CRANFIELD UNIVERSITY

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TECHNOECONOMIC EVALUATION OF TRIGENERATON PLANT: GAS TURBINE PERFORMANCE, ABSORPTION COOLING AND DISTRICT HEATING

SCHOOL OF MECHANICAL ENGINEERING

PhD THESIS

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Technoeconomic Evaluation of Trigeneraton Plant:
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Supervisor: P. Pilidis

November 2006

This thesis is submitted for the degree of PhD

Dedicated to my Dad and Mum

Liverio and Mirsini

ABSTRACT

This PhD thesis is a demand led study taking into account changes in ambient conditions and power settings of a tri-generation power plant. Includes an evaluation tool for combined heat, cooling and power generation plant. The thesis is based on an overall technical-economic analysis of the tri-generation system, including:

1. Energy demand analysis and evaluation of actual tri-generation case studies.
2. Modelling of the prime mover (Gas Turbine, GT)
3. Modelling of the absorption cooling system, (LiBr/Water).
4. Economic analysis and evaluation of the entire tri-generation plant.

Initially, the main effort is to carry out research concerning the energy demands of different actual cases. The research includes sourcing, collecting, classification and evaluation of the available information. The cases cover a wide range of economic life and the resulting data specifies the energy needs which the purposed tri-generation power plant needs to cover.

The second part deals with the prime mover (namely the Gas Turbine, GT) modelling and simulation. The technical part of the assessment includes the Design Point (DP) and Off Design (OD) analysis of the GT. In other words, the performance analysis simulates different thermodynamic cycles (Simple, or with Heat Exchanger), and different configurations (one or two shafts). Also, the computer programming code is capable of simulating the effects of the use of different types of fuel, ambient conditions, part load conditions, degradation, or the extraction of power for district heating or for absorption cooling.

The third part includes the simulation of the absorption cooling system alone and/or in co-operation with the prime mover. The simulation is based upon the premise that the original prime mover is replaceable.

Finally, an evaluation methodology of tri-generation plants, is introduced taking into account, both technical facts and economic data -based on certain cases from Greek reality- helping the potential users to decide whether it is profitable to use such technology or not. The economic scene will include the basic economic facts such as initial cost, handling and operational cost (fuel prices, maintenance etc), using methodology based on Net Present Value (NPV).

This thesis suggests several tri-generation technology modes. The more economic favourable than the conventional technology is the 2-shaft simple cycle mode for the cases of international airport (12MW total power demand) and the isolated island (120MW), while the 1-shaft simple cycle mode is the more economic favourable for the case of hotel (1MW).

The main contribution of the thesis is that it provides an intergraded realistic tool, which simulates the future operation (technical and economic) of a trigeneration plant, capable of helping the potential investor decide if it is profitable to proceed with the investment.

ACKNOWLEDGMENTS

I would like to thank my supervisor Prof. Pilidis, the academic and administration staff of Cranfield University for their support, help and guidance throughout those years.

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SYMBOLS and NOTATION

A	ALTITUDE
APP	AUTONOMOUS POWER PLANTS
C	MASS FACTOR
C_P	SPECIFIC HEAT
CCPP	COMBINED CYCLE POWER PLANT
CFCS	CHLOROFLUOROCARBONS
CHCP	COMBINED HEAT COOLING AND POWER
CHP	COMBINED HEAT AND POWER
CO	CRUDE OIL
COP	COEFFICIENT OF PERFORMANCE
CW	COMPRESSOR WORK
D	DIFFERENCE
DEPA	PUBLIC ENTERPRISE OF GAS S.A
DLE	DRY LOW EMISSIONS SYSTEMS
DP	DESIGN POINT
ELPE	GREEK OILS S.A
EU	EUROPEAN UNION
F	CIRCULATION FACTOR
FAA	FEDERAL AVIATION ADMINISTRATION
FAR	FUEL AIR RATIO
FCV	FUEL CALORIFIC VALUE
H	ENTHALPY
HC	HYPOCARBONS
HCV	HIGH CALORIFIC VALUE
HE	HEAT EXCHANGER
HI	HEAT INPUT
HTSO	HELLENIC TRANSMISSION SYSTEM OPERATOR
GT	GAS TURBINE
IATA	INTERNATIONAL ASSOCIATION OF TRANSPORT AIRLINES
ICAO,	INTERNATIONAL CIVIL AVIATION ORGANIZATION
ISA	INTERNATIONAL STANDARD ATMOSPHERE
ISO	INTERNATIONAL ORGANIZATION OF STANDARDIZATION
LAT	LATITUDE
LNG	LIQUEFIED NATURAL GAS
LON	LONGITUDE
M	MACH NUMBER
$\dot{m}$	MASS FLOW
NG	NATURAL GAS
NPV	NET PRESENT VALUE
OD	OFF DESIGN
O&M	OPERATION AND MAINTENANCE
P	STATIC PRESSURE
PPC	PUBLIC POWER COMPANY
Q	HEAT
R	PRESSURE RATIO
RAE	ENERGY REGULATION AUTHORITY
RSE	RENEWABLE SOURCES OF ENERGY
RT	TONS OF REFRIGERATION
SCR	SELECTIVE CATALYTIC REDUCTION
SFC	SPECIFIC FUEL CONSUMPTION
SW	SPECIFIC POWER
T	STATIC TEMPERATURE
TET	TURBINE ENTRY TEMPERATURE
TW	TURBINE WORK
UW	USEFUL WORK
V	SPECIFIC VALUE

VIP	VERY IMPORTANT PERSON
X	MASS FRACTION

SUBSCRIPTS

B	BOILER
C	COLD
C	COMPRESSOR
CC	COMBUSTION CHAMBER
COOL	COOLING
CT	COMPRESSOR TURBINE
D	DESORBER
DE	DEGRADATION
E	EVAPORATOR
EXH	EXHAUST
F	FUEL
H	HOT
H	HIGH TEMPERATURE
HEC	HEAT EXCHANGER COLD PART
HEH	HEAT EXCHANGER HOT PART
HP	HEAT PUMP
HYP	HYPOTHETIC
IN	INTAKE, INLET
IS	ISENTROPIC
L	LOW TEMPERATURE
LOSS	LOSSES
O	TOTAL OR STAGNATION
POL	POLYTROPIC
PR	PRICE
PT	POWER TURBINE
RA	ABSORPTION SYSTEM
RE	REFRIGERATION
RE	REAL
T	TURBINE

SUPERSCRIPT

DP	DESIGN POINT
OD	OFF DESIGN

GREEK

A	AMBIENT CONDITION
Γ	GAMMA
Δ	DIFFERENCE
H	EFFICIENCY
N	SPECIFIC VOLUME DENSITY
P	DENSITY

1. INTRODUCTION

1.1 Aim of the Thesis

The target of this thesis is to explore the technical and economic principles and the applications of trigeneration technology. The thesis provides a powerful computational (using FORTRAN as programming language) tool, which simulates in a realistic way:

1. the technical operation of the prime mover (GT), at the design point and off design point conditions, the absorption cooling system in a modular or intergraded way, and
2. the economic behaviour and the perspectives of a trigeneration power plant.

In effect, it aims to help policymakers, potential investors and other professionals to understand and evaluate this energy saving potential solution, which is now receiving a great deal of positive attention, both for its energy efficiency and environmental benefits.

1.2 What is trigeneration?

The usual (conventional) way to cover needs in electricity and heat, is to purchase electricity from the local grid and generate heat by burning fuel in a boiler, a furnace, etc. However, a considerable decrease in total fuel consumption is achieved, if cogeneration (known also as **Combined Heat and Power, CHP**) is applied.

Cogeneration is the thermodynamically sequential production of two or more usuable forms of energy from a single primary energy source.

The two most usual forms of energy are mechanical and thermal energy. Mechanical energy is usually used to drive an electric generator. This is why the following definition, though restrictive, often appears in the literature:

Cogeneration is the combined production of electrical (or mechanical) and useful thermal energy from the same primary energy source.

The mechanical energy produced can be used also to drive auxiliary equipment, such as compressors and pumps. Regarding the thermal energy produced, it can be used either for heating or for cooling. Cooling is effected by an absorption unit, which can operate through hot water, steam or hot gases.

So, **trigeneration (Combined Heat Cooling and Power, CHCP)** which is actually an extension of the CHP system for cooling production, can be defined as the conversion of a single fuel source into three energy products: electricity, steam or hot water and chilled water, resulting in lower pollution and greater efficiency than producing the three products separately

In recent years district cooling has been considered in many locations as a method for meeting the space cooling requirements of buildings in the residential, commercial and, at times, industrial sector. It is particularly suitable in urban areas with high-density arrangement offices and residential dwellings requiring air conditioning.

In this application absorption chillers are often favoured because they don't use chlorofluorocarbons and they can be used in conjunction with cogeneration systems for thermal and electrical energy. The chilling equipment can be based centrally, with chilled water piped to users, or can be located on the premises of the user. The most economic choice depends on the application and geographical distribution.

District cooling systems using absorption chillers often complement district heating systems, when both use heat supplied from a cogeneration plant. The heat demand in summer is lower than in winter and heat-driven district cooling, which requires the heat mainly in summer, can help to balance the seasonal demands for cogenerated heat. This increases the overall efficiency of the cogeneration system and therefore increases the environmental and other benefits that the system could bring.

District cooling is a recent concept, but is already relatively widely used in the USA and Japan. In Europe, there is awareness of the technology, but there is certainly less experience –with the possible exception of Sweden. An additional barrier that these systems face in Europe, apart of the fact that installing cooling increases the initial costs of the system considerably, is that the most suitable applications will be found in the South of Europe, which means, in countries where there is less experience of district heating (and where networks would have to be built), and hence less history among consumers or suppliers of the provision of this type of central energy.

During the operation of a conventional power plant, large quantities of heat are released in the atmosphere either through the cooling circuits (steam condensers, cooling towers, water coolers in Diesel or Otto engines, etc.) or through the exhaust gases. Most of this heat can be recovered and used to cover thermal or cooling needs (depending on the application demands), thus increasing the efficiency from 30-50% of a power plant to 80-90% of a trigeneration system.

A trigeneration system encompasses a range of technologies, but will usually include a prime mover, an electricity generator a heat recovery system and an absorption cooling system.

In conventional electricity generation, further losses of around 5-10% are associated with the transmission and distribution of electricity from relatively remote power stations via the electricity grid. These losses are greatest when electricity is delivered to smaller or isolated consumers.

The electricity generated by the trigeneration plant is normally used locally, and so transmission and distribution losses are negligible. Trigeneration therefore offers energy savings ranging between 15-40% when compared against the supply of electricity and heat from conventional power stations and boilers. Because transporting electricity over long distances is easier and cheaper than transporting heat, cogeneration installations are usually sited as near as possible to the place where the heat is consumed and, ideally, are built to a size to meet the heat demand. Otherwise an additional boiler will be necessary, and the environmental advantages will be partly hindered. This is the central and most fundamental principle of cogeneration.

When less electricity is generated than needed, it will be necessary to buy extra. However, when the scheme is sized according to the heat demand, normally more electricity than needed is generated. The surplus electricity can be sold to the grid or supplied to another customer via the distribution system (wheeling).

1.3 The benefits of trigeneration

Provided the trigeneration is optimised in the way described above (i.e. sized according to the heat or cooling demand), the following benefits arise:

- Increased efficiency of energy conversion and use. A well-designed and operated cogeneration scheme will always provide better energy efficiency than conventional plant, leading to both energy and cost savings. A single fuel is used to generate heat and electricity, so cost savings are dependent on the price-differential between the primary energy fuel and the bought-in electricity.However, although the profitability of trigeneration generally results from its cheap electricity, its success depends on using recovered heat productively, so the prime criterion is a suitable heat or cooling requirement. As a rough guide, cogeneration is likely to be suitable where there is a fairly constant demand for heat or cooling for at least 4,500 hours in the year. The timing of the site's electricity demand will also be important as the trigeneration installation will be most cost effective when it operates during periods of high electricity tariffs, that is, during the day. At current fuel prices and electricity tariffs, and allowing for installation and life-cycle maintenance costs, payback periods of three to five years can be achieved on many cogeneration installations.
- Lower emissions to the environment, in particular of CO_2, the main greenhouse gas.
- In some cases, where there are biomass fuels and some waste materials such as refinery gases, process or agricultural waste (either anaerobically digested or gasified), these substances can be used as fuels for cogeneration schemes, thus increasing the cost-effectiveness and reducing the need for waste disposal.
- Large cost savings, providing additional competitiveness for industrial and commercial users, and offering affordable heat or cooling for domestic users.
- An opportunity to move towards more decentralised forms of electricity generation, where a plant is designed to meet the needs of local consumers, providing high efficiency, avoiding transmission losses and increasing flexibility in system use. This will particularly be the case if natural gas is the energy carrier.
- Improved local and general security of supply -local generation, through trigeneration, can reduce the risk of consumers being left without supplies of electricity and/or heating and/or cooling. In addition, the reduced fuel need which cogeneration provides reduces the import dependency- a key challenge for Europe's energy future.
- An opportunity to increase the diversity of generation planst, and provide competition in generation. Trigeneration provides one of the most important vehicles for promoting liberalisation in energy markets.

1.4 Where is trigeneration suitable?

In recent years the greater availability and wider choice of suitable technology has meant that cogeneration has become an attractive and practical proposition for a wide range of applications. These include the process industries, commercial and public sector buildings and district heating schemes, all of which have considerable heat demand. These applications are summarised in the list below.

Industrial

Pharmaceuticals and fine chemicals, paper and board manufacture, brewing, distilling and malting, ceramics, brick, cement, food processing, textile processing, minerals processing, oil refineries, iron and steel, motor industry, horticulture and glasshouses, timber processing.

Buildings

District heating, hotels, shopping centers, hospitals, leisure centres and swimming pools, college campuses and schools, airports, prisons, police stations, barracks etc., supermarkets and large stores, office buildings, individual houses.

Islands

Isolated, or small tourist destination islands.

National Grid Supply

It can perform as a base load unit with high overall efficiency.

1.5 Thesis description

Chapter 2 includes the results of analytical energy and power research, concerning five actual Greek cases studies. An airport, a large island, a small island, a group of resorts hotels in the North Greece and a hotel in southern Greece. In addition, an overview of the Greek national energy scene is presented.

Chapter 3 concerns the Gas turbine modelling and simulation of the design point and off design performance. The configurations of the engines are simple cycle and cycle with heat exchanger with both one shaft and two shafts.

Chapter 4 provides a simulation tool of the LiBr/Water absorption cooling system, which cooperates with the GT.

Chapter 5 presents an economic overview of the entire project. Based on realistic economic data and potential operating modes, is giving the opportunity to the future investor to make a first estimation of the profitability of his investment in trigeneration.

Chapter 6 concludes the thesis and provides suggestions for future developments and potential optimisations.

2. ENERGY DEMAND

2.1 Introduction
The purpose of this chapter is to identify the energy demands of different cases in which it is possible to apply a tri-generation power unit (electrical, heating, cooling power).

The **selection** of these case studies is based on the following **criteria**:
1. Usage of a relatively small or medium tri-generation power unit (**10KW- 300MW**), having a gas turbine as a prime mover.
2. **Different kinds of applications**. A variety of applications have been chosen to cover the varying energy needs.
3. Various locations**,** in other words, **various climatic conditions.** The climatic conditions are one of the most important factors that form the energy needs of the application.

The **case studies** which have been chosen (covering four-order of magnitude e.g. 0.5-1.5-15-150MW of total power demand) are:
1. The new International "Macedonia" Airport of Thessaloniki, Greece
2. Lemnos island
3. Rhodes island
4. Hotel in Rethimno-Crete
5. Hotel in northern Greece

The study of each case was based on a fundamental methodology, which was modulated relatively to each case. The fundamental **methodology** consists of the following **steps**:
1. **Collection of the available statistical data** (the source was the Public Power Company, [PPC] or the owner company of the application).
2. **Statistic processing.** (sorting out the previous years energy bills, identification of the uniqueness of each application, extrapolation of the missing data, etc)
3. **Adjustment to the modification** (regular and random) **of the energy consumption**
4. **Adjustment to the climatic data of the region** (temperature, cloudiness, etc).
5. **Adjustment to the year's season** (variation of night and day hours)
6. During the heating calculation, except for electricity-based technologies used for heating, such as inverters we took into account, **heating technologies** such as **central heating systems**, using diesel fuel**.**
7. The calculation of the energy consumption and the power demand is based on a **typical day** per month (the average day per month).
8. In order to cover the **worst** case in energy and power, the values, coming from the 6th step, are multiplied by a coefficient 1.2 (20% increase)
9. Finally, in order to cover the **future increase** in energy consumption for the next 10 years or at least for a period of time exceeding the payment period of the investment, the values, coming from the 8th step, should be multiplied by a coefficient 1.1 (10% increase).

At APPENDIX A, a general analysis of the energy system presenting the energy situation and policy of a EU country, namely **Greece,** is carried out. The analysis assists in understanding the energy and the economic status in which the trigeneration plant will operate and concerns the following issues:

- **Raw material** (lignite, natural gas, renewable sources of energy)
- **Power units** (steam plants, plants using internal combustion engine, Combined Cycle Power Plant, CCPP)
- **Production - Sales** (history background, future development potentials)
- **Distribution network** (connected and independent network)
- **Interconnections with abroad** (major connections with the neighbouring countries)
- **Energy disposal relative to the usage** (commercial, industrial, domestic, public, agricultural)
- **Handling of peak loads** (significant events such as Olympic Games, peak hours during summertime, etc)
- **Development plans** and **new technologies** which are economical and environmentally friendly.

2.2 The new International "Macedonia" Airport of Thessaloniki, Greece

<u>2.2.1 General description of the airport</u>

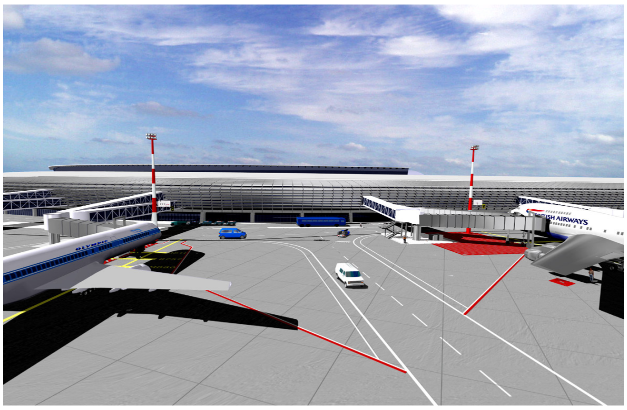

Fig.2.1: Animation aspects of International "Macedonia" Airport [8]

The new International "Macedonia" Airport of Thessaloniki (*Fig. 2.1*) has the following operational characteristics:

- It services 8,000,000 passengers per year (IATA B service level). The estimation at peak hours is 2,800 passengers either for arrivals or for departures, while the total number of passengers arriving or departing, is estimated at peak hours to be 3,500.
- There are 100 check in counters, two bulky counters with full automatic systems CUTE and CUSS leading to a full automatic separation and distribution system of the luggage where the security check takes place (Hold Baggage Screening 100%). There are also 8 luggage belts for the arriving passengers.
- A recently extended manoeuvre airfield for 36 airplanes and creation of new airplane parking places (E and F class), 14 of which are parked at contact places. The new airport offers 14 double-boarding bridges.
- A first and second order national road network has been constructed and the fields surrounding the airport have been formed.
- The operational system of the new International "Macedonia" Airport of Thessaloniki, fulfils all the specifications of international organisations (ICAO, FAA, IATA) for the security of flights, while it has centralized security systems and access control which gives graduate licence access to the personnel.

