!		Pressure recovery
5	-1.0	!	COMPRE	Z Surge margin (default 0.85)

6	-1.0	!		PCN Speed (default 100%)
7	12.0	!		Pressure ratio
8	0.860	!		Isentropic efficiency
9	0.0	!		Error selection (off)
10	5.0	!		Map number
11	0.050	!	PREMAS	Cooling flow
12	0.0	!		Bleed
13	1.0	!		Pressure recovery
14	0.0	!		Pressure loss
15	0.333	!	PREMAS	Flow split
16	0.0	!		Bleed
17	1.0	!		Pressure recovery
18	0.0	!		Pressure loss
19	0.333	!	PREMAS	Flow split
20	0.0	!		Bleed
21	1.0	!		Pressure recovery
22	0.0	!		Pressure loss
23	0.050	!	BURNER	Pressure loss
24	1.0	!		Combustion efficiency
25	-1.0	!		Fuel flow (stated later)
26	0.0	!	TURBIN	Auxiliary work
27	-1.0	!		TF Non-dimensional flow (default 0.8)
28	-1.0	!		CN Non-dimensional speed (default 0.6)
29	0.880	!		Isentropic efficiency
30	-1.0	!		PCN Speed
31	1.0	!		Compressor number
32	5.0	!		Map number
33	-1.0	!		Power law (constant)
34	29000000	!	TURBIN	Auxiliary work
35	-1.0	!		TF Non-dimensional flow (default 0.8)
36	-1.0	!		CN Non-dimensional speed (default 0.6)
37	0.880	!		Isentropic efficiency
38	-1.0	!		PCN Speed
39	-1.0	!		Compressor number (free turbine)
40	5.0	!		Map number
41	-1.0	!		Power law (constant)
42	-1.0	!		Compressor work (n/a)
43	0.0	!	DUCTER	Reheat switch (off)
44	0.020	!		Pressure loss
45	0.0	!		Combustion efficiency
46	1000000	!		Flow limit (n/a)
47	-1.0	!	NOZCON	Floating area switch (off)
48	1.0	!	PERFOR	Propellor efficiency
49	0.0	!		Scaling index
50	0.0	!		Required thrust
-1				
1 2	150.0	!	Mass Flow	
5 6	1200.0	!	Combustor outlet temperature	
11 7	50.0	!	Outlet velocity	
-1				

C.3 Three Shaft Engine

Figure C.3 Three Shaft Layout



Program

Configuration THREE SHAFT - TWO SPOOL GT ////

OD SI KE SP

-1

-1

INTAKE	S1-2	D1-4	R70	
COMPRE	S2-3	D5-10	R71	V1,2,5 W2,2,6
COMPRE	S3-4	D11-16	R72	V3,2,11 W4,2,12
PREMAS	S4,15,5	D17-20		
PREMAS	S15,17,16	D21-24		
PREMAS	S17,19,18	D25-28		
BURNER	S5-6	D29-31	R73	V5,1,6,6
MIXEES	S6,16,7			
TURBIN	S7-8	D32-39,72		V6,2,33
MIXEES	S8,18,9			
TURBIN	S9-10	D40-47,71		V7,2,41
MIXEES	S10,19,11			
TURBIN	S11-12	D48-56		V8,2,49
DUCTER	S12-13	D57-60	R74	
NOZCON	S13-14,1	D61	R75	
PERFOR	S1,0,0	D48,62-64,75,70,73		
CODEND				