The structured area of the airport is distributed as follows:

- Public-passenger-check area 55,500m^2
- VIPs area: 3,000m^2
- Commercial and catering services: 8,000m^2

* Offices: 8,000m^2
* Luggage distribution area: 11,250m^2
* Personal dressing rooms: 3,750m^2
* Traffic areas: 17,500m^2
* Electromechanical installation: 8,000m^2

Total 115,000m^2

2.2.2 Climatic data analysis

The main climatic data, which affect the energy design of the airport, are the air temperature, solar radiation, wind and humidity (APPENDIX B.1):

* Air temperature: Affects the calculation of the heating and cooling loads.
* Solar radiation: Affects the heating, cooling and lighting loads.
* Wind: Affects heating, and cooling loads through the calculation of the losses (ventilation losses during the wintertime, or the cooling during summertime).
* Humidity: Affects the calculation of the heating and cooling loads.

The climate of the region of Thessaloniki, is "Mediterranean", i.e. hot summers and mild winters (*Table 2.1*).

Table 2.1: Climatic data (Thessaloniki region, East Longitude (Lon) 22^o 58΄/ North Latitude (Lat) 40^o 31΄

MONTH	BAROMETRIC PRESSURE mmhg	AVERAGE AIR TEMPERATURE ^oC	ABSOLUTE MAXIMUM AIR TEMPERATURE ^oC	ABSOLUTE MINIMUM AIR TEMPERATURE ^oC	HOURS OF SUNLIGHT h	RELATIVE HUMIDITY %	AVERAGE CLOUDINESS (scale of 8)
1	1,019.1	5.2	20.8	-14	91.6	76.1	4.7
2	1,017.9	6.7	23.2	-12.8	94.8	73	4.8
3	1,016.6	9.7	25.8	-7.2	150.2	72.4	4.9
4	1,013.3	14.2	31.2	-1.2	203.5	67.8	4.4
5	1,013.9	19.6	36	3	267.2	63.8	4.1
6	1,013.1	24.4	39.8	6.8	288.6	55.9	3.2
7	1,012.8	26.6	42	9.6	320.4	53.2	2.2
8	1,013.4	26	40.4	8.2	263.8	55.3	2.1
9	1,016.4	21.8	36.2	2.6	221	62	2.7
10	1,018.9	16.2	31.6	-1.4	161.8	70.2	3.9
11	1,018.6	11	26.6	-6.2	121	76.8	4.7
12	1,018.1	6.9	22.6	-9.2	102.9	78	4.8
AVER		**15,69**			**191**	**67.04**	

2.2.3 Presentation of today's Greek Airport energy needs

Airports in general have high-energy consumption. Also, the significant variation of the population in the building in short periods of time affects the energy behavior of the building. Also, the airport buildings are very interesting because of the architectural particularities (transparent surfaces).

In general, an energy analysis of the 14 biggest Greek airports reveals:

- Average consumption heating energy: 68kWh/m^2
- Average consumption lighting and motion energy: 172kWh/m^2
- Total consumption of energy: 240kWh/m^2

Electric consumption is mainly due to the air-conditioning, lighting of the internal space of the building, lighting of the airfield and the electromechanical installation. Particularly, it has been evident that the major part of the installed power is used for the central air-conditioning system while the 1/3 and 1/5 of the installed power is used for the mechanical installation and the lighting respectively.

2.2.4 Energy needs of the new main building and the airfield

Some of the requirements needed for the energy analysis, are the following:

- Regional climatic conditions.
- Energy analysis of the building structure (heat-insulation, bioclimatic design, etc).
- Analysis of the ventilation conditions.
- Sunshine / shading analysis during the year.
- Considerations of comfort of the passengers.

At this point, it must be stressed, that **the energy analysis also contains the energy consumption of the airfield**.

Cooling

The cooling analysis is adapted to the given that the cooling area is restricted to a height less than 2.5-3m, and the average internal summer temperature is regulated to be 25^{o}C.

The procedure, which leads to the final cooling consumption energy and the cooling power needed, is presented in *Table B.1* and *Fig. B.4, B.5* (APPENDIX B.1)

Heating

The heating analysis is based on the following assumptions:

- It does not take into account the heating coming from lights, electromechanical installation, existence of people and heating derived from the sun. The heating is coming exclusively from the boiler.
- Internal temperature during winter 22^{o}C.
- The operation of the heating system during the months June, July, August is restricted to heating water purposes only.
- The operation hours of the heating system are shown in *Table 2.2*

Table 2.2: Operation hours of the heating system for a typical day per month

Months	1	2	3	4	5	6	7	8	9	10	11	12
Operation hours per day	18	16	11	6	1.5	0.1	0.1	0.1	1	6	7	15

The procedure, which leads to the final heating consumption energy and the heating power needed, is presented in APPENDIX B.1.

Lighting-Motion

The lighting level is controlled by photoelectric cells. Thus, the natural lighting coming from the architectural design of the building is taken advantage of. The electrical network includes:

- Main conductors: 20KV / 50Hz
- 400V / 50Hz network, including UPS and generators
- 400Hz electrical aircraft supply network
- Lighting of the aircraft parking field
- Security and perimeter lighting
- Lighting and signaling of the runways
- Electric motors
- Motion belts
- Escalators

The procedure, which leads to the final lighting consumption energy, is presented in APPENDIX B.1.

Table 2.3: Airport, power demand in MW, 2003, (typical day)

MONTHS 30 days per month	**HEATING** (MW_t)	**COOLING** (MW_c)	**LIGHTING & MOTION** (MW_e)	**TOTAL POWER**
JAN	6.750	2.313	1.831	10.894
FEB	6.000	3.333	1.778	11.111
MAR	4.125	6.021	1.752	11.898
APR	2.250	7.325	1.698	11.273
MAY	0.563	8.688	1.700	10.951
JUN	0.038	9.958	1.645	11.641
JUL	0.038	9.938	1.640	11.616
AUG	0.038	9.908	1.651	11.597
SEP	0.375	9.125	1.679	11.179
OCT	2.250	7.350	1.747	11.347
NOV	2.625	5.692	1.838	10.155
DEC	5.625	3.188	1.859	10.672

Fig.2.2: Airport power demand in MW, 2003, (typical day)

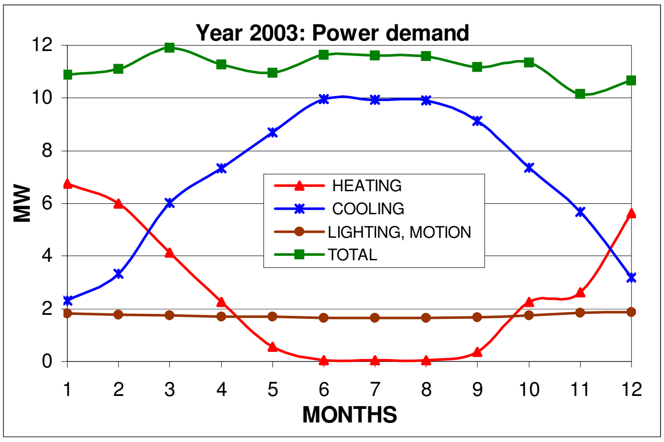

Table 2.4: Airport energy demand in MWh, 2003, (typical day)

MONTHS 30 days per month	**HEATING** (MWh_t)	**COOLING** (MWh_c)	**LIGHTING & MOTION** (MWh_e)	**TOTAL ENERGY**
JAN	162	55.5	43.951	261.451
FEB	144	79.99	42.680	266.679
MAR	99	144.49	42.046	285.545
APR	54	175.8	40.753	270.553
MAY	13.5	208.5	40.791	262.791
JUN	0.9	238.99	39.479	279.378
JUL	0.9	238.5	39.352	278.752
AUG	0.9	237.79	39.631	278.33
SEP	9	219	40.295	268.295
OCT	54	176.4	41.940	272.34
NOV	63	136.59	44.101	243.699
DEC	135	76.5	44.618	256.118

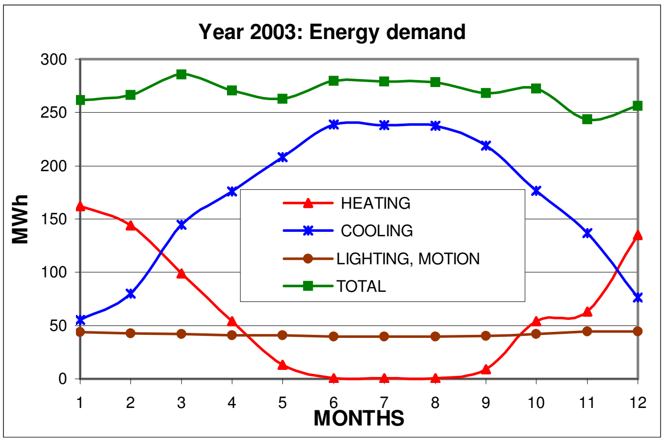

Fig.2.3: Airport energy demand in MWh, 2003, (typical day)

2.3 Study of energy needs of island Lemnos

2.3.1 General description of island Lemnos

Lemnos, with extent of 476km^2, is the second largest island in the Lesvos prefecture, in the northern Aegean Sea. Its total population, at last census (1991), is 17,645 residents. (*Fig. 2.4*) The capital of island and main harbour is Myrina. Myrina is located 84 naval miles from Mitilini and 160 naval miles from Thessaloniki.

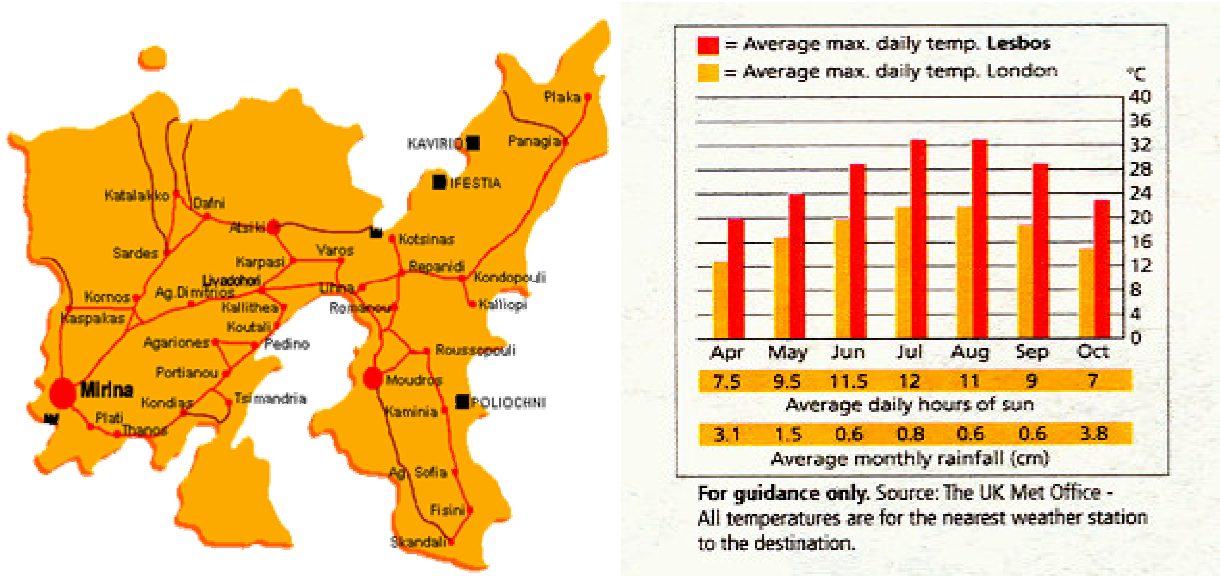

Fig.2.4: The island of Lemnos

2.3.2 Climatic conditions of Lemnos

The climate is Mediterranean, that is to say, mild winter and hot summertime. Climatic data for each month is analytically reported in *Table 2.6*

Table 2.5: Climatic data (Lemnos, East Longitude (Lon) 25^{o}04'/ North Latitude (Lat) 39^{o}53')

MONTH	BAROMETRIC PRESSURE mmhg	AVERAGE AIR TEMPERATURE ^{o}C	ABSOLUTE MAXIMUM AIR TEMPERATURE ^{o}C	ABSOLUTE MINIMUM AIR TEMPERATURE ^{o}C	HOURS OF SUNLIGHT h	RELATIVE HUMIDITY %	AVERAGE CLOUDINESS (scale of 8)
1	1,019.4	7.4	18.8	-5.2	82	76.8	4.8
2	1,018.3	7.7	19	-5.8	110.7	74.8	4.7
3	1,016.8	9.7	22	-6	162.1	75.1	4.1
4	1,013.6	13.6	25.8	0	221.7	73.8	3.5
5	1,014.0	18.4	31	3	294.4	68.7	2.9
6	1,012.8	23.4	34.4	3.4	326.7	60.4	1.8
7	1,012.6	25.6	39.4	12	344.7	57.2	0.9
8	1,013.1	24.9	35.8	12	338.4	61.7	0.9
9	1,016.1	21.4	32.8	8.2	264.9	66.3	1.5
10	1,018.5	16.7	31.8	1.6	197.8	73.2	3.0
11	1,018.8	12.2	24	-2	127.6	77.8	4.4
12	1,018.4	9.2	19.2	-3.6	94.6	78.6	4.8
AVER.		**15.4**			**256.6**		

2.3.3 Characteristics of Electric Generation of in the Greek Islands in general

The electricity generation of Lemnos faces the same problems and has the same peculiarities as the rest of the Greek islands. The peculiarities that could be pinpointed are:

- The lack of extensive interconnections with a bigger electric system, for a high reliability factor. There are many disturbances during high peak periods.
- The relative isolation of Autonomous Power Plants (APP) from the industrial centers of the country which results in increased operation cost.
- Due to the explosive tourist growth of most Aegean islands, the Power Network has to face extraordinarily high rates of increase of demand in electric energy. Annual rates of increase of demand 10% and more are common. This, in combination with the high cost of exploitation, impedes the financing of new investments and extensions of equipment and networks.
- A lot of cases, the inhabited regions or those that are focused on tourist activities were developed so much, that they reached limits of APP. This fact creates frictions with the neighbours and in a lot of cases renders the discovery of locations impossible for the required extension reinforcement of the local grid. The problem is intensified by the lack of government-owned intervention for the arrangement of uses of land in the islands.
- Because the Aegean islands usually have high wind potential, the Public Power Company, PPC, since the beginning of 1980, has begun to produce wind power. Recently this fact began to attract the interest of private investors for wind generation of electricity. The electricity production cost for Lemnos, is estimated at 0.60Euro/KWh (2003).

Notice:
The 1st column in *Tables 2.6* and *2.7* includes the electricity supplies the overhead lighting, PCs, electric devices, air-conditioners (cooling) electric inverters, electric heaters (part of heating), etc in MWh.

More data (years 2001, 2002) are available in APPENDIX B.2

Table 2.6: Lemnos energy demand in MWh, 2003, (typical day)

YEAR: *2003*	COOLING MWh_c	LIGHTING & OTHER MWh_e	HEATING MWh_t		TOTAL ENERGY
			ELECTR.	BOILER	
JAN	0	109.13	36.37	72.76	218
FEB	0	119.45	26.25	52.41	198
MAR	8.99	116.88	23.93	48.00	198
APR	13.39	104.42	16.09	32.10	166
MAY	28.32	78.79	15.99	32.02	155
JUN	37.43	103.64	2.83	5.81	150
JUL	51.23	128.07	3.70	7.28	190
AUG	59.02	131.82	5.86	11.85	209
SEP	36.72	96.04	8.44	16.98	158
OCT	20.79	88.35	20.76	41.61	172
NOV	0	106.90	26.70	53.48	187
DEC	0	115.26	40.54	80.95	237

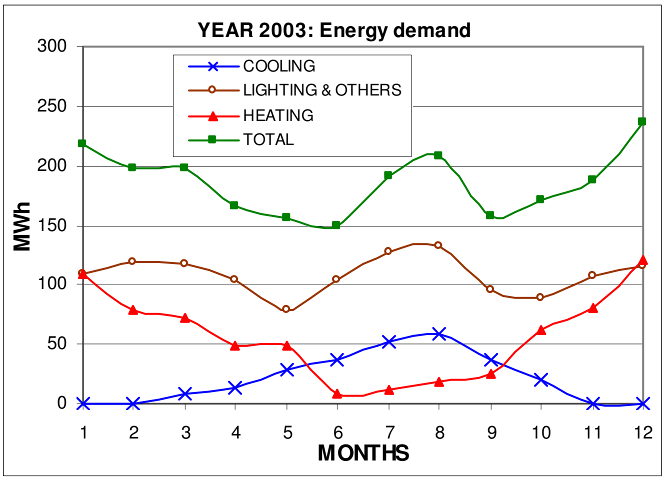

Fig.2.5: Lemnos energy denand in MWh, 2003, (typical day)

Table 2.7: Lemnos power demand in MW, 2003, (typical day)

YEAR: *2003*	COOLING MW_c	LIGHTING & OTHER MW_e	HEATING MW_t		TOTAL POWER
			ELECTR.	BOILER	
JAN	0	4.55	1.51	3.04	9.09
FEB	0	4.98	1.09	2.19	8.25
MAR	0.37	4.87	1.00	2.00	8.24
APR	0.56	4.35	0.67	1.34	6.92
MAY	1.18	3.28	0.67	1.33	6.46
JUN	1.56	4.32	0.12	0.24	6.24
JUL	2.13	5.34	0.15	0.31	7.93
AUG	2.46	5.49	0.25	0.49	8.69
SEP	1.53	4	0.35	0.71	6.59
OCT	0.87	3.68	0.86	1.74	7.15
NOV	0	4.45	1.12	2.22	7.79
DEC	0	4.8	1.69	3.37	9.86

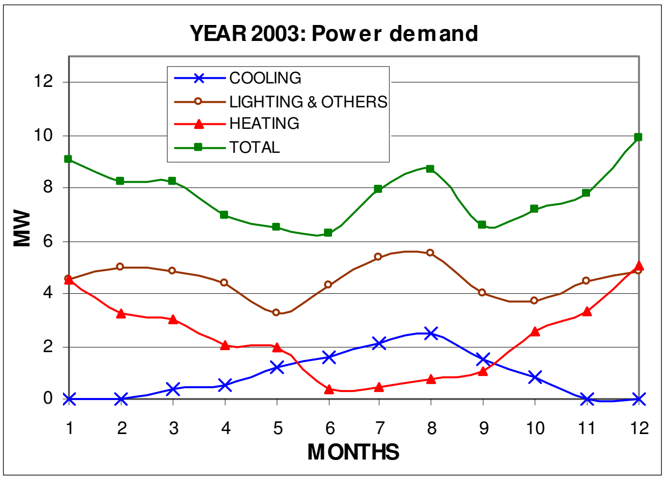

Fig.2.6: Lemnos power demand in MW, 2003, (typical day)

2.4 Study of energy needs of the island of Rhodes

2.4.1 General description of the island of Rhodes

Rhodes (or Rhodos) is an island in the south Aegean Sea, the largest of the Dodecanese group of islands (*Fig.2.7*). It is located at the south-eastern edge of the Aegean Sea, facing the shores of Turkey, which are 9-10 kilometres away. The population of the island exceeds 115,000 and it covers an area of 1,398km^2. Its landscape mainly comprises of hills and low mountains, which in their majority are covered with forests. Refreshing westerly winds moderate the summer heat, while the winter is nearly always mild, with long periods of sunshine.

Facts in brief: Country: Greece, Surface Area: 1,398km^2, Coastline: 220km, Capital city: Rhodes or Rhodos (population: ~60,000) (*Table B.10,* APPENDIX B.3).

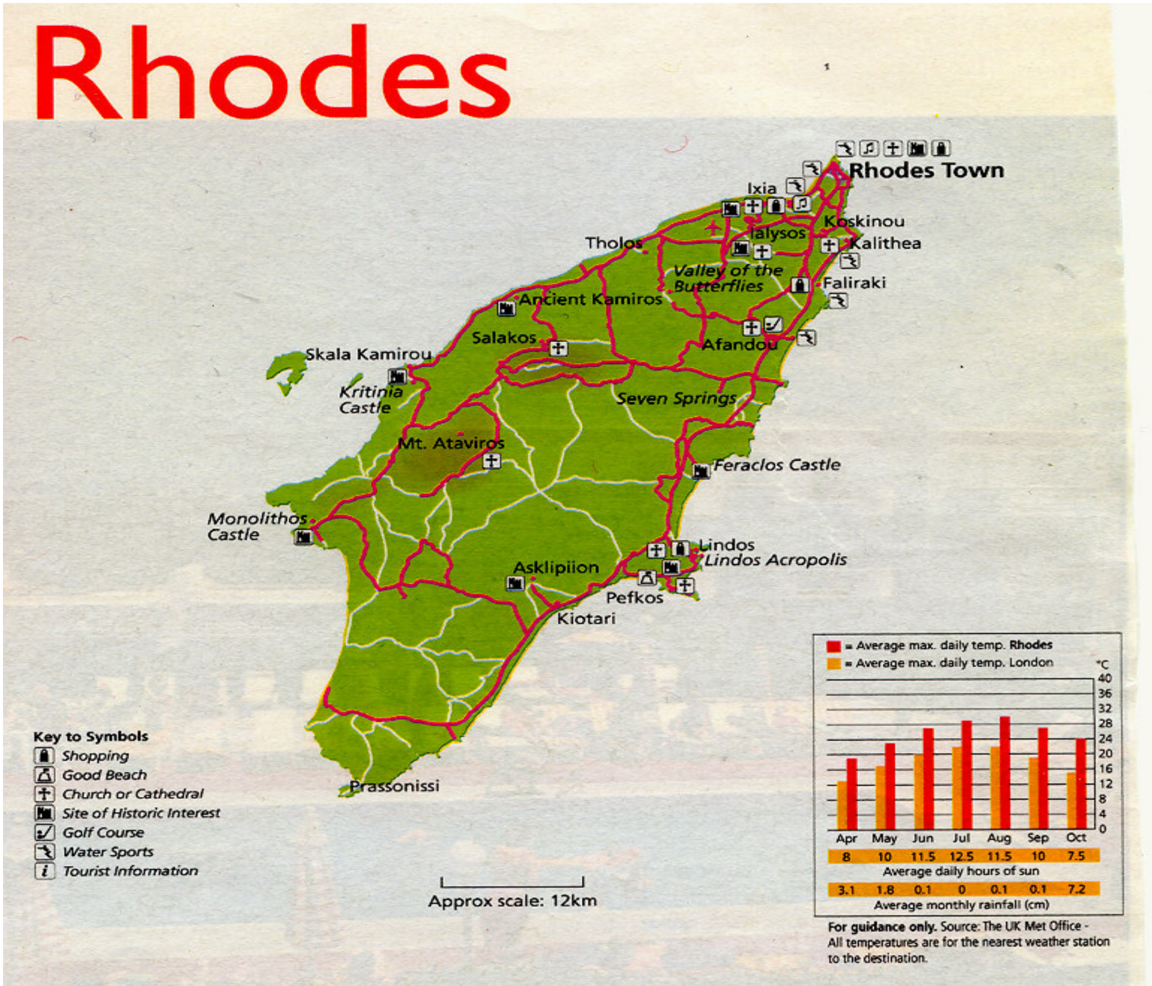

Fig.2.7: Rhodes Island

2.4.2 Climatic conditions of Rhodes

The climate is Mediterranean, that is to say, mild winter and hot summertime. Analytically, climatic statistical data for each month are reported in *Table 2.8:*

Table 2.8: Climatic data (Rhodes, East Longitude (Lon) 28°4'58" / North Latitude (Lat) 36°23'59")

MONTH	BAROMETRIC PRESSURE mmhg	AVERAGE AIR TEMPERATURE °C	ABSOLUTE MAXIMUM AIR TEMPERATURE °C	ABSOLUTE MINIMUM AIR TEMPERATURE °C	HOURS OF SUNLIGHT h	RELATIVE HUMIDITY %	AVERAGE CLOUDINESS (scale of 8)
1	1,015.7	11.9	22	-4	135.7	70.1	4.3
2	1,014.8	12.1	22	-2.2	142.0	69.1	4.2
3	1,013.4	13.6	27.4	0.2	206.0	68.7	3.9
4	1,012	16.6	30.6	5.2	246.7	66.5	3.5
5	1,011.7	20.5	34.8	5	314.5	64.4	2.9
6	1,009.8	24.7	37.4	12.6	355.5	58.5	1.1
7	1,006.9	26.9	40	14.6	387.1	57.6	0.3
8	1,007.5	27.1	42	17	373.3	59.9	0.3
9	1,011.4	24.6	36.6	10.6	313.6	61.4	0.8
10	1,014.7	20.8	33.2	7.2	239.6	67.5	2.4
11	1,016.4	16.5	28.4	2.4	184.4	71.4	3.5
12	1,015.8	13.4	22.4	1.2	142.1	72.4	4.2
AVER		**19.1**			**3,041**	**65.6**	

2.4.2 Description of the existing situation

The electrification of Dodecanese islands is based on autonomous petrol stations, while the geographic location of the islands has not allowed, up to now, their connection with the central national network of electric energy, or with the network of Turkey. (*Table B.11,* APPENDIX B.3).

PPC's electricity generation system is comprised of an interconnected system of production of the mainland (continental) country and the independent systems of production of Crete, Rhodes and other of smaller islands nearby. (*Table 2.9*)

Table 2.9: Installed power and net production of each independent system of production of PPC (1998)

ELECTRIC SYSTEM	MW	%	GWh	%
Interconnected	9,152	89.0	38,454	92.0
Crete	529	5.1	1,776	4.2
Rhodes	206	2.0	472	1.1
Rest of Islands	404	3.9	1,132	2.7
TOTAL	10,296	100.0	41,834	100.0

The limited installed power of units of production and networks of transport of electric energy of each island don't ensure sufficiency and stability in cases of peaks. This results in several problems in the network and, in certain cases, provisional interruption of electrification, mainly in the summertime period which is the peak tourists' season.

The energy demand in Rhodes presents intense fluctuation during the year, because of the change of population. Specifically, while the population of Rhodes is under 120,000 in terms of permanent residents, it increases considerably during certain periods throughout the year Apart from the summer tourist period, which is the main period of increased population, important changes of population also occur during two-days or three-days holidays throughout the year, at Christmas and at Easter. (*Table B.12,* APPENDIX B.3)

The autonomous stations electricity generation stations provide the electric charge during both periods of smooth change of demand and peak periods.

The procedure, which leads to the final heating, cooling, lighting consumption energy and the heating, cooling, lighting power needed, is presented in APPENDIX B.3.

Rhodes also is presents high development rates and it has been predicted that in the next five years the average rate of increase will be approximately 6%. It is noteworthy that the rate of demand for commercial use approaches the 50% mark and this reveals the direct dependence of the economic growth of the island on tourism. (*Table B.13,* APPENDIX B.3).

<u>Conclusions</u>
The energy consumption of the island varies throughout the year. This is due to variations in the population and the climate conditions.

The autonomous petrol stations of PPC and the network of transport of electric energy face frequent problems. Solving these problems requires the manufacturing of units which:

- produce large quantities of electric energy for the continuously increasing needs of the island,
- occupy small areas,
- are environmental friendly

The ideal solution to the energy problem seems to be the creation of modern units of trigeneration, in combination with the exploitation of R.S.E. the utilization of these methods could limit the energy problem of the island by providing a high degree of output, which is also environmentally friendly.

Table 2.10: Rhodes energy demand in MWh, 2003, (typical day)

YEAR: 2003	COOLING MWh_c	LIGHTING & OTHER MWh_e	HEATING MWh_t		TOTAL ENERGY
			ELEC.	BOILER	
JAN	0	1,072	396	397	1,865
FEB	31	1,294	234	234	1,793
MAR	117	1,201	147	146	1,611
APR	391	921	84	84	1,480
MAY	540	1,045	101	101	1,787
JUN	757	1,303	42	42	2,144
JUL	1,043	1,512	51	53	2,658
AUG	1,202	1,509	83	85	2,878
SEP	802	1,279	86	87	2,254
OCT	503	1,128	104	104	1,840
NOV	35	1,008	129	129	1,301
DEC	0	1,092	425	424	1,941

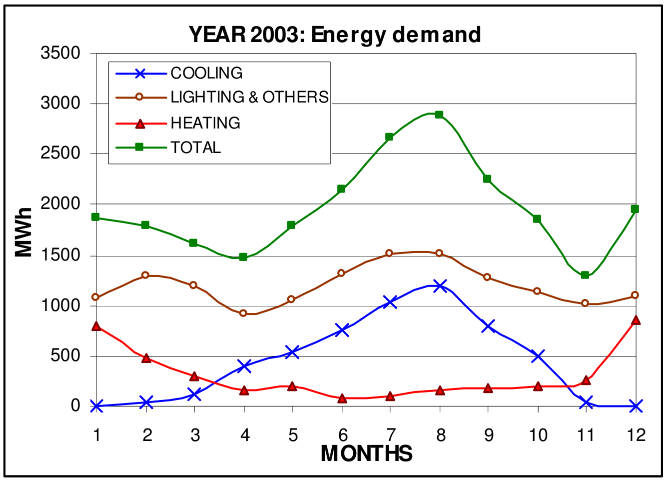

Fig.2.8: Rhodes energy demand in MWh, 2003, (typical day)

Table 2.11: Rhodes power demand in MW, 2003, (typical day)

YEAR: 2003	COOLING MW_e	LIGHTING & OTHER MW_e	HEATING MW_t		TOTAL POWER
			ELECTR.	BOILER	
JAN	0	44.670	16.520	16.520	77.710
FEB	1.300	53.910	9.740	9.750	74.700
MAR	4.880	50.050	6.100	6.110	67.130
APR	16.290	38.390	3.490	3.490	61.660
MAY	22.480	43.560	4.210	4.220	74.470
JUN	31.520	54.290	1.750	1.750	89.310
JUL	43.440	62.990	2.170	2.170	110.800
AUG	50.070	62.880	3.450	3.540	119.900
SEP	33.410	53.270	3.610	3.610	93.900
OCT	20.970	47.000	4.340	4.340	76.650
NOV	1.460	41.990	5.380	5.360	54.200
DEC	0	45.500	17.700	17.690	80.890

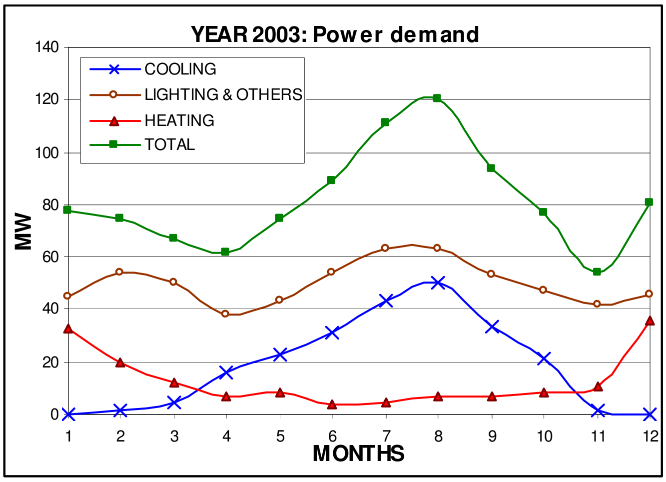

Fig.2.9: Rhodes power demand in MW, 2003, (typical day)

2.5 Study of energy needs of a hotel in Rethimno-Crete

2.5.1 The hotel sector in Greece from the energy point of view

Tourism is one of the most important parts of the Greek economy. Tourism provides the 9.6% of the Gross National Income, while it creates 320,000 jobs. The number and the capacity of hotel units, is growing continuously. The majority of the existing hotels are consuming relatively high amounts of energy.

The conclusions that follow below are based on a statistical study, which includes hotel energy consumption data, from different regions of Greece.

The hotel sector in Greece consumes about 14.271GWh per year in total. This amount of energy is relatively high and it could be reduced by using new energy production techniques. Particularly, the statistical analysis concludes that the electricity is the most consumed energy source, while petroleum and natural gas are lagging. The usage of renewable energy sources is still low.

The main (75% of the total energy consumption) energy activities of a hotel are:

- space heating
- cooling
- water heating
- cooking
- lighting

The study showed that the main reasons, which obstruct the application of new less energy consuming technologies, are:

- lack of information about the new technologies and their application to the hotel sector
- lack of information regarding economic incentives, provided by government or EU funds
- large investment per capita and delayed cost effectiveness, relative to the initial investment

In order to overcome these problems, the following actions must be taken:

- demonstration projects
- legislation
- promotion
- government subsidies
- classification of the hotels according to their energy consumption, and environment friendliness
- Usage of an Intergraded Energy Management System. The customer can choose the room temperature between 21-25^{o}C. During his absence and for a short period of time during night, the system automatically reduces the temperature it allows higher temperatures in the winter or lower ones in summer. It is worth mentioning that each Celsius degree corresponds to 6-10% conservation of energy.

2.5.2 General description of the hotel in Rethimno-Crete

The hotel begun its operation in 1991 and was expanded in 1995. It is three-star hotel and its capacity is 140 rooms or 280 beds. The operation period is 8 months, namely March to October. The hotel is located 2 km outside of Rethimno, which is in northern Crete (island in the south-east Mediterranean). (*Fig.2.10*)

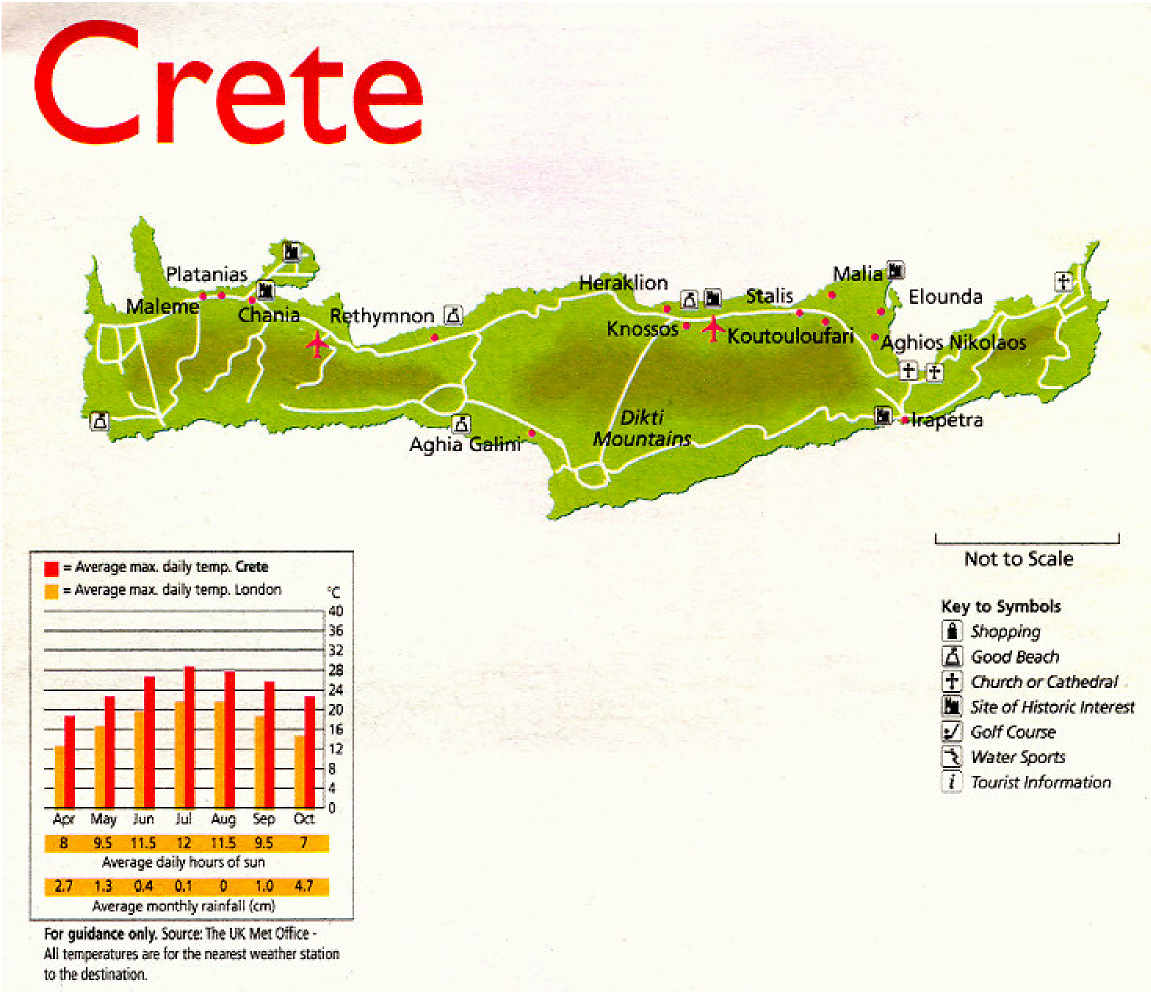

Fig.2.10: Crete Island

The hotel consists of the main building, and four independent smaller buildings. The main building consists of:

- a ground floor: lounge, cafeteria, restaurant, kitchen
- a basement: entertainment halls, conference room, electromechanical installation.
- First and second floor: 30 rooms

The four independent two-floor buildings consist of 49, 11, 41 and 9 rooms respectively (110 rooms totally). Total surface: 3,821m^2, from which 2,572m^2 correspond to rooms, 908m^2 correspond to public space, 150m^2 correspond to electromechanical installation and the remaining 191m^2 correspond to others.

Analytically, all the data, which assists in the calculation of the energy consumption - namely climatic data (*Table 2.12),* number of guests, power installation etc- are shown in APPENDIX B.4.