Brick Data for TWIN SHAFT - SINGLE SPOOL GT ////

1	0.0	!	INTAKE	Altitude
2	0.0	!		I.S.A. deviation
3	0.0	!		Mach number
4	0.990	!		Pressure recovery
5	-1.0	!	COMPRES 1	Z Surge margin (default 0.85)
6	-1.0	!		PCN Speed (default 100%)
7	3.4641	!		Pressure ratio
8	0.860	!		Isentropic efficiency
9	0.0	!		Error selection (off)
10	5.0	!		Map number
11	-1.0	!	COMPRES 2	Z Surge margin (default 0.85)
12	-1.0	!		PCN Speed (default 100%)
13	3.4641	!		Pressure ratio
14	0.860	!		Isentropic efficiency
15	0.0	!		Error selection (off)
16	5.0	!		Map number
17	0.050	!	PREMAS	Cooling flow
18	0.0	!		Bleed
19	1.0	!		Pressure recovery
20	0.0	!		Pressure loss
21	0.333	!	PREMAS	Flow split
22	0.0	!		Bleed
23	1.0	!		Pressure recovery
24	0.0	!		Pressure loss
25	0.333	!	PREMAS	Flow split
26	0.0	!		Bleed
27	1.0	!		Pressure recovery
28	0.0	!		Pressure loss
29	0.050	!	BURNER	Pressure loss
30	1.0	!		Combustion efficiency
31	-1.0	!		Fuel flow (stated later)
32	0.0	!	TURBIN	Auxiliary work
33	-1.0	!		TF Non-dimensional flow (default 0.8)
34	-1.0	!		CN Non-dimensional speed (default 0.6)
35	0.880	!		Isentropic efficiency
36	-1.0	!		PCN Speed
37	2.0	!		Compressor number
38	5.0	!		Map number
39	-1.0	!		Power law (N/A)
40	0.0	!	TURBIN	Auxiliary work
41	-1.0	!		TF Non-dimensional flow (default 0.8)
42	-1.0	!		CN Non-dimensional speed (default 0.6)
43	0.880	!		Isentropic efficiency
44	-1.0	!		PCN Speed
45	1.0	!		Compressor number
46	5.0	!		Map number
47	-1.0	!		Power law (N/A)
48	29000000	!	TURBIN	Auxiliary work
49	-1.0	!		TF Non-dimensional flow (default 0.8)
50	-1.0	!		CN Non-dimensional speed (default 0.6)
51	0.880	!		Isentropic efficiency
52	-1.0	!		PCN Speed
53	-1.0	!		Compressor number (free turbine)
54	5.0	!		Map number
55	-1.0	!		Power law (constant)

56	-1.0	!		Compressor work (n/a)
57	0.0	!	DUCTER	Reheat switch (off)
58	0.020	!		Pressure loss
59	0.0	!		Combustion efficiency
60	1000000	!		Flow limit (n/a)
61	-1.0	!	NOZCON	Floating area switch (off)
62	1.0	!	PERFOR	Propellor efficiency
63	0.0	!		Scaling index
64	0.0	!		Required thrust
-1				
1 2	150.0	!		Mass Flow
6 6	1200.0	!		Combustor outlet temperature
13 7	50.0	!		Outlet velocity
-1				

APPENDIX D — ENGINE DATA

The following tables of data are obtained from the references [143], [203] and [49]. It only includes engines of over 5 MW. The information is only approximate as the references vary in assumptions, including or not inlet/exhaust losses, natural gas or distillate, shaft power or electrical power. Also they would normally include a manufacturers factor for unit to unit variation. Reference [49] is an attempt to compare engines on a more even basis. Wherever possible in this table the latest figures are used and are for the situation of natural gas and electrical power, with inlet and exhaust losses included. Where the TRIT figures are from reference [49] they are indicated by a *.