Table 2.12: Climatic data (Rethimno-Crete East Longitude (Lon) 24°31'1" / North Latitude (Lat) 35°21'0")

MONTH	BAROMETRIC PRESSURE mmhg	AVERAGE AIR TEMPERATURE °C	ABSOLUTE MAXIMUM AIR TEMPERATURE °C	ABSOLUTE MINIMUM AIR TEMPERATURE °C	HOURS OF SUNLIGHT h	RELATIVE HUMIDITY %	AVERAGE CLOUDINESS (scale of 8)
1	1015.7	12.9	24.9	0.8	110.3	69	5.6
2	1014.8	13.1	25.4	2	132.2	67	5.3
3	1013.4	14.4	28.5	3	157	65	4.6
4	1012	19	33.2	5.4	218	64	3.8
5	1011.7	20.7	37	9.8	300	64	2.7
6	1009.8	24.9	37.5	13.6	335	61	1.6
7	1006.9	26.8	41.4	15	373.1	60	0.9
8	1007.5	26.9	39.3	16.4	350.2	61	1
9	1011.4	24.3	38	13.6	263.7	64	2.4
10	1014.7	20.8	35	8.8	166.1	67	4.1
11	1016.4	17.9	30.5	6.8	165.8	68	4.3
12	1015.8	14.8	28	2.4	112.9	67	5.3
		19.7			**223.69**	**64.75**	

Table 2.13: Hotel in Rethimno energy demand in kWh, 2002, (typical day)

YEAR: *2002*	COOLING kWh_c	ELECTRICITY kWh_e	HEATING kWh_t	TOTAL ENERGY
JAN	0	2,893	0	2,893
FEB	0	2,700	0	2,700
MAR	1,375	9,200	11,139	21,714
APR	8,320	20,369	27,083	55,772
MAY	12,844	21,870	30,516	65,230
JUN	23,850	23,850	22,278	69,978
JUL	28,533	26,338	17,729	72,600
AUG	31,161	29,939	20,475	81,575
SEP	26,990	31,683	20,063	78,736
OCT	9,322	25,203	13,198	47,723
NOV	0	12,302	0	12,302
DEC	0	1,953	0	1,953

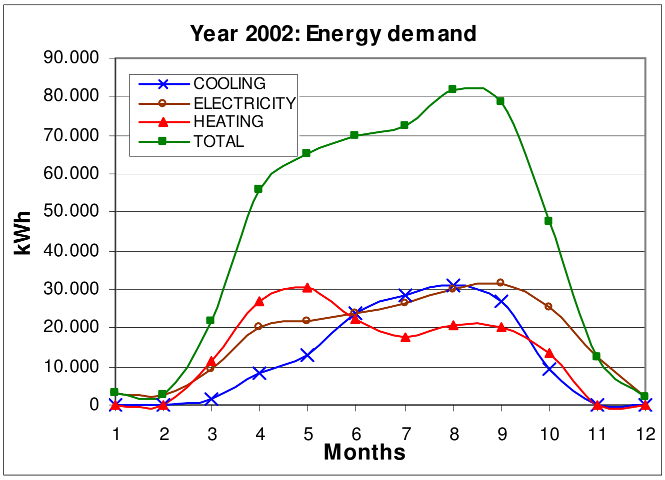

Fig.2.11: Hotel in Rethimno energy demand in kWh, 2002, (typical day)

Table 2.14: Hotel in Rethimno power demand in kW, 2002, (typical day)

YEAR: *2002*	COOLIN G kW_c	ELECTRICIT Y kW_e	HEATING kW_t	TOTAL POWER
JAN	0	4.0	0	4.02
FEB	0	3.8	0	3.75
MAR	1.9	12.8	15.5	30.16
APR	11.6	28.3	37.6	77.46
MAY	17.8	30.4	42.4	90.60
JUN	33.1	33.1	30.9	97.19
JUL	39.6	36.6	24.6	100.83
AUG	43.3	41.6	28.4	113.30
SEP	37.5	44	27.9	109.36
OCT	12.9	35	18.3	66.28
NOV	0	17.1	0	17.09
DEC	0	2.7	0	2.71

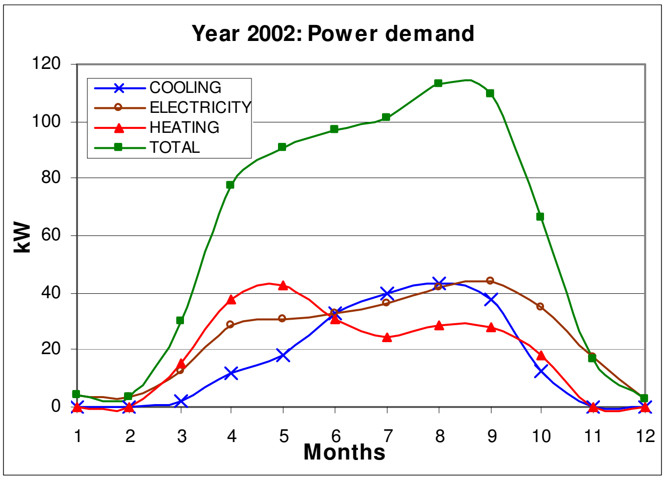

Fig.2.12: Hotel in Rethimno power demand in kW, 2002, (typical day)

2.6 Energy scene of the SANI BEACH HOTEL GROUP

2.6.1 General description of the SANI BEACH HOTEL GROUP

The SANI BEACH HOTEL group consists of three different hotel complexes:

- Sani Beach Hotel
- Porto Sani
- Club Hotel

Geographical characteristics of the SANI BEACH HOTEL GROUP region (*Fig.2.13*): Geographic location: northern Greece (Kassandra-Halkidiki, 85km south-east of Thessaloniki). The climatic data are the same as shown in the case of Thessaloniki (*Paragraph 2.2.2*)

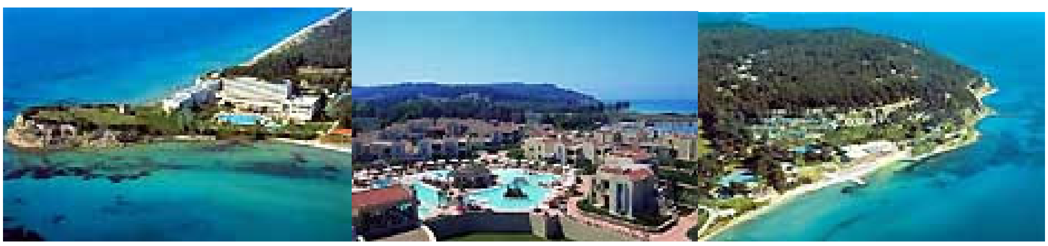

Fig.2.13: The Sani Beach Hotel Group

2.6.2 Sani Beach Hotel

The operation of the *Sani Beach Hotel* began at 1980. It is five-star hotel and it is constituted by a central building group of 2 buildings, of total surface 4,139m^2. Main specifications: 500 rooms, 1000 beds, bar, pools, restaurants, health center, conference room, tennis and basket courts.

The operation of the hotel is seasonal. Namely, it begins its operation in April, and closes in October each year, although, there is the possibility, the operation period to be extended proportionally to the tourist demands.

The data where the study is based are shown in APPENDIX B.5.

The *Sani Beach Hotel* energy results for the year 2001 are shown below (*Table 2.15, 2.16, Figs. 2.14, 2.15*):

Table 2.15: Sani Beach Hotel energy demand in kWh, 2001, (typical day)

YEAR: *2001*	**COOLING** kWh$_c$	**LIGHTING & OTHER** kWh$_e$	**HEATING** kWh$_t$		**TOTAL ENERGY**
			ELECTR.	BOILER	
JAN	240	1,652	13	0	1,905
FEB	240	1,333	13	0	1,586
MAR	240	1,073	12	1	1,327
APR	1,002	3,068	1,824	55	5,950
MAY	3,371	5,853	2,402	72	11,698
JUN	4,809	6,316	2,291	0	13,416
JUL	6,027	8,704	2,513	0	17,243
AUG	6,385	10,012	2,988	0	19,385
SEP	5,832	9,407	2,306	70	17,615
OCT	4,406	7,389	1,753	35	13,583
NOV	240	4,800	13	0	5,053
DEC	240	1,123	13	0	1,376

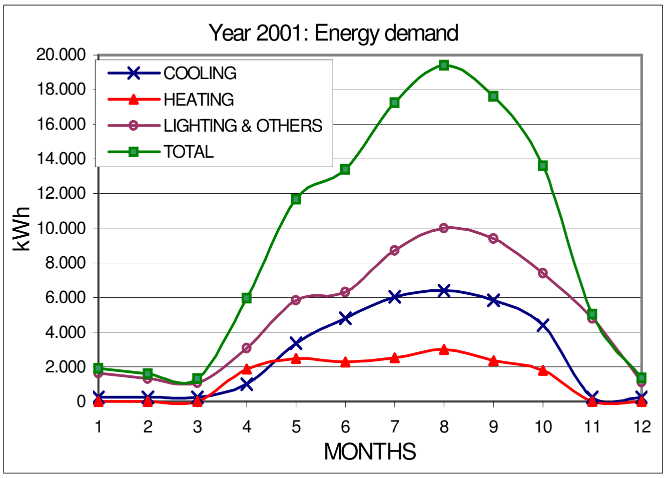

Fig.2.14: Sani Beach Hotel energy demand in kWh, 2001, (typical day)

Table 2.16: Sani Beach Hotel power demand in kW, 2001, (typical day)

YEAR: *2001*	**COOLING** kW$_c$	**LIGHTING & OTHER** kW$_e$	**HEATING** kW$_t$		**TOTAL POWER**
			ELECTR.	BOILER	
JAN	20	82.45	0.05	1.6	104.1
FEB	20	65.19	0.04	1.6	86.84
MAR	20	51.13	0.03	1.6	72.78
APR	78.9	140	4.50	152.1	375.5
MAY	200.7	296.9	5.90	200.3	703.7
JUN	244.9	357.7	0.00	190.9	793.5
JUL	289.3	508.6	0.00	209.4	1,007
AUG	328.1	560.1	0.00	249.0	1,137
SEP	273	550.4	5.80	192.2	1,021
OCT	214.4	423.5	2.90	146.1	786.9
NOV	20	253	0.00	1.6	274.6
DEC	20	53.8	0.04	1.6	75.46

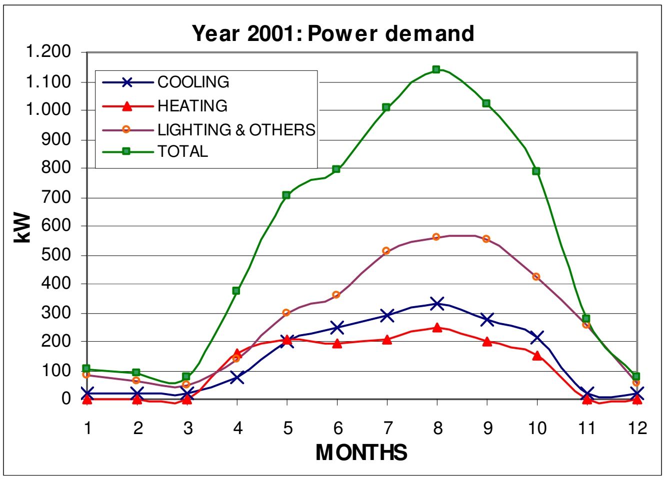

Fig.2.15: Sani Beach Hotel power demand in kW, 2001, (typical day)

2.6.3 Porto Sani Hotel

The operation of the *Porto Sani Hotel* began at May 1997. It is five-star hotel and it is constituted by a central building group of 2 buildings, of total surface 4,355m^2.

Main specifications: 103 rooms, 299 beds, bar, four pools, two restaurants, roof garden, conference room, tennis and basket courts.

The operation of the hotel is seasonal. Namely, it begins its operation in April, and closes in October each year.

The *Porto Sani Hotel* energy results for the year 2001 are shown below (*Table 2.17, 2.18, Figs. 2.16, 2.17*):

Table 2.17: Porto Sani Hotel energy demand in kWh, 2001, (typical day)

YEAR: *2001*	COOLING KWh$_c$	LIGHTING & OTHER kWh$_e$	HEATING kWh$_t$		TOTAL ENERGY
			ELEC..	BOIL.	
JAN	148.8	1,024	0.2	10.6	1,184
FEB	166.8	926	0.2	10.5	1,104
MAR	336.0	1,504	0.0	10.8	1,850
APR	570	2,718	45.0	1,473	4,807
MAY	1,692	2,916	59.0	1,939	6,606
JUN	2,433	2,820	0.0	1,856	7,109
JUL	3,088	4,459	0.0	2,035	9,582
AUG	3,666	5,748	0.0	2,420	11,833
SEP	2,844	5,313	56.0	1,863	10,076
OCT	870	4,702	28.0	1,418	7,017
NOV	153.7	3,073	0.3	10.3	3,237
DEC	211.3	988	0.7	10.1	1,210

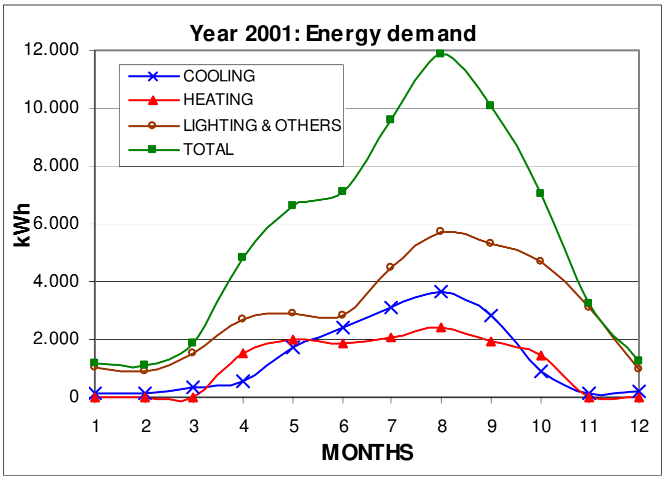

Fig.2.16: Porto Sani Hotel energy demand in kWh, 2001, (typical day)

Table 2.18: Porto Sani Hotel power demand in kW, 2001, (typical day)

YEAR: *2001*	COOLING kW$_e$	LIGHTING & OTHER kW$_e$	HEATING kW$_t$		TOTAL POWER
			ELEC.	BOIL.	
JAN	12.4	51.1	0.1	0.5	64.1
FEB	13.9	45.3	0.0	0.6	59.8
MAR	28.0	71.6	0.1	0.5	100.2
APR	39.6	135.9	5.1	62.9	243.9
MAY	80.6	165.6	6.6	83.4	336.1
JUN	112.7	171.9	0.0	84.0	368.1
JUL	139.8	268.9	0.1	113.9	523.0
AUG	160.8	349.1	0.0	136.0	646.0
SEP	133.2	305.4	6.3	101.7	547.0
OCT	72.5	227.6	3.2	77.8	381.4
NOV	12.8	162.0	0.0	0.6	175.4
DEC	17.6	47.4	0.0	0.6	65.6

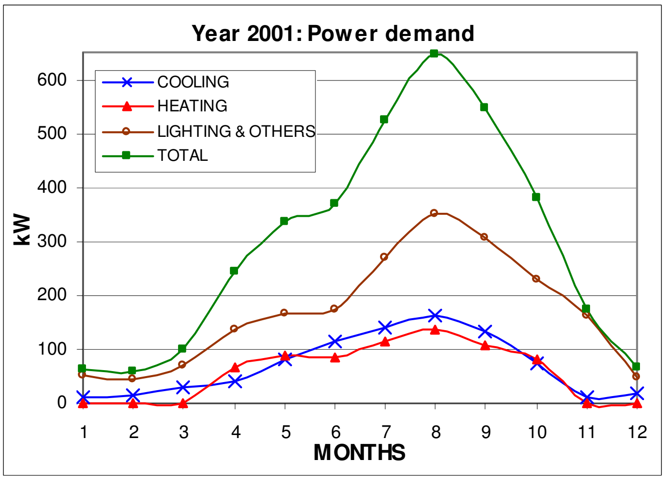

Fig.2.17: Porto Sani Hotel power demand in kW, 2001, (typical day)

2.6.4 Club Hotel

The operation of the *Club Hotel* began at 1978. It is five-star hotel and it is constituted by a central building, of total surface 2,580m^2.

Main specifications: 215 rooms, 425 beds, bar, pools, restaurants, health center, conference room, tennis and basket courts.

The operation of the hotel is seasonal. Namely, it begins its operation in April, and closes in October each year.

The *Club Hotel* energy results for the year 2001 are shown below (*Table 2.19, 2.18, Figs.2.18, 2.19*):

Table 2.19: Club Hotel energy demand in kWh, 2001, (typical day)

YEAR: *2001*	**COOLING** KWh$_c$	**LIGHTING & OTHER** kWh$_e$	**HEATING** kWh$_t$ ELECTR.	**HEATING** kWh$_t$ BOILER	**TOTAL ENERGY**
JAN	64.3	442	0.7	4.5	512
FEB	93.6	520	0.4	4.8	619
MAR	102.3	458	0.7	4.5	565
APR	75	365	0.0	5.0	445
MAY	455	787	11.0	1,861	3,114
JUN	2,653	2441	0.0	1,816	6,910
JUL	2,891	4175	1.0	1,904	8,972
AUG	3,250	5096	1.0	2,094	10,442
SEP	2,521	4748	11.0	1,823	9,103
OCT	428	2869	3.0	875	4,175
NOV	34.3	686	0.7	4.5	725
DEC	70.4	330	0.6	4.6	405

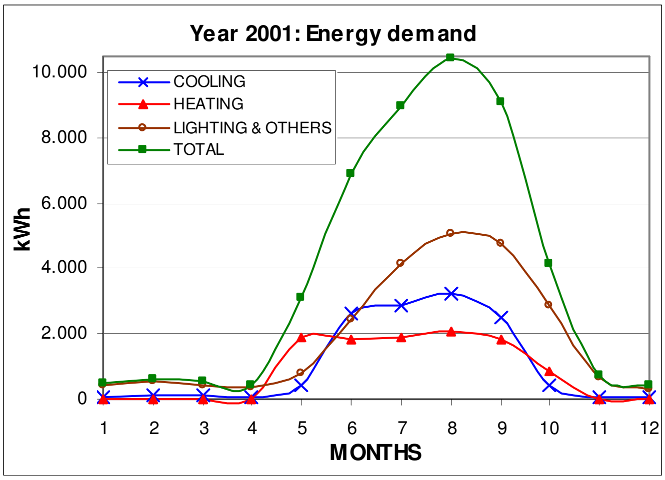

Fig.2.18: Club Hotel energy demand in kWh, 2001, (typical day)

Table 2.20: Club Hotel power demand in kW, 2001, (typical day)

YEAR: *2001*	**COOLING** KW$_c$	**LIGHTING & OTHER** kW$_e$	**HEATING** kW$_t$ ELECTR.	**HEATING** kW$_t$ BOILER	**TOTAL ENERGY**
JAN	5.4	22.1	0.0	0.9	28.3
FEB	7.8	25.4	0.0	0.9	34.1
MAR	8.5	21.8	0.0	0.9	31.2
APR	11.6	12.2	0.0	0.9	24.7
MAY	27.1	38.9	1.9	63.1	131.0
JUN	122.8	153.1	0.0	56.0	331.5
JUL	130.9	251.8	0.1	70.9	453
AUG	142.6	309.6	0.0	103.0	555
SEP	118.0	274.4	1.9	57.1	451
OCT	20.7	157.5	0.6	42.4	220.8
NOV	2.9	36.1	0.0	0.9	39.9
DEC	5.9	15.8	0.0	0.9	22.5

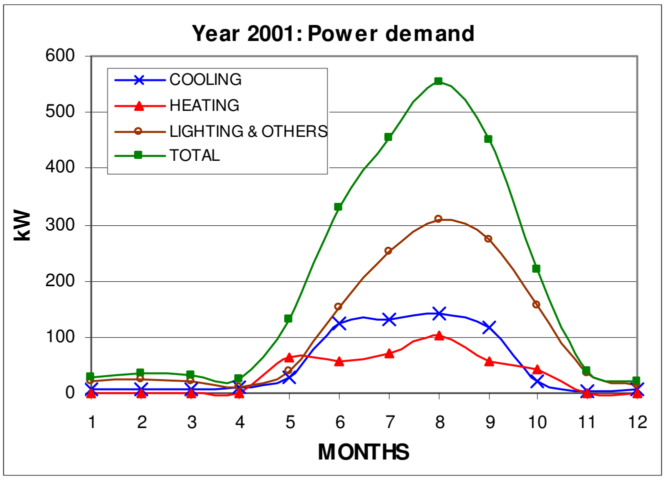

Fig.2.19: Club Hotel power demand in kW, 2001, (typical day)

3. GAS TURBINE PERFORMANCE

3.1 Introduction

Cycle analysis studies the thermodynamic changes of the working fluid (air and products of combustion in most cases) as it flows through the engine. It is divided into two types of analysis:

1. **Parametric cycle analysis** (also called **design-point, DP**) and
2. **Engine performance analysis** (also called **off-design**, **OD**).

Parametric cycle analysis determines the performance of engines in different ambient conditions and values of design choice (e.g., compressor pressure ratio, R_C) and design limit (e.g., Turbine Entry Temperature, TET) parameters. Engine performance analysis determines the performance of a specific engine in different ambient conditions, possible degradation and throttle settings.

Design point performance is central to the engine concept design process. The engine configuration, cycle parameters, component performance levels and sizes are selected to meet a given specification. Design point performance must be defined before analysis of any other operating conditions is possible. The resulting overall performance of the final engine will be crucial to its commercial success or failure.

The aim of this chapter is to produce two computer programs, which simulate the DP and OD performance of four types of Gas Turbines. These are:

1. 1-shaft simple cycle
2. 2-shaft simple cycle
3. 1-shaft with heat exchanger cycle
4. 2-shaft with heat exchanger cycle

These types of engines are the most suitable (as will be explained in the following paragraphs), but the programs can easily be used for the simulation of all other types of GTs, with the application of small modifications. The program is constructed using FORTRAN as a programming language. The development of these programs has been done, with the aim of making them both as accurate as possible and flexible enough to cooperate with simulation programs of cooling performance and economic evaluation, which will follow in the next chapters.

The results are presented mainly by means of diagrams and were checked through the TURBOMATCH [21] simulation program provided by Cranfield University.

3.2 Calculation Procedure of Design Point Performance

Initially, the operating conditions under which an engine will spend the most time has been traditionally chosen as the engine's design point. For an industrial unit this would normally be the ISO base load, or for an aero-engine cruise at any altitude on an 1SA day. Either way, at the design point, the engine configuration, component performance and cycle parameters are optimised. The method used is the design point performance calculation. Each time input parameters are changed and this calculation procedure is repeated, **the resulting change to the engine design requires different engine geometry, at the fixed operating condition**.

The main objective of parametric cycle analysis is to relate the engine performance parameters (primarily specific power SW, and specific fuel consumption sfc) to design choices (R_C, etc.), to design limitations (TET, compressor exit pressure, component efficiencies, etc.), and to environmental conditions (ambient pressure and temperature, etc.). From parametric cycle analysis, we can easily determine which engine type (e.g., 1 or 2-shaft), engine characteristics (TET, R_C) and component design characteristics best satisfy a particular need.

The value of parametric cycle analysis depends directly on the realism with which the engine components are characterized.

Generally, the way of determining the design point analysis is as follows:

Initially the **engine layout** is specified. This is done by using a block diagram, where every component of the engine is represented. The inlet and outlet of every component is characterized by a sequent number (**station vector**). For example, intake 1-2, compressor 2-3 etc. Thermodynamic equations for every part of the engine are applied, ensuring the continuity of the values throughout the engine. The thermodynamic status (namely P_o, T_o, $\dot{m}$) for every station must be calculated.

When the above values are known then the **performance analysis** can be determined. This includes the calculation of the following:

Compressor Work CW in MW
Turbine Work TW in MW
Useful Work UW in MW
Heat Input HI in MW
Fuel flow $\dot{m}_f$ in kgr/s
Specific Fuel Consumption sfc in kgr/(MW·s) = kgr/MJ
Thermal efficiency η_{GT}

The most effective method for presenting the performance characteristics of the engine is to plot the variation of specific fuel consumption and specific work on a single figure for a range of values of pressure ratio and turbine entry temperature. **Each point on such plots represents a different engine cycle**. These plots are very important because they:

- Provide an indication of the optimum combination of cycle parameters for a given engine type.
- Compare the performance of different engine configurations which may be considered for a given engine requirement.

To carry out the calculation procedures using the design point computer programs the following parameters must be known or defined:

- Ambient conditions (T_α, P_α,)
- Air mass flow ($\dot{m}$)
- Component efficiencies (η_{isc}, η_{in}, η_b, η_{exh})
- Component pressure losses (ΔP_{loss})
- Specific Heat C_p throughout the engine (depending on the chemical composition of the working fluid and to the temperature)
- Cooling air percentage
- Fuel calorific value (FCV)
- Turbine entry temperature TET (depending on the thermal durability of the inlet blades of the first turbine row)
- Exhaust pressure

3.3 Design Point Performance Simulation
3.3.1 1-Shaft Simple Cycle Simulation Procedure

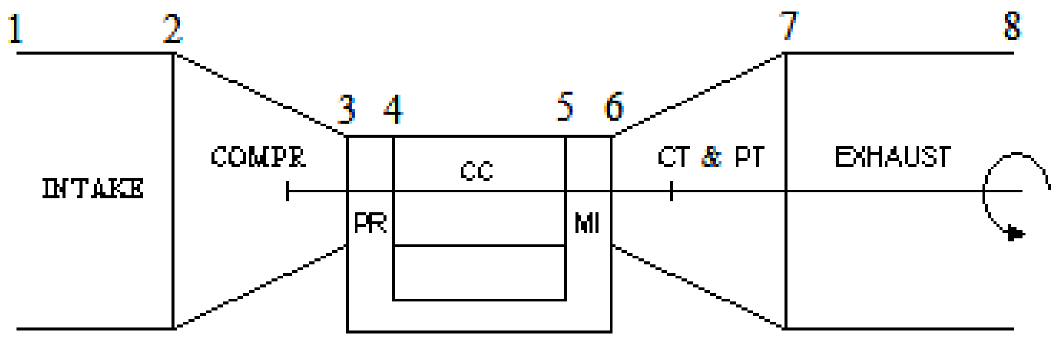

Fig. 3.1: 1-Shaft simple cycle engine layout

<u>Input File (APPENDIX C.1)</u>

Ambient inputs

ISO Atmosphere: (Ambient pressure in kPa, Ambient temperature in K: $P\alpha$=101.3, $T\alpha$=288.

Working fluid inputs

The working fuel is air, with mass flow: $\dot{m}$ (kg/s)

Specific heat, Cp kJ/(kgr.K)) assumptions:

- The working fluid throughout the engine is assumed to be a perfect gas. In reality the fuel is less than 2% of the working fluid mass.(Fuel/Air Ratio, FAR very small)
- For the "cold" section of the engine Cp_c=1,005, γ_c=1.4 (representing cold air) and for the "hot" Cp_h=1,150, γ_h=1.333 (representing hot gas)

Intake inputs

In order to observe the design point performance independently from the size of the engine, is assumed that $\dot{m}=1$ (kgr/s),

The inlet mach number is M_{in}=0, the engine is stationary

Intake surroundings are assumed to be adiabatic, so intake pressure loss and intake temperature loss assumed to be zero. $DT_{inloss}=DP_{inloss}=0$

Compressor inputs

Compressor pressure ratio R_C, **varies** between 5÷30.

Compressor isentropic efficiency, η_{isc} is assumed to be between, 0.77÷0.91 (η_{isc}=0.85). The value of the compressor isentropic efficiency decreases when R_C arising.

Actually, compressor "work" is power and is represented as CW in MW.

Compressor degradation, P_{Cde} (%)=0

Premass inputs

Cooling air percentage Dm_C (%), is a function of TET. The cooling is necessary only if TET>1,300K.

Combustion Chamber inputs

Combustion efficiency, η_{cc}=0.985÷0.995 (η_{cc}=0.99)

Combustor pressure loss, DP_{ccloss} = 3÷5% (DP_{ccloss} =5%)

Fuel calorific value, FCV in MJ/kg

Mixer inputs

Temperature reduction due to cooling, $DT_{cooling}$, is a function of TET

Turbine inputs

Turbine entry temperature TET **varies** between 800 ÷1,700(K). The lower value is restricted by the compressor pressure ratio R_C, while the upper by the thermal durability of the first row blades.
Turbine isentropic efficiency η_{ist}, is assumed to be between 0.83÷0.97 (η_{ist}=0.9). The value of the turbine isentropic efficiency increases when R_t arising.
Turbine degradation, P_{tde} (%)=0
Actually, turbine "work" is power and is represented as TW in MW.
Exhaust inputs
Exhaust surroundings are assumed to be adiabatic, so exhaust pressure loss and exhaust temperature loss assumed to be zero, $DT_{exloss}=DP_{exloss}=0$
It is assumed to be a minimum pressure difference between the exhaust and ambient pressure DPex=2÷3% (or $P_{o,ex}=1.003P_\alpha$ which gives an exit velocity of the gases approximately 80÷100km/s.
Attention: All the units are in International Standard System, IS.

<u>Calculation procedure</u>

1-2 INTAKE

$$P_{o1} = P\alpha \cdot (1 + \frac{\gamma_c - 1}{2} \cdot M_{in}^2)^{\frac{\gamma_c}{\gamma_c - 1}} \quad (3\text{-}1)$$

$$P_{o2} = P_{o1} \cdot (1 - \frac{DPinloss}{100}) \quad (3\text{-}2)$$

$$T_{o1} = T\alpha \cdot (1 + \frac{\gamma_c - 1}{2} \cdot M_{in}^2) \quad (3\text{-}3)$$

$$T_{o2} = T_{o1} \quad (3\text{-}4)$$

$$\dot{m}_1 = \dot{m}_2 = \dot{m} \quad (3\text{-}5)$$

2-3 COMPRESSOR

$$P_{o3} = P_{o2} \cdot R_c \quad (3\text{-}6)$$

$$T_{o3} = (T_{o2} \cdot \frac{R_c^{\gamma_c - 1/\gamma_c} - 1}{\eta_{isc}} + T_{o2}) \quad (3\text{-}7)$$

$$\dot{m}_3 = \dot{m} \quad (3\text{-}8)$$

3-4 PREMASS

$$P_{o4}=P_{o3} \quad (3\text{-}9)$$

$$T_{o4}=T_{o3} \quad (3\text{-}10)$$

5-6 MIXER

If TET = T_{o5} is less than 1,300 then it is assumed that there is no need for by pass mass flow to cool the turbine section, thus

$$D_{mc}=0,\ DT_{cooling}=0 \quad (3\text{-}11)$$

On the contrary, if T_{o5} greater or equal to 1,300 then there is a portion of mass flow (D_{mc}), which reduces the TET per $DT_{cooling}$:

$$D_{mc}=0.025*T_{o5}-25,\ DT_{cooling}=0.333*T_{o5}-333.333 \quad (3\text{-}12)$$

3-4 PREMASS

$$\dot{m}_4 = \dot{m} \cdot \left(1 - \frac{Dmc}{100}\right) \quad (3\text{-}13)$$

4-5 COMBUSTION CHAMBER (BURNER)

$$P_{o5} = P_{o4} \cdot \left(1 - \frac{DPccloss}{100}\right) \quad (3\text{-}14)$$

$$T_{o5}=TET \quad (3\text{-}15)$$

$$\dot{m}_f = \frac{\dot{m}_4 \cdot (C_{ph} \cdot T_{o5} - C_{pc} \cdot T_{o4})}{\eta_{cc} \cdot FCV \cdot 1.000.000)} \quad (3\text{-}16)$$

$$\dot{m}_5 = \dot{m}_4 + \dot{m}_f \quad (3\text{-}17)$$

$$FAR_{45}= \dot{m}_f / \dot{m}_4 \quad (3\text{-}18)$$

5-6 MIXER

$$P_{o6} = P_{o5} \quad (3\text{-}19)$$

$$T_{o6}=T_{o5}\text{-}DTcooling \quad (3\text{-}20)$$

$$\dot{m}_6 = \dot{m} + \dot{m}_f \quad (3\text{-}21)$$

7-8 EXHAUST

$$P_{o8} = P_\alpha \cdot 1{,}003 \quad (3\text{-}22)$$

$$\dot{m}_8 = \dot{m}_6 \quad (3\text{-}23)$$

6-7 COMPRESSOR & POWER TURBINE

$$P_{o7} = \frac{P_{o8} \cdot 100}{100 \text{ - } DPexhloss} \quad (3\text{-}24)$$

$$T_{o7} = T_{o6} \cdot \left[1 - \left[1 - (\frac{P_{o7}}{P_{o6}})^{(\gamma_h - 1)/\gamma_h}\right] \cdot n_{ist}\right] \quad (3\text{-}25)$$

$$\dot{m}_7 = \dot{m}_6 \quad (3\text{-}26)$$

7-8 EXHAUST

$$T_{o8} = T_{o7} \cdot (1 - \frac{DTexhloss}{100}) \quad (3\text{-}27)$$

PERFORMANCE

$$CW=\frac{\dot{m}_2 \cdot C_{pc} \cdot (T_{o3} - T_{o2})}{1.000.000} \quad (MW) \quad (3\text{-}28)$$

$$TW=\frac{\dot{m}_6 \cdot C_{ph} \cdot (T_{o6} - T_{o7})}{1.000.000} \quad (MW) \quad (3\text{-}29)$$

$$HI= \dot{m}_f \cdot FCV \quad (MW) \quad (3\text{-}30)$$

$$UW=TW\text{-}CW \quad (MW) \quad (3\text{-}31)$$

$$SW=UW/\dot{m} \quad (MJ/kgr) \quad (3\text{-}32)$$

$$\eta_{th}=UW/HI \quad (3\text{-}33)$$

$$sfc=1/(\eta_{th}*FCV) \quad (kgr/(MW*sec)=kgr/MJ) \quad (3\text{-}34)$$

The results of the above calculation procedure, when varying TET and R_C -using the FORTRAN program (APPENDIX C.2) developed by the author- are represented in *Table 3.1* and *Fig. 3.2*.

Table 3.1: 1-Shaft GT and 2-Shaft GT Design point performance

		1-Shaft GT			2-Shaft GT		
TET (K)	Rc	SW [(MW*sec)/kgr]	Sfc [kgr/(MW*sec)]	η_{th}	SW [(MW*sec)/kgr]	Sfc [kgr/(MW*sec)]	η_{th}
900	5	0.1027	0.1105	0.1862	0.1011	0.1122	0.1834
	10	0.0853	0.1042	0.1975	0.0845	0.1052	0.1956
	15	0.0556	0.1298	0.1585	0.0541	0.1333	0.1543
	20	-	-	-	-	-	-
	25	-	-	-	-	-	-
	30	-	-	-	-	-	-
1,000	5	0.1369	0.1003	0.2051	0.1352	0.1016	0.2025
	10	0.1311	0.0861	0.2391	0.1308	0.0863	0.2385
	15	0.1071	0.0897	0.2295	0.1073	0.0895	0.2299
	20	0.0806	0.1028	0.2001	0.0803	0.1032	0.1993
	25	0.0544	0.1322	0.1557	0.0528	0.1363	0.1510
	30	-	-	-	-	-	-
1,100	5	0.1714	0.0941	0.2187	0.1692	0.0953	0.2160
	10	0.1770	0.0772	0.2665	0.1768	0.0773	0.2661
	15	0.1589	0.0755	0.2726	0.1599	0.075	0.2744
	20	0.1361	0.0784	0.2623	0.1377	0.0776	0.2653
	25	0.1127	0.085	0.2420	0.114	0.084	0.2449
	30	0.0897	0.0963	0.2138	0.0903	0.0956	0.2151
1,200	5	0.2060	0.0899	0.2290	0.2034	0.091	0.2260
	10	0.2232	0.0719	0.2860	0.2228	0.0721	0.2855
	15	0.2109	0.0682	0.3018	0.2123	0.0677	0.3037
	20	0.1919	0.0681	0.3023	0.1944	0.0672	0.3062
	25	0.1712	0.0699	0.2944	0.1743	0.0687	0.2997
	30	0.1503	0.0733	0.2807	0.1536	0.0718	0.2867
1,300	5	0.2407	0.0868	0.2370	0.2376	0.088	0.2339
	10	0.2696	0.0684	0.3007	0.2688	0.0686	0.2999
	15	0.2632	0.0637	0.3229	0.2645	0.0634	0.3246
	20	0.2480	0.0623	0.3302	0.251	0.0616	0.3341
	25	0.2300	0.0624	0.3297	0.2341	0.0613	0.3355
	30	0.2113	0.0635	0.3242	0.2161	0.0621	0.3316
1,400	5	0.2756	0.0845	0.2435	0.2719	0.0856	0.2403
	10	0.3161	0.0659	0.3123	0.3149	0.0661	0.3111
	15	0.3156	0.0607	0.3391	0.3168	0.0605	0.3403
	20	0.3043	0.0586	0.3509	0.3074	0.058	0.3545
	25	0.2891	0.0579	0.3552	0.2937	0.057	0.3608
	30	0.2725	0.058	0.3549	0.2782	0.0568	0.3623
1,500	5	0.3107	0.0826	0.2490	0.3064	0.0838	0.2456
	10	0.3629	0.064	0.3216	0.3612	0.0643	0.3200
	15	0.3684	0.0585	0.3518	0.3691	0.0584	0.3526
	20	0.3609	0.0561	0.3671	0.3637	0.0556	0.3700
	25	0.3485	0.0549	0.3747	0.3531	0.0542	0.3797
	30	0.3340	0.0544	0.3779	0.3401	0.0535	0.3848