Table D.1 Electrical Generation Gas Turbines														
Model	Year	Type	Power	Heat	Effy.	rP	TRIT	Speed	Exhaust	Comp.	Turb.	Number	Number	
		Aero		Rate						Stages	Stages	Shafts	Comb.	
		/Ind	MW	Btu/kWhr	%		° C		kg/s	°C				
ASEA Brown Boveri														
GT9			34.30	12960	26.3	9.0	N/A		163.0	510	15	4	1	
GT8	1984	I	46.90	11430	29.9	16.3	1185 *	6339	180.0	523	12	3	1	
GT11	1971	I	71.90	11600	29.4	11.0	1022 *	3600	290.0	520	17	5	1	
GT11N	1987	I	81.60	11140	30.6	12.4	1063 *	3600	316.0	515	18	5	1	
Mars	1977	I	7.70	11390	30.0	16.0	1057 *	8600	36.0	466	15	2/2	2	
Mars	1977	I	8.83	11040	30.9	16.0	1093	8600	38.0	482	15	2/2	2	
GT35	1968		16.53	10750	31.7	12.0	850	3600	93.0	374	10/8	1/2/3	3	
GT10			21.15	10180	33.5	13.6	1055	7700	78.0	519			1	
GT13	1970	I	98.20	11200	30.5	12.5	1005 *	3000	406.0	489	18	5	1	
GT13E	1983	I	147.90	10392	32.8	13.9	1147 *	3000	502.0	516	21	5	1	
Alison														
571-KA	1986		5.74	7510	45.4	12.7	803	11437		533	13	5	2	
Alsthom														
PG9001F	1988	I	212.20	9995	34.1	13.5	1260	3000	612.0	583	17	3	1	
PB6541B	1978	I	38.34	10860	31.4	11.8	1104	5100	139.0	540	17	3	1	
PG5371PA	1958	I	26.30	11820	28.9	10.2	957	5100	125.0	483	17	2	1	
PG9161E	1976	I	116.40	10310	33.1	12.1	1104	3000	409.0	529	17	3	1	
Cooper rolls														
2000 AVON		A	14.42	12039	28.3	9.2	876	5500	78.0	441	17	3/2	2	
3000 SPEY		A	12.55	9918	34.4	19.0	988	6350	57.0	404	5/11	2/2/2	3	
6000 RB211		A	24.94	9584	35.6	20.0	1164	4800	90.0	465	7/6	1/1/2	3	
Dresser-Rand														
DJ-290R			25.00	9538	35.8	19.3	1162	5400	90.0	464	7/6	1/1/2	3	
DJ-460G			33.77	9334	36.6	26.8	1155	3600	122.5	445				
GT-60			12.80	9965	34.2	21.5	1218	6800	45.0	483				
GT-54			15.00	11830	28.8	8.7	875	5500	79.0	430				
GT-61			21.30	9515	35.9	18.1	1231	5500	68.0	532				
DJ-300G			22.10	9406	36.3	18.1	1231	3600	68.0	528				
Fiat														
TG 20	1971		36.90	11725	29.1	11.0	1044	4920			18	3	1	
TG 50	1975		101.00	11050	30.9	12.0	1096	3000			20	4	1	
TG 20	1984		41.40	10715	31.8	14.0	1085	5400			19	3	1	
LM 2500-20		A	12.85	10041	34.0	14.8	618				16	2/6	2	
LM 2500-30		A	20.10	9435	36.2	18.0	766				16	2/6	2	
TG 50	1985		124.60	10205	33.4	14.0	1085	3000			20	4	1	
TG 16	1968		17.70	13490	25.3	7.0	788	4850			15	3/2	2	

Model	Year	Type Aero /Ind	Power MW	Heat Rate Btu/kWhr	Effy. %	rP	TRIT ° C	Speed	Exhaust kg/s	° C	Comp. Stages	Turb. Stages	Number Shafts	Number Comb.
GE Business														
PG7191(F)	1987	I	150.00	9880	34.5	13.5	1260	3600	425.0	583	17	3	1	10
PG9161(E)	1976	I	116.90	10310	33.1	12.1	1104	3000	411.0	529	17	3	1	14
PG9281(F)	1988	I	212.20	9995	34.1	13.5	1260	3000	613.0	583	17	3	1	14
PGLM1600PA	1988	I	12.10	9450	36.1	20.3	1210 *		43.0	482	3/17	1/1/2	3	
PGLM2500PE	1970	I	21.10	9790	34.9	18.7	1187	3600	67.0	516	16	2/6	2	1
PG5371(PA)	1958	I	26.30	11820	28.9	10.2	957	5100	124.0	483	16	2	1	12
PG7161F	1987	I	132.80	10510	32.5	13.5	1260 *	3600	403.0	594	17	3	1	10
PG6531(B)	1978	I	38.34	10860	31.4	11.8	1104	5100	139.0	539	17	3	1	10
PG7111(EA)	1976	I	83.50	10480	32.6	12.4	1104	3600	296.0	530	17	3	1	10
PGLM5000PC	1986	A	33.07	9720	35.1	30.0	1202	3600	125.0	449	5/14	1/1/2	3	1
GE Industrial														
LM5000PC	1986	A	34.30	9215	37.0	25.3	1174 *	3600	122.9	454	5/14	1/1/2	3	1
LM5000PC	1986	A	34.30	9240	36.9	26.0	1174 *	3000	126.1	451	5/14	1/1/2	3	1
LM2500PE	1970	A	21.30	9555	35.7	18.7	1212 *	3000	67.1	526	16	2/6	2	1
LM1600	1988	A	13.40	9450	36.1	21.5	1210 *		43.6	485	3/17	1/1/2	3	
LM2500PE	1970	A	22.00	9280	36.8	18.7	1212 *	3600	66.7	523	16	2/6	2	1
Mitsubishi														
MW-701		I	130.55	10070	33.9	14.0			453.0	513	19	4		8
MW-151		I	21.14	12170	28.0	11.0			93.0	518				
MF-111A		I	12.61	11250	30.3	13.0			49.0	547				
MF-111B		I	14.57	11020	31.0	15.0			55.0	535				
MW-191		I	17.79	13440	25.4	7.0					15	5		6
MW-501		I	104.57	10260	33.3	14.0			366.0	521	19	4		14
MW-251		I	41.38	10810	31.6	14.0			159.0	505	18	3		8
MW-501F		I	145.00	10000	34.1	14.2			423.0	571				
Nuovo Pignone														
PGT16			13.17	10200	33.5	21.5		7900	45.0	470				
PGT10	1986		9.98	11250	30.3	14.0	1063 *	7900	41.2	462	17	4		
MS3002	1952	I	10.45	14060	24.3	7.1	943 *	6500	52.3	526	15	2	2	
MS5001	1958	I	26.83	12340	27.7	10.2	957 *	5100	123.7	479	16	2	1	
LM5000	1986	A	33.76	6960	49.0	30.0		3600	122.0	450	5/14	1/1/2	3	
LM2500/30	1970	A	21.69	10230	33.4	18.0	1213 *	3600	66.7	513	16	2/6	2	
MS7001		I	83.50	11050	30.9	12.4		3600	642.0	530	17	3	1	
MS9001	1976	I	118.69	10800	31.6	11.6	1104 *	3000	405.8	524	17	3	1	
PGT25	1981		21.00	10190	33.5	18.0	1213 *	6500	66.7	513	16	4	2	
MS6001	1978	I	38.98	11380	30.0	11.5	1104 *	5100	137.9	537	17	3	1	
Siemens														
V94	1974	I	149.25	10135	33.7	10.6	1050 *	3000	511.0	545	16	4	1	
V84	1985	I	102.53	10150	33.6	10.6		3600	356.0	539	17	4	1	
V64		I	52.90	10000	34.1	15.8			172.0	536				
Solar														
Centaur H	1985		5.80	8270	41.3	10.2	1010 *	16520			11	3	1	1
Centaur H	1985		5.50	8655	39.4	9.6	904 *	15500			8	3	1	1
Jupiter	1968		23.11	7790	43.8	12.0								
Mars	1977		12.60	7700	44.3	15.7					15	2/2	2	1
Sulzer-Escher														
R3			5.91	10560	32.3	9.9	970	10600	33.3	365	17	3	1	4
3	1976		6.36	12230	27.9	10.0	970	10600	33.3	464	17	3	1	4
10	1981		20.90	7700	44.3	13.6	1120	7700			10	2/2	2	1
R7			10.30	10970	31.1	7.7	925	6400	64.1	342	13	6	1	9