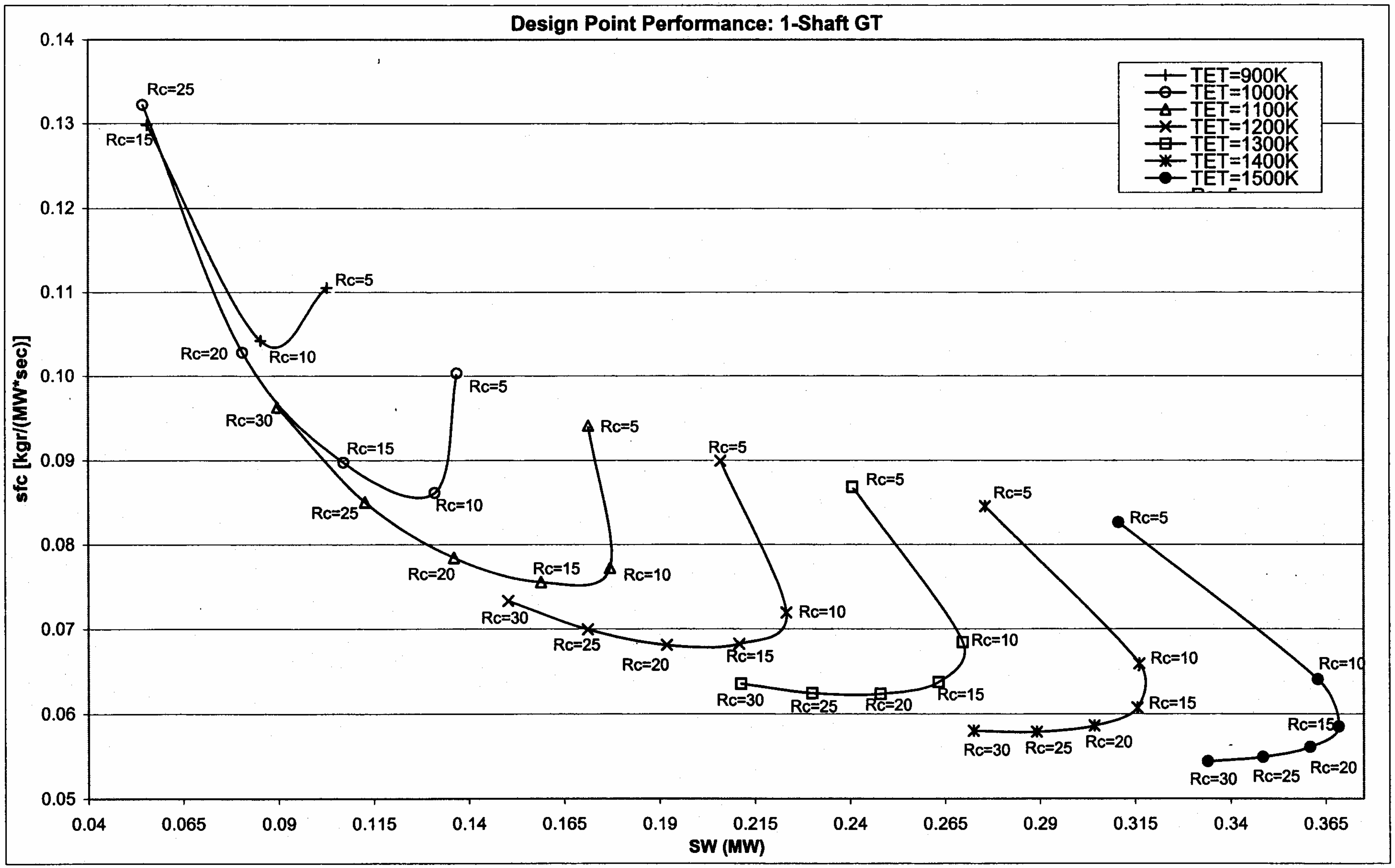

Fig. 3.2:1-Shaft GT Design Point performance

3.3.2 2-Shaft Simple Cycle Simulation Procedure

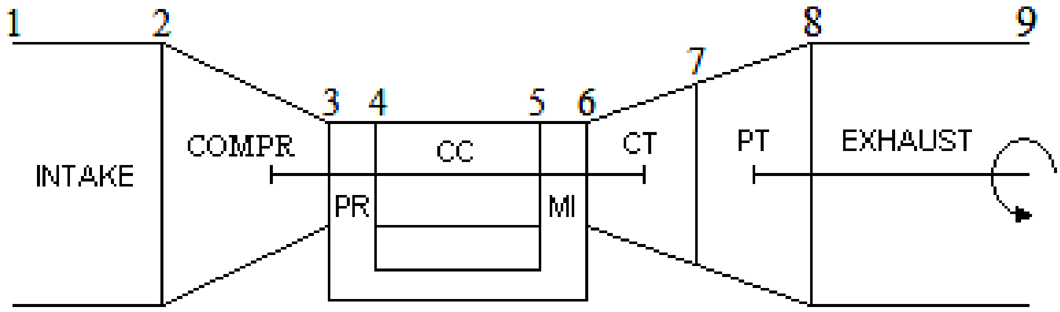

Fig. 3.3: 2-Shaft simple cycle engine layout

Input File (APPENDIX C.1)

It is the same with those in paragraph 3.3.1 with two exceptions:

- The isentropic efficiency of the compressor turbine (n_{isct}=0.89), should be lower than the value assumed for the turbine of the simple cycle, due to the lower compressor turbine pressure ratio. Accordingly the polytropic efficiency of the compressor turbine is assumed to be lower than isentropic, n_{polct}=0.86.
- Similarly, the isentropic efficiency of the power turbine is considered to be n_{ispt}=0.88.

Notice:

If the isentropic efficiencies of the two types of turbines assumed to be equal to the one of the 1-Shaft GT, then the efficiency of the 2-Shaft GT would be found slightly higher. (1-Shaft GT: $R_C \approx R_T$, 2-Shaft GT: $R_C \approx R_{CT} \cdot R_{PT}$).

Calculation procedure

1-2 INTAKE

Equations (3-1) έως (3-5)

2-3 COMPRESSOR

Equations (3-6) έως (3-8)

$$CW = \frac{\dot{m}_2 \cdot C_{pc} \cdot (T_{o3} - T_{o2})}{1.000.000} \quad \text{(MW)} \qquad (3\text{-}35)$$

3-4 PREMASS

Equations (3-9) - (3-10)

5-6 MIXER

Equations (3-11) - (3-12)

3-4 PREMASS

Equation (3-13)

4-5 COMBUSTION CHAMBER (BURNER)

Equations (3-14) - (3-18)

5-6 MIXER

Equations (3-19) - (3-21)

6-7 COMPRESSOR TURBINE

$$T_{o7}=T_{o6}-\frac{CW}{C_{ph}\cdot\dot{m}_6\cdot 0{,}000001} \tag{3-36}$$

$$P_{o7}=\left[P_{o6}\cdot\left(1-\frac{1-\frac{T_{o7}}{T_{o6}}}{\eta_{isct}}\right)^{\frac{\gamma_h}{\gamma_h-1}}\right] \tag{3-37}$$

Equation (3-26)

8-9 EXHAUST

$$P_{i9}=P\alpha\cdot 1.003 \tag{3-38}$$

$$\dot{m}_9=\dot{m}_6 \tag{3-39}$$

7-8 POWER TURBINE

$$P_{o8}=\frac{P_{o9}\cdot 100}{100-DPexhloss} \tag{3-40}$$

$$T_{o8}=T_{o7}\cdot\left[1-\left[1-(\frac{P_{o8}}{P_{o7}})^{(\gamma_h-1)/\gamma_h}\right]\cdot n_{ispt}\right] \tag{3-41}$$

$$\dot{m}_8=\dot{m}_6 \tag{3-42}$$

$$PTW=\frac{\dot{m}_7\cdot C_{ph}\cdot(T_{o7}-T_{o8})}{1.000.000}\quad (MW) \tag{3-43}$$

8-9 EXHAUST

$$T_{o9}=T_{o8}\cdot(1-\frac{DTexhloss}{100}) \tag{3-44}$$

PERFORMANCE

$$UW=PTW\qquad (MW) \tag{3-45}$$

Equations (6-30) and (6-32) - (6-34)

The results of the above calculation procedure, when varying TET and R_C -using the FORTRAN program (APPENDIX C.2) developed by the author- are represented in *Table 3.1* and *Fig. 3.4*.

Notice: For comments and result analysis see paragraph 3.4

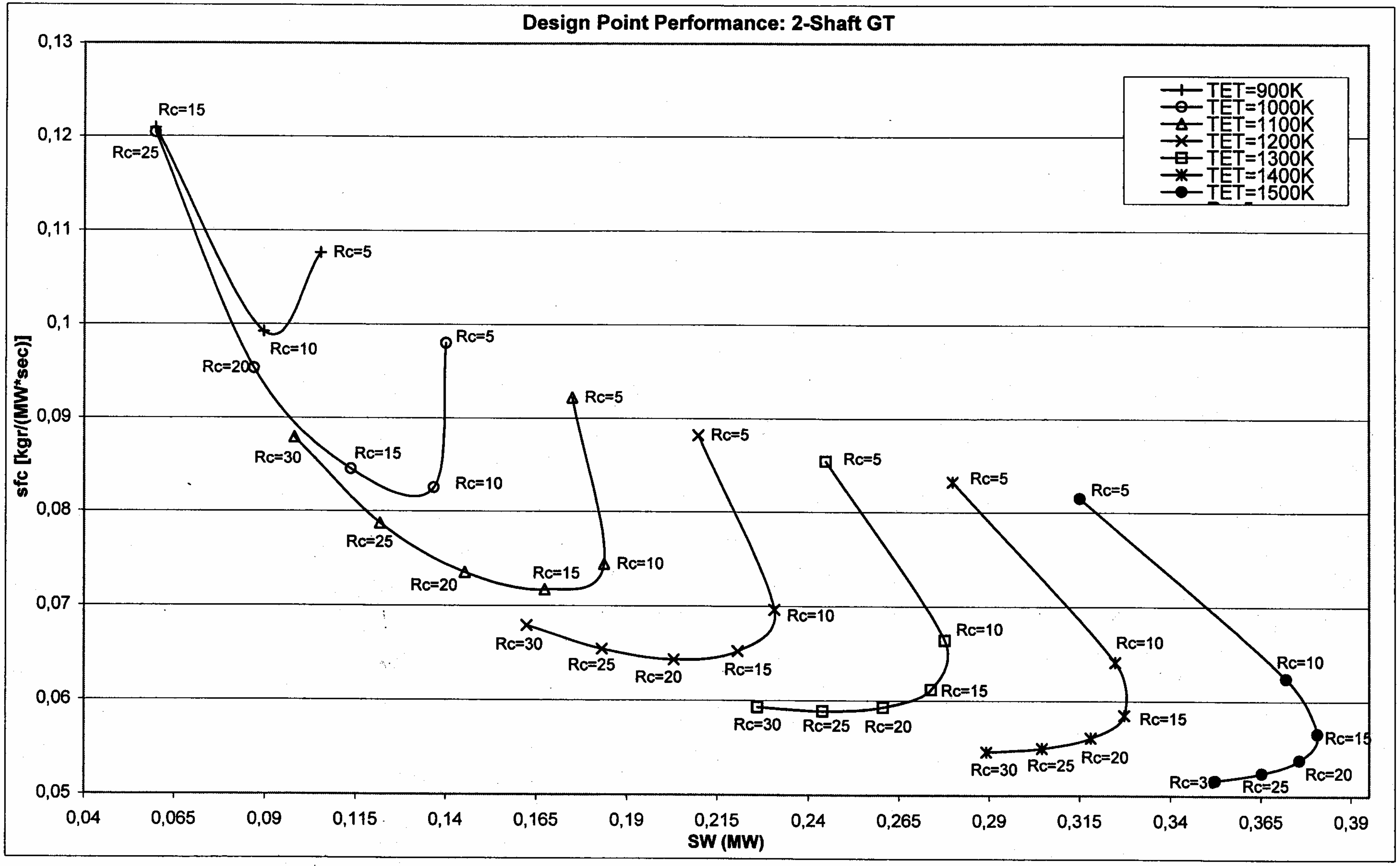

Fig. 3.4: 2-Shaft GT Design Point performance

3.3.3 1-Shaft with Heat Exchanger Cycle Simulation Procedure

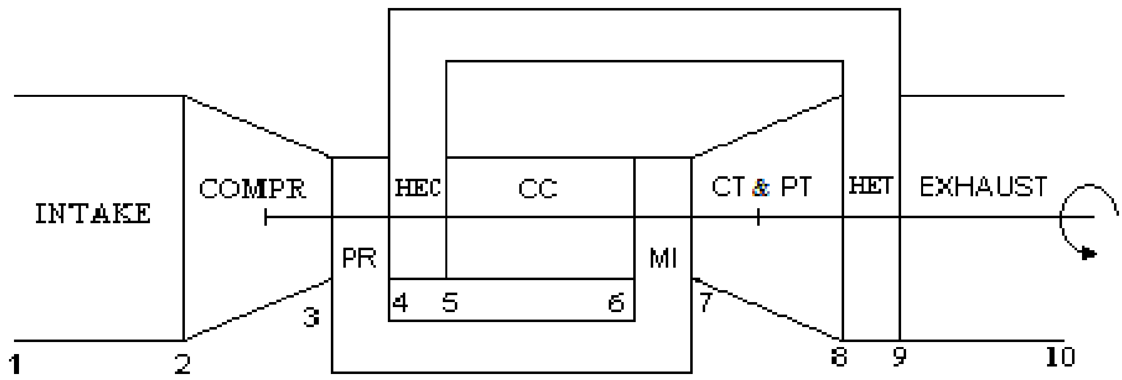

Fig. 3.5: 1-Shaft with heat exchanger cycle engine layout

Input File (APPENDIX C.1)

It is the same with those in paragraph 3.3.1 with the addition of the inputs for the two more components heat exchanger cold (HEC), heat exchanger hot (HET), simulating the heat exchanger.

Heat Exchanger inputs

The heat exchanger has effectiveness about 0.7÷0.85. So, we assume efficiencies for the two simulation components HEC and HET: $\eta_{che} = \eta_{hhe}$=0.84÷0.92 ($\eta_{che} = \eta_{hhe}$=0.9).
Pressure losses in the two components HEC and HET: DPcheloss=DPhheloss=1-4%

Calculation procedure

1-2 INTAKE
Equations (3-1) - (3-5)
2-3 COMPRESSOR
Equations (3-6) - (3-8) and (3-35)
3-4 PREMASS
Equations (3-9) - (3-10)
6-7 MIXER
If TET = T_{o6} is less than 1,300 then it is assumed that there is no need for by pass mass flow to cool the turbine section, thus

$$D_{mc}=0, \ DT_{cooling}=0 \tag{3-46}$$

On the contrary, if T_{o6} is grater or equal to 1,300 then there is a portion of mass flow (D_{mc}), which reduces the TET per $DT_{cooling}$:

$$D_{mc}=0.025*T_{o6}-25, \ DT_{cooling}=0.333*T_{o6}-333.333 \tag{3-47}$$

3-4 PREMASS
Equation (3-13)

4-5 HEAT EXCHANGER COLD

$$P_{o5} = P_{o4} \cdot \left(1 - \frac{DPcheloss}{100}\right) \tag{3-48}$$

A hypothetical temperature is assumed, which approaches the outlet of the component HEC. This is done in order to help the calculation to continue and to find the "real" value of T_{o5} which is then compared with the hypothetical. This iteration is repeated until those two temperatures are almost equal. και συγκρίνεται με την υποθετική.
So, firstly it is assumed that

$$T_{o5hyp} = 1{,}100 \tag{3-49}$$

$$\dot{m}_5 = \dot{m}_4 \tag{3-50}$$

5-6 COMBUSTION CHAMBER (BURNER)

$$P_{o6} = P_{o5} \cdot \left(1 - \frac{DPccloss}{100}\right) \tag{3-51}$$

$$T_{o6} = TET \tag{3-52}$$

$$\dot{m}_f = \frac{\dot{m}_5 \cdot (C_{ph} \cdot T_{o6} - C_{pc} \cdot T_{o5hyp})}{\eta_{cc} \cdot FCV \cdot 1.000.000)} \tag{3-53}$$

$$\dot{m}_6 = \dot{m}_5 + \dot{m}_f \tag{3-54}$$

$$FAR_{56} = \dot{m}_f / \dot{m}_5 \tag{3-55}$$

6-7 MIXER

$$P_{o7} = P_{o6} \tag{3-56}$$

$$T_{o7} = T_{o6} - DTcooling \tag{3-57}$$

$$\dot{m}_7 = \dot{m} + \dot{m}_f \tag{3-58}$$

9-10 EXHAUST

$$P_{o10} = P\alpha \cdot 1{,}003 \tag{3-59}$$

$$\dot{m}_{10} = \dot{m}_7 \tag{3-60}$$

8-9 HEAT EXCHANGER HOT

$$P_{o9} = \frac{P_{o10} \cdot 100}{100 - DPexhloss} \tag{3-61}$$

$$\dot{m}_9 = \dot{m}_7 \tag{3-62}$$

7-8 COMPRESSOR & POWER TURBINE

$$P_{o8} = P_{o9} \cdot (1 + \frac{DThheloss}{100}) \tag{3-63}$$

$$T_{o8} = T_{o7} \cdot \left[1 - \left[1 - (\frac{P_{o8}}{P_{o7}})^{(\gamma_h - 1)/\gamma_h}\right] \cdot n_{ist}\right] \tag{3-64}$$

At that point, it should be checked that the heat exchanger is working in a mode of inverse operation. So we check that

$$T_{o8} > T_{o4} \tag{3-65}$$

If that condition is not always fulfilled then the calculation should stop.

$$\dot{m}_8 = \dot{m}_7 \tag{3-66}$$

8-9 HEAT EXCHANGER HOT

$$T_{o9} = T_{o8} - \eta_{hhe} \cdot \left(T_{o8} - T_{o4}\right) \quad (3\text{-}67)$$

$$T_{o5re} = T_{o4} + \eta_{che} \cdot \left(T_{o8} - T_{o4}\right) \quad (3\text{-}68)$$

$$\text{If } \left|\frac{T_{o5hyp} - T_{o5re}}{T_{o5re}}\right| < 0.0001 \text{ then } T_{o5hyp}=T_{o5re}, \quad (3\text{-}69)$$

If that condition is not true then a new T_{o5hyp} equal with the "real" -which have been already calculated- assumed and the procedure is repeated until converge is occur

9-10 EXHAUST

$$T_{o10} = T_{o9} \cdot (1 - \frac{DTexhloss}{100}) \quad (3\text{-}70)$$

PERFORMANCE

$$CW = \frac{\dot{m}_2 \cdot C_{pc} \cdot (T_{o3} - T_{o2})}{1.000.000} \quad (MW) \quad (3\text{-}71)$$

$$TW = \frac{\dot{m}_7 \cdot C_{ph} \cdot (T_{o7} - T_{o8})}{1.000.000} \quad (MW) \quad (3\text{-}72)$$

Equations (3-30) - (3-34)

The results of the above calculation procedure, when varying TET and R_C -using the FORTRAN program (APPENDIX C.2) developed by the author- are represented in *Table 3.2* and *Fig. 3.6*.