Model	Year	Type	Power Aero /Ind MW	Heat Rate Btu/kWhr	Effy. %	rP	TRIT ° C	Speed	Exhaust kg/s	°C	Comp. Stages	Turb. Stages	Number Shafts	Number Comb.
Thomassen														
PG9161(E)	1979	I	116.40	10350	33.0	12.1	1104	3000	411.0	529	17	3	1	14
PG6531(B)	1981	I	37.40	10860	31.4	11.5	1104	5100	137.0	543	17	3	1	10
LM2500PE	1970	A	21.10	9790	34.9	18.7	1212	3600	67.5	516	16	2/6	2	1
LM2500PE	1970	A	20.40	10085	33.8	18.7	1211	3000	67.5	520	16	2/6	2	1
LM5000PC	1986	A	33.07	9860	34.6	30.0	1224	3000		454	5/14	1/1/2	3	1
LM5000PC	1986	A	33.07	9720	35.1	30.0	1224	3600	122.9	454	5/14	1/1/2	3	1
PG5371(PA)	1981	I	26.30	11820	28.9	10.2	957	5100	122.6	483	16	1/1	1	10
G3142(J)	1949	I	10.45	13320	25.6	7.1	943	6500	52.3	526	15	1/1	2	6
LM2500PD	1970	A	12.86	10040	34.0	14.8	1211	3000	56.7	415	16	2/6	2	1
LM2500PD	1970	A	12.86	9670	35.3	14.8	1212	3600	54.9	425	16	2/6	2	1
G5261(R)	1972	I	19.40	13260	25.7	8.1	938	5100	96.6	513	15	2	1	10
United Tech.														
V64.3			57.00	10250	33.3	14.0			174.0	550	16	3/3		
V84.2			103.00	10310	33.1	10.5			356.0	540				
FT8		A	25.40	8950	38.1	20.0			85.3	450				
Westinghouse Can.														
CW191PG		I	17.70	13390	25.5	7.5		4912	123.0	412	15	5		
CW251PG		I	42.50	10845	31.5	14.7		5426	162.0	512				
Westinghouse Elec														
501F		I	145.00	10000	34.1	14.0		3600	424.0	572	19	4		14
W-501D		I	104.40	10290	33.2	14.0		3600	367.0	521	19	4		14

Table D.2 Electrical Generation Model Combined Cycles (and Steam Injection)								
Company	Model	No. GT's	GT Type	GT MW	ST MW	Total MW	Heat rate Btu/kWhr	Eff %
ABB	KA M-1	1	MARS	8.6	4.0	12.6	7668	44.5
ABB	KA 35-1	1	GT35	17.8	7.5	25.3	7737	44.1
ABB	KA 10-1	1	GT10	20.8	10.5	31.3	7183	47.5
ABB	KA 8-1	1	GT8	45.3	24.8	70.1	7183	47.5
ABB	KA 11N-1	1	GT11N	78.1	43.0	121.1	7035	48.5
ABB	KA 13-1	1	GT13	94.0	50.2	144.2	7094	48.1
ABB	KA 13E-1	1	GT13E	142.1	71.7	213.8	6743	50.6
Alsthon	VEGA 105PA	1	5371PA	25.9	12.9	38.8	8140	41.9
Alsthon	VEGA 106B	1	6541B	37.7	18.3	56.0	7560	45.1
Alsthon	VEGA 205PA	2	5371PA	51.8	26.5	78.3	7094	48.1
Alsthon	VEGA 206B	2	6541B	75.4	39.1	114.5	7392	46.2
Alsthon	VEGA 109E	1	9161E	115.3	56.0	171.3	7134	47.8
Alsthon	VEGA 209E	2	9161E	230.6	114.3	344.9	7085	48.2
Fiat	CC 100	2	TG20	82.0	38.0	120.0	7370	46.3
Fiat	CC 300	2	TG20	248.0	116.0	364.0	7055	48.4
General Electric	S-106	1	MS6001B			56.8	7326	46.6
General Electric	S-206	2	MS6001B			116.6	7147	47.8
General Electric	S-107E	1	MS7001E			124.2	7048	48.4
General Electric	S-109E	1	MS9001E			173.4	6948	49.1

Company	Model	No. GT's	GT Type	GT MW	ST MW	Total MW	Heat rate Btu/kWhr	Eff %
General Electric	S-207E	2	MS7001E			250.5	6988	48.8
General Electric	S-109F	1	MS9001F			320.5	6620	51.6
General Electric	S-209E	2	MS9001E			349.3	6899	49.5
Mitsubishi	MPCP1(501)	1	MW501	98.7	45.3	144.0	7260	47.0
Mitsubishi	MPCP1(701)	1	MW701	119.6	56.2	175.8	7390	46.2
Mitsubishi	MPCP2(501)	2	MW501	197.4	92.5	289.9	7215	47.3
Mitsubishi	MPCP2(701)	2	MW701	239.2	114.2	353.4	7360	46.4
Mitsubishi	MPCP3(501)	3	MW501	296.1	139.4	435.5	7200	47.4
Mitsubishi	MPCP3(701)	3	MW701	358.8	172.1	530.9	7350	46.4
Mitsubishi	MPCP4(501)	4	MW501	394.8	186.2	581.0	7200	47.4
Mitsubishi	MPCP4(701)	4	MW701	478.4	229.8	708.2	7340	46.5
Mitsui Engineering	SB30	1	SB30	5.6	2.6	8.3	8968	38.1
Mitsui Engineering	SB60	1	SB60	12.3	5.4	17.7	8230	41.5
Mitsui Engineering	SB90	1	SB90	14.7	9.5	24.2	8706	39.2
Mitsui Engineering	SB120	1	SB120	22.3	10.5	32.8	8048	42.4
Nuovo Pignone	CC-210	2	PGT 10	19.0	9.3	28.3	8100	42.1
Nuovo Pignone	CC-106	1	MS6001	37.9	19.5	57.4	7680	44.4
Nuovo Pignone	CC-205	2	MS5001	51.4	29.5	80.9	8100	42.1
Nuovo Pignone	CC-206	2	MS6001	75.8	42.2	118.0	7500	45.5
Nuovo Pignone	CC-109	1	MS9001	114.5	60.0	174.5	7250	47.1
Nuovo Pignone	CC-209	2	MS9001	229.0	123.0	352.0	7200	47.4
Siemens, KWU	GUD 1.84	1	V84	95.0	55.0	150.0	6700	50.9
Siemens, KWU	GUD 1.94	1	V94	140.0	85.0	225.0	6680	51.1
Siemens, KWU	GUD 2.84	2	V84	190.0	115.0	305.0	6670	51.2
Siemens, KWU	GUD 2.94	2	V94	280.0	170.0	450.0	6640	51.4
Siemens, KWU	GUD 3.84	3	V84	285.0	175.0	460.0	6620	51.6
Siemens, KWU	GUD 3.94	3	V94	420.0	260.0	680.0	6600	51.7
Solar	Centaur T-4500	1	T4500	3.0	1.2	4.2	9610	35.5
Solar	Centaur H	1	TYPE H	3.8	1.6	5.3	9030	37.8
Solar	Mars	1	MARS	8.7	3.3	12.0	8000	42.7
Stewart & Stevenson	CCI-501KB5	1	Alison K	3.7	1.6	5.3	8241	41.4
Stewart & Stevenson	CCI-571K	1	Alison 5	5.6	2.1	7.7	7730	44.2
Stewart & Stevenson	CCI-1600	1	LM1600	13.4	4.6	18.0	7127	47.9
Stewart & Stevenson	CCI-2500	1	LM2500	22.2	8.2	30.5	6865	49.7
Stewart & Stevenson	CCI-5000	1	LM500	33.8	10.6	44.4	7109	48.0
Thomassen	LM125-SP	1	LM2500PE	20.1	8.0	28.1	7210	47.3
Thomassen	LM125-DP	1	LM2500PE	20.1	8.9	29.0	6960	49.0
Thomassen	LM150-SP	1	LM5000PC	32.1	9.2	41.3	7470	45.7
Thomassen	106B-SP	1	PG6531B	36.9	17.7	54.6	7520	45.4
Thomassen	106B-DP	1	PG6531B	36.9	20.1	57.0	7200	47.4
Thomassen	LM225-DP	2	LM2500PE	42.0	18.0	60.0	6800	50.2
Thomassen	109E-SP	1	PG9161E	114.7	49.7	164.4	7400	46.1
Thomassen	109E-DP	1	PG9161E	114.7	57.3	172.0	7080	48.2
Thomassen	STIG LM2500PE	1	LM2500PE	24.0		24.0	9270	36.8
Thomassen	STIG LM2500PH	1	LM2500PH	26.3		26.3	8910	38.3
Thomassen	STIG LM5000PC	1	LM5000PC	51.4		51.4	8080	42.2
United Technologies	FT8	1	FT8	24.8	8.4	33.2	6815	50.1
United Technologies	FT8	2	FT8	49.9	17.0	66.9	6765	50.5
United Technologies	V84.2	1	V84.2	100.0	55.0	155.0	6700	50.9
Westinghouse	PACE 200(F)	1	501F	136.2	63.8	200.0	6800	50.2
Westinghouse	PACE 300	2	W501D5	210.5	96.5	307.0	7000	48.8