Notice: For comments and result analysis see paragraph 3.4

Table 3.2: 1-Shaft GT HE and 2-Shaft GT HE Design point performance

		1-Shaft GT HE			2-Shaft GT HE		
TET (K)	**Rc**	**SW [(MW*sec)/kgr]**	**Sfc [kgr/(MW*sec)]**	**η_{th}**	**SW [(MW*sec)/kgr]**	**Sfc [kgr/(MW*sec)]**	**η_{th}**
900	5	0.0986	0.0851	0.2419	0.1853	0.0528	0.3896
	10	-	-	-	0.1708	0.0656	0.3137
	15	-	-	-	-	-	-
	20	-	-	-	-	-	-
	25	-	-	-	-	-	-
	30	-	-	-	-	-	-
1,000	5	0.132	0.0715	0.2879	0.2258	0.0485	0.4246
	10	0.128	0.0861	0.2389	0.2214	0.0565	0.3639
	15	-	-	-	-	-	-
	20	-	-	-	-	-	-
	25	-	-	-	-	-	-
	30	-	-	-	-	-	-
1,100	5	0.1654	0.0633	0.3249	0.2663	0.0454	0.4530
	10	0.173	0.0708	0.2906	0.2722	0.0508	0.4047
	15	-	-	-	0.2572	0.0572	0.3595
	20	-	-	-	-	-	-
	25	-	-	-	-	-	-
	30	-	-	-	-	-	-
1,200	5	0.1989	0.0579	0.3553	0.307	0.0432	0.4766
	10	0.2182	0.0618	0.3331	0.323	0.0469	0.4386
	15	-	-	-	0.3133	0.0514	0.4000
	20	-	-	-	0.2976	0.0563	0.3657
	25	-	-	-	-	-	-
	30	-	-	-	-	-	-
1,300	5	0.2325	0.054	0.3807	0.3477	0.0414	0.4965
	10	0.2635	0.0558	0.3686	0.374	0.044	0.4673
	15	0.259	0.0608	0.3382	0.3696	0.0474	0.4341
	20	-	-	-	0.3573	0.0509	0.4041
	25	-	-	-	0.3425	0.0546	0.3771
	30	-	-	-	-	-	-
1,400	5	0.2661	0.0511	0.4024	0.3886	0.0401	0.5136
	10	0.3089	0.0516	0.3988	0.4252	0.0418	0.4919
	15	0.3104	0.055	0.3739	0.426	0.0444	0.4634
	20	0.3007	0.0592	0.3476	0.4171	0.0471	0.4369
	25	-	-	-	0.4048	0.0498	0.4129
	30	-	-	-	0.3912	0.0526	0.3909
1,500	5	0.2999	0.0489	0.4210	0.4295	0.0389	0.5284
	10	0.3544	0.0484	0.4248	0.4764	0.0401	0.5132
	15	0.3619	0.0508	0.4047	0.4825	0.0421	0.4888
	20	0.3562	0.0539	0.3821	0.4772	0.0442	0.4654
	25	-	-	-	0.4674	0.0464	0.4439
	30	-	-	-	0.4557	0.0485	0.4241

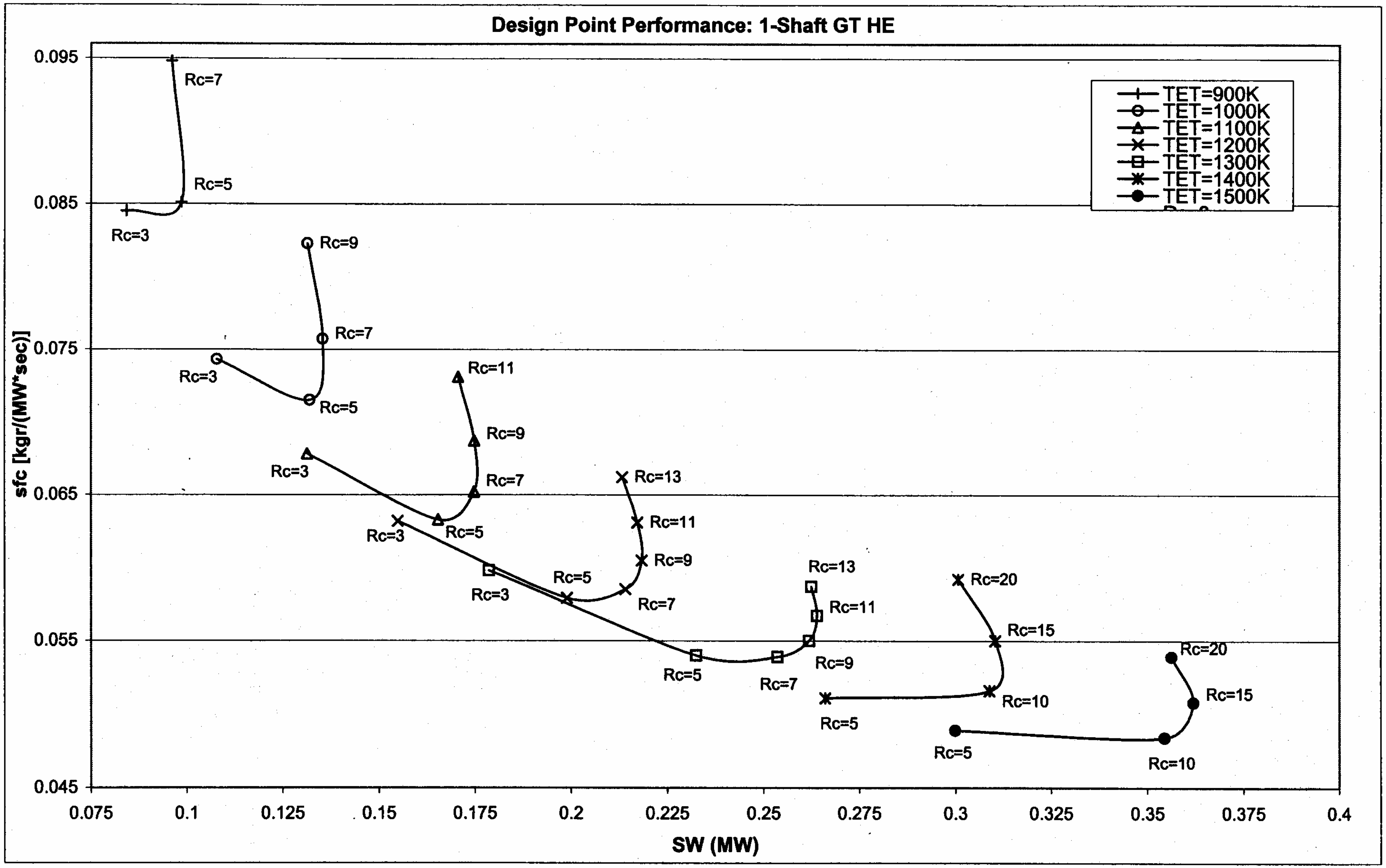

Fig. 3.6: 1-Shaft GT HE Design Point performance

3.3.4 2-Shaft with Heat Exchanger Cycle Simulation Procedure

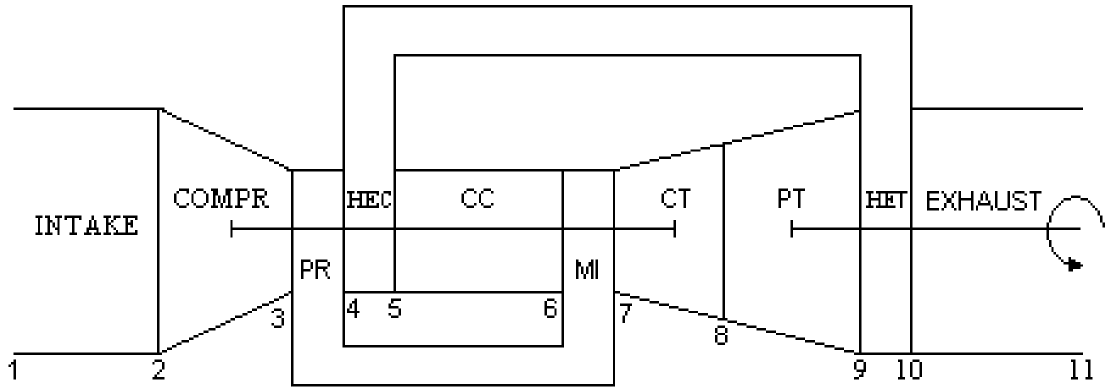

Fig. 3.7: 2-Shaft with heat exchanger cycle engine layout

Input File (APPENDIX C.1)

Is the same with those in paragraph 3.3.2 with the addition of the inputs for heat exchanger of paragraph 3.3.3.

Calculation procedure

1-2 INTAKE
Equations (3-1) - (3-5)
2-3 COMPRESSOR
Equations (3-6) - (3-8) and (3-35)
3-4 PREMASS
Equations (3-9) - (3-10)
6-7 MIXER
Equations (3-46) - (3-47)
3-4 PREMASS
Equation (3-13)
4-5 HEAT EXCHANGER COLD
Equations (3-48) - (3-50)
5-6 COMBUSTION CHAMBER (BURNER)
Equations (3-51) - (3-55)
6-7 MIXER
Equations (3-56) - (3-58)
8-9 COMPRESSOR TURBINE

$$T_{o8}=T_{o7}-\frac{CW}{C_{ph}\cdot \dot{m}_7\cdot 0.000001} \tag{3-73}$$

$$P_{o8}=P_{o7}\cdot\left(1-\frac{1-\frac{T_{o6}}{T_{o6}}}{\eta_{isct}}\right)^{\frac{\gamma_h}{\gamma_h-1}} \tag{3-74}$$

$$\dot{m}_8=\dot{m}_7 \tag{3-75}$$

10-11 EXHAUST

$$P_{ï11} = P\alpha \cdot 1.003 \quad (3\text{-}76)$$

$$\dot{m}_{11} = \dot{m}_7 \quad (3\text{-}77)$$

9-10 HEAT EXCHANGER HOT

$$P_{o10} = \frac{P_{o11} \cdot 100}{100 - \text{DPexhloss}} \quad (3\text{-}78)$$

$$\dot{m}_{10} = \dot{m}_7 \quad (3\text{-}79)$$

8-9 POWER TURBINE

$$P_{o9} = P_{o10} \cdot (1 + \frac{\text{DThheloss}}{100}) \quad (3\text{-}80)$$

$$T_{o9} = T_{o8} \cdot \left[1 - \left[1 - (\frac{P_{o8}}{P_{o7}})^{(\gamma_h - 1)/\gamma_h}\right] \cdot n_{ispt}\right] \quad (3\text{-}81)$$

At that point, it should be checked that the heat exchanger is working in a mode of inverse operation. So, we check that

$$T_{o9} > T_{o4} \quad (3\text{-}82)$$

If that condition is not always fulfilled then the calculation should stop.

$$\dot{m}_9 = \dot{m}_7 \quad (3\text{-}83)$$

$$\text{PTW} = \frac{\dot{m}_8 \cdot C_{ph} \cdot (T_{o8} - T_{o9})}{1.000.000} \quad (\text{MW}) \quad (3\text{-}84)$$

9-10 HEAT EXCHANGER HOT

$$T_{o10} = T_{o9} - \eta_{hhe} \cdot (T_{o9} - T_{o4}) \quad (3\text{-}85)$$

$$T_{o5re} = T_{o4} + \eta_{che} \cdot (T_{o9} - T_{o4}) \quad (3\text{-}86)$$

$$\text{If } \left|\frac{T_{o5hyp} - T_{o5re}}{T_{o5re}}\right| < 0.0001 \text{ τότε } T_{o5hyp} = T_{o5re} \quad (3\text{-}87)$$

If that condition is not true then a new T_{o5hyp} equal with the “real” -which have been already calculated- assumed and the procedure is repeated until converge is occur

10-11 EXHAUST

$$T_{o11} = T_{o10} \cdot (1 - \frac{\text{DTexhloss}}{100}) \quad (3\text{-}88)$$

PERFORMANCE

Equations (3-45), (3-30) and (3-32) - (3-34)

The results of the above calculation procedure, when varying TET and R_C -using the FORTRAN program (APPENDIX C.2) developed by the author- are represented in *Table 3.2* and *Fig. 3.7.*

Notice: For comments and result analysis see paragraph 3.4

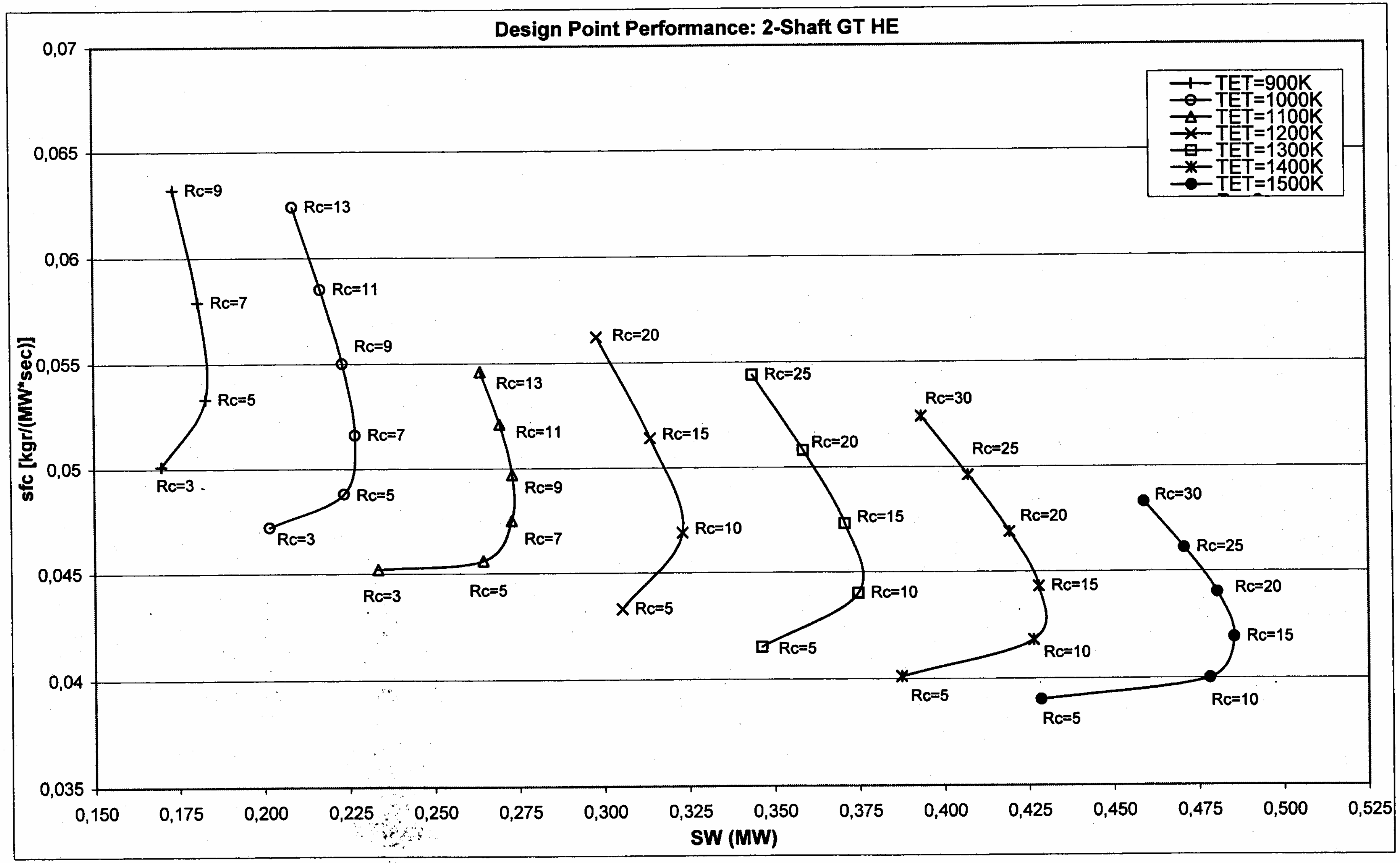

Fig. 3.8: 2-Shaft GT HE Design Point performance

3.4 Design Point Performance Results Analysis

Industrial engine selection is directly related to the type of application that the engine will be assigned to undertake. Power generation applications most usually require operation of the gas turbine at base load conditions. The operating time intervals are considered to be very long, when compared to other applications such as, aero or emergency power gas turbines.

For power generation purposes, two fundamental engine configurations are most commonly used. These are the 1-shaft engines and the 1-shaft ones. In **1-shaft engines** the entire rotating components of the engine are articulated in one rotating shaft. This implies that the engine's turbine is assigned to provide power to the engine's compressor and at the same time, produce the demanded output power in order to drive an electricity generator via a shaft. In **2-shaft engines**, the turbine of the gas generator is assigned to provide power to the compressor only. The compressor turbine is not connected via a shaft to any other engine component. The required power for the electricity generator is provided by another turbine, namely the power turbine, which rotates on a separate shaft. The power turbine is placed after the gas generator turbine, and it is aerodynamically connected to it. Thus, the required power to drive the power turbine is provided by the hot gases exiting the gas generator at a specific temperature and pressure. It is evident that in 1-shaft engines the turbine expands directly to ambient pressure, while in the 2-shaft engine the power generator turbine expands, not to ambient but to a certain pressure level and the ambient expansion is accomplished by the power turbine.

By comparing the results of the DP performance for the 1-shaft and 2-shaft configuration, it can be observed that the performance (η_{th}, SW) is **almost the same**. However must be noted that they are slightly better in the case of 2-shaft (*Tables 3.1, 3.2*). The difference would be greater if the same isentropic efficiencies of the turbines for the 1-shaft and 2-shaft configuration in the programming are used. Normally, the performance at the design point should be the same. So, the author assumed different isentropic efficiencies of the turbines as mentioned in paragraph 3.3.2. The slight difference is explained by the fact that the assumed turbine's isentropic efficiencies are not exact and this difference becomes more pronounced when TET is low and R_C is high.

As previously mentioned two different types of thermodynamic cycles are selected: simple and with the use of a heat exchanger. The reasons for this selection are the simplicity, which characterizes the first, and the higher thermal efficiency, which is characteristic of the second one.

A brief resume the analyses of ideal gas turbine cycles can be very helpful to understand the shape of the curves in final diagrams SW-sfc. (*Figs 3.2, 3.4, 3.6)*. The assumption of ideal conditions will be taken to imply the following:

- Compression and expansion processes are reversible and adiabatic, i.e. isentropic.
- The change of kinetic energy of the working fluid between inlet and outlet of each component is negligible.
- There are no pressure losses in the components.
- The working fluid is a perfect gas with constant specific heats.

- The mass flow of gas is constant throughout the cycle. (f) Heat transfer in a heat-exchanger is 'complete" so that in conjunction with (d) and (e) the temperature rise on the cold side is the maximum possible and exactly equal to the temperature drop on the hot side.

Simple gas turbine cycle

The ideal cycle for the simple gas turbine is the Joule (or Brayton) cycle, i.e. cycle 1234 in *Fig. 3.9.*

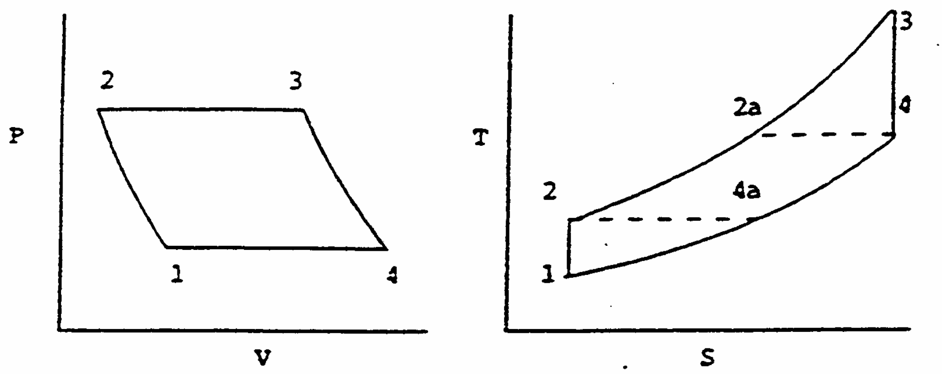

Fig. 3.9: Ideal simple cycle, [16]

The first thermodynamic law for steady flow with relevant slow velocities or small velocity differences and for open systems can be written:

where *Q* and Ware the heat and work transfers per unit mass flow. Applying this to each component, bearing in mind the above assumptions, we have

$$q - w = \left[h_2 - h_1 + \frac{\vec{V}_2^2 - \vec{V}_1^2}{2} \right] \quad (3\text{-}89) \Rightarrow$$

$$\Rightarrow q - w = h_{exit} - h_{inlet} \qquad (3\text{-}90)$$

where q and w are the heat and work transfers per unit mass flow.

Applying this to each component, bearing in mind the previous assumptions we have:

<u>1-2 Adiabatic compression (q = 0)</u>

$(3\text{-}90) \Rightarrow 0 - w_{12} = h_2 - h_1 \Rightarrow w_{12} = -(h_2 - h_1) \Rightarrow [h = C_p T]$

$$\Rightarrow \text{Compression "Work", } CW = -C_p (T_2 - T_1) \Rightarrow CW = -C_P T_2 \left(1 - \frac{T_1}{T_2} \right) \text{(J/kgr)} \quad (3\text{-}91)$$

Notice: Compression "Work", is actually Compression power because

$$CP = \dot{m} \cdot CW = 1 \cdot CW = CW \quad (W)$$

<u>2-3 Constant pressure combustion (w = 0)</u>

$(3\text{-}1) \Rightarrow q_{23} - 0 = h_3 - h_2 \Rightarrow q_{23} = (h_3 - h_2) \Rightarrow$

$$\Rightarrow \text{Heat input, } HI = C_p (T_3 - T_2) \qquad (3\text{-}92)$$

3-4 Adiabatic expansion (q = 0)

(3-1) $\Rightarrow 0 - w_{34} = h_4 - h_3 \Rightarrow w_{34} = -(h_4 - h_3) \Rightarrow w_{34} = (h_3 - h_4) \Rightarrow$

$$\Rightarrow \text{Expansion "Work", EW} = C_p\,(T_3 - T_4) \Rightarrow EW = C_P\,T_3\left(1 - \frac{T_4}{T_3}\right) \quad (3\text{-}93)$$

Bearing in mind the perfect gas laws, we have

$$\left(\frac{T_2}{T_1}\right) = \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} \Rightarrow (P_1 = P_4,\ P_2 = P_3) \Rightarrow \left(\frac{P_4}{P_3}\right)^{\frac{\gamma-1}{\gamma}} = \frac{T_4}{T_3} \Rightarrow \frac{T_1}{T_2} = \frac{T_4}{T_3} \quad (3\text{-}94)$$

The **Useful "Work", UW** is then given by

$$UW = TW + CW \Rightarrow (3\text{-}91),\ (3\text{-}93) \Rightarrow UW = C_P\,T_3\left(1 - \frac{T_4}{T_3}\right) - C_P\,T_2\left(1 - \frac{T_1}{T_2}\right) \Rightarrow$$

$$\Rightarrow (3\text{-}94) \Rightarrow UW = C_P\,(T_3 - T_2)\left(1 - \frac{T_1}{T_2}\right) \Rightarrow \quad (3\text{-}95)$$

$$\Rightarrow UW = C_P\,[T_3 - T_1 R_C^{(\gamma-1)/\gamma}]\,[1 - R_C^{-(\gamma-1)/\gamma}] \quad (3\text{-}96)$$

where $Rc = P_2/P_1$ is the pressure ratio.

The cycle thermal efficiency (**η_{th}**) is given by

$$\eta_{th} = \frac{UW}{HI} = \frac{C_p(T_3 - T_2)\cdot\left(1 - \frac{T_1}{T_2}\right)}{C_P(T_3 - T_2)} \Rightarrow \eta_{th} = 1 - \frac{T_1}{T_2} \quad (3\text{-}97)$$

$$\left(\frac{T_2}{T_1}\right) = \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} \Rightarrow \frac{T_1}{T_2} = \frac{1}{\left(\frac{P_2}{P_1}\right)^{(\gamma-1)/\gamma}} = \frac{1}{(R_C)^{(\gamma-1)/\gamma}} \Rightarrow$$

$$\Rightarrow \eta_{th} = 1 - \frac{1}{R_C^{(\gamma-1)/\gamma}} \Rightarrow \eta_{th} = 1 - \left(\frac{1}{R_C}\right)^{(\gamma-1)/\gamma} \quad (3\text{-}98)$$

Eq. (3-98) implies that the thermal efficiency depends on γ (*Fig. 3.10*) and on Rc (*Figs 3.10 and 3.13b*). *Fig 3.10* shows the variation of η_{th} in respect of Rc for different γ values.

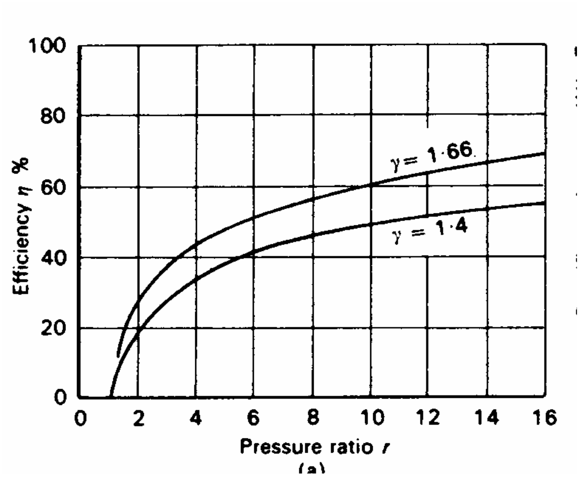

Fig. 3.10: η_{th} in respect of Rc for different γ values (Ar or He: γ=1.66, air, γ=1.4) [16]

From *Fig 3.12b* it is obvious that when Rc increases the η_{th} also increases towards the unity. Equation (3-96) implies that the UW depends on T_3=TET which in turn depends on the cooling process, construction materials of the turbine blades. The value of the ratio T_3/T_1 is usually in the range of 3.5-4 for the industrial GTs.

The UW=0 (*Fig. 3.11*) when

a) $T_2 = T_3$ no combustion has taken place, thus Rc = $\left(\frac{T_3}{T_1}\right)^{\gamma/(\gamma-1)}$

b) $T_1 = T_2$ no compression has taken place, thus Rc =1 $\Rightarrow P_1 = P_2$

$$Rc \rightarrow \left(\frac{T_3}{T_1}\right)^{\gamma/(\gamma-1)} \qquad Rc \rightarrow R_C^{opt} \qquad Rc \rightarrow 1$$

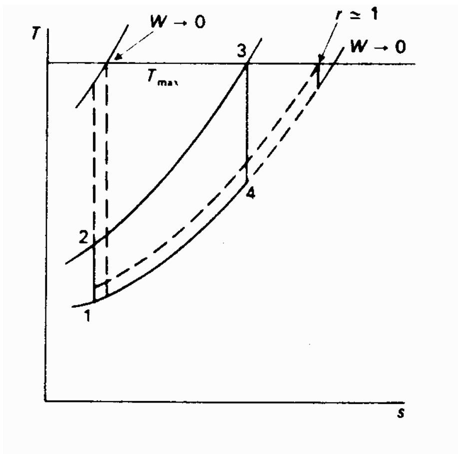

Fig. 3.11: Modification of the simple cycle accordingly to the Rc variations, [14]

It becomes obvious that if we want to draw the UW against the Rc, we would observe the existence of an upper limit. (*Fig. 3-12a*). The Rc value where the UW is getting this higher value can be found with the following procedure:

$$\frac{\partial(UW)}{\partial T_2} = 0 \Rightarrow T_2 = \sqrt{T_1 \cdot T_3} \Rightarrow \frac{T_2}{T_1} = \sqrt{\frac{T_3}{T_1}} \quad (3\text{-}99)$$

Because the two processes 1-2 και 3-4 are adiabatic between same pressures P_1–P_2 results that:

$$\frac{T_1}{T_2} = \frac{T_4}{T_3} \Rightarrow \frac{T_1 \cdot T_3}{T_2} = T_4 \Rightarrow \sqrt{T_1 \cdot T_3} = \sqrt{T_2 T_4} \Rightarrow (3\text{-}10) \Rightarrow T_2 = \sqrt{T_2 T_4} \Rightarrow T_2 = T_4 \quad (3\text{-}100)$$

which means that the compressor and turbine exit temperatures are equal and so

$$R_C^{opt} = \left(\frac{T_3}{T_1}\right)^{\gamma/(2(\gamma-1))} \quad (3\text{-}101)$$

For Rc values that are between 1 to R_C^{opt}, then T_2 is less than T_4, so with the use of a heat exchanger we can take advantage of the excess amount of heat which is due to temperature difference T_4 - T_2 in order the compressed air to be preheated before entering the combustion chamber (Brayton Cycle with Heat Exchanger).

The **specific "Work", SW** is defined as

$$SW = \frac{UW}{\dot{m}} \quad (3\text{-}102)$$

and if $\dot{m}=1$, as it is assumed previously, then SW = UW, and in dimensionless form:

$$\frac{SW}{C_P T_1} = \frac{T_3}{T_1} \cdot \left[1 - R_C^{(\gamma-1)/\gamma}\right] \cdot \left[R_C^{(\gamma-1)/\gamma} - 1\right] = \frac{(T_3 - T_2) \cdot (T_2 - T_1)}{T_2 \cdot T_1} \quad (3\text{-}103)$$

(a) (b)

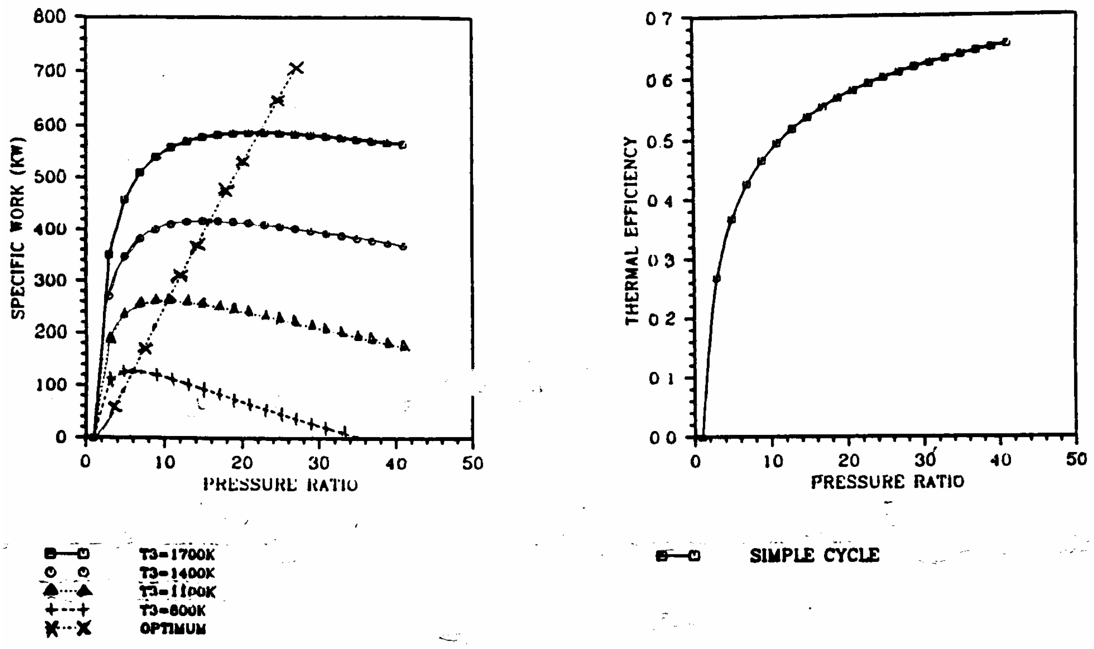

Fig. 3.12: (a) SW - Rc for different T3 (b) η_{th} – Rc, [16]

The SW is giving an idea of the GT's size and especially of the front area of the GT or the mass flow getting in the GT. From *Fig. 3.12a* it can be concluded that for constant T_3, there is an Rc value for which the SW becomes greatest. Also, as the T_3 increases, the SW does the same and so the size of the GT.

When component losses are taken into account, the efficiency of the simple cycle becomes dependent upon the maximum cycle temperature, TET as well as pressure ratio R_C. Furthermore, for each TET, the efficiency has a peak value at a particular pressure ratio. As Rc increases, there is a reduction in fuel supply in order to reach the fixed TET, which increases the efficiency. However, there is also a decrease in efficiency at high-pressure ratios because of the increased compressor work and this effect is the dominant one. (*Fig.3.13*) Based on the same consideration and from the same figure, we can conclude that there will be a maximum value for SW when R_C increases and TET is constant.

Although the optimum pressure ratio for maximum efficiency differs from that for maximum specific output, the curves are fairly (*Figs 3.2, 3.4*) flat near the peak and a pressure ratio between the two optima can be used without much loss in efficiency. It is worth pointing out that the lowest pressure ratio, which will give an acceptable performance, is always chosen, as it might even be slightly lower than either optimum value. Mechanical design considerations such as the number of compressor and turbine stages required, the avoidance of excessively small blades at the high-pressure end of the compressor, and whirling speed and bearing problems associated with the length of the compressor-turbine combination, may affect the choice.

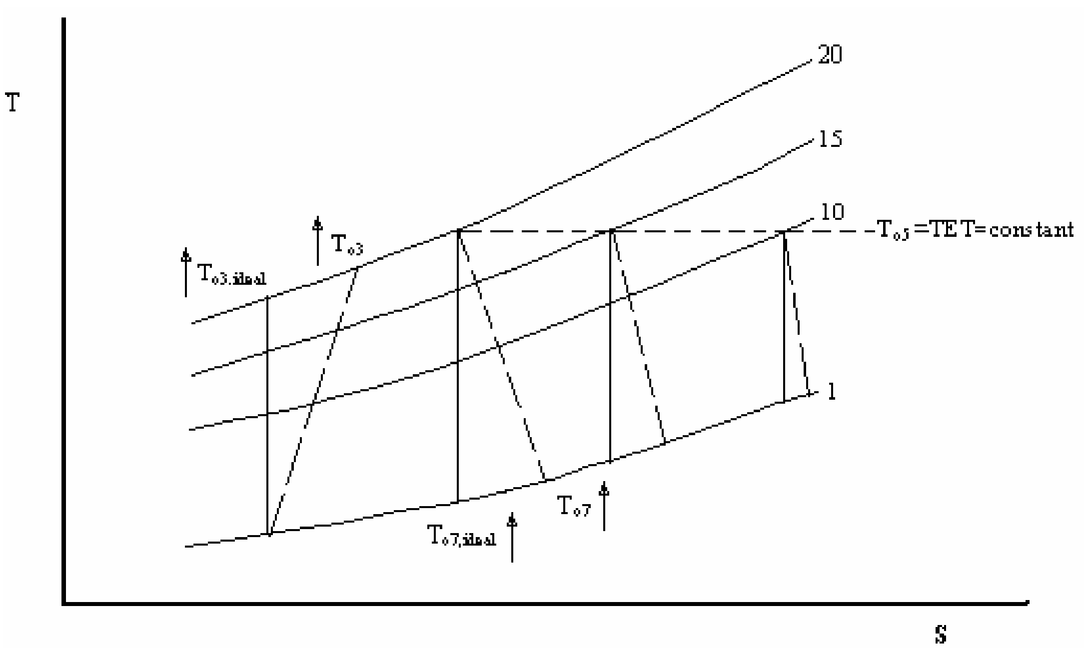

Fig. 3.13: Explanation of maximum η_{th}, when R_C increases for constant TET

The advantage of using as higher as possible TET, and the need to use a higher-pressure ratio to take advantage of a higher permissible temperature, is evident from the curves (*Figs 3.2, 3.4, 3.6*). The efficiency increases with TET because the component losses become relatively less important as the ratio of positive turbine work to negative compressor work increases, although the gain in efficiency becomes marginal as TET is increased beyond 1,300K (particularly if a higher temperature requires a complex turbine blade cooling system which incurs additional losses). (*Fig. 3.14*) Extremely high TET also implies the use of high performance materials or hi-tech coatings for the turbine blades which raises sufficiently the purchase cost of the GT. Finally, the higher the TET the more frequent the service should be (lower reliability).

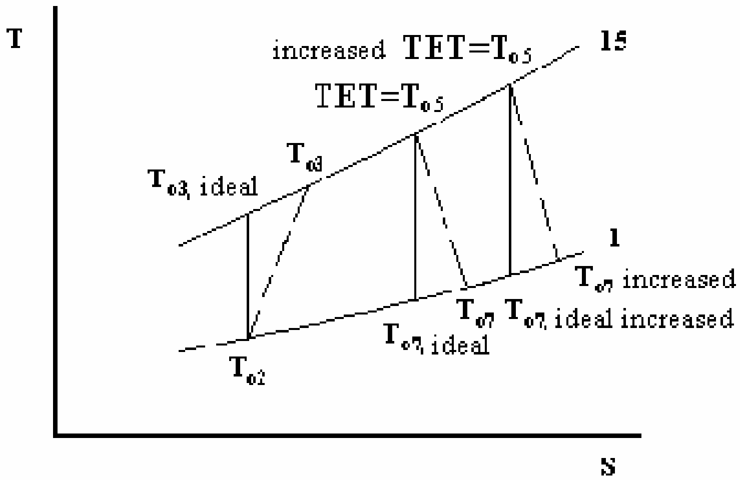

Fig. 3.14: Explanation of increasing η_{th}, when TET increases for constant R_C

There is a small gain, however, in specific work output with an increase in TET. The consequent reduction in the size of a plant for a given power is very marked, and this is particularly important for aircraft or for small industrial plant gas turbines.

Cycle with Heat Exchanger or Recuperated or Regenerated

The most significant modification to the Brayton Simple Cycle and enhance his thermal efficiency, is the use of a heat exchanger. The basic idea is to take advantage of the exhaust gases' heat (temperature T_4), in order the compressed air to be preheated.

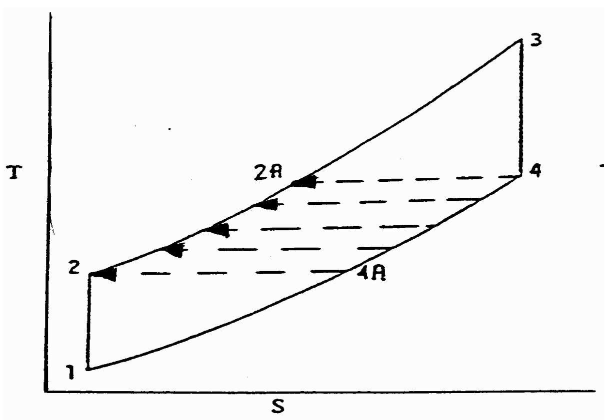

Fig. 3.15: Cycle with Heat Exchanger, [16]

With this modification the heat input (HI) reduces, while the compressor and turbine work remain unchanged. Hence, the use of heat exchanger has meaning only when exhaust gas temperature is sufficiently greater than the compressor exit temperature. Assuming that the heat exchanger is ideal, results that $T_{2A} = T_4$ and $T_{4A} = T_2$ (*Fig.3.15)*.

Compressor and expansion work (CW and EW) are calculated as in the case of the simple cycle, mentioned previously. Thus, the quantities useful work (UW) and specific work (SW) are remaining unchanged (*Fig.3.16*a). Although, the HI changes, and becomes:

$$\left.\begin{array}{l} HI = (h_3 - h_{2a}) \Rightarrow HI = C_p\,(T_3 - T_{2a}) \\ \qquad\qquad\qquad\qquad T_{2a} = T_4 \end{array}\right| \Rightarrow$$

$$\left.\begin{array}{l} \Rightarrow HI = C_p\,(T_3 - T_4) \Rightarrow HI = C_p T_3 \left(1 - \dfrac{T_4}{T_3}\right) \\ \text{From (3-94)} \Rightarrow \dfrac{T_1}{T_2} = \dfrac{T_4}{T_3} \end{array}\right| \Rightarrow HI = C_p T_3 \left(1 - \dfrac{T_1}{T_2}\right) \qquad (3\text{-}104)$$

The thermal efficiency ($\eta_{th,HE}$) in this case becomes as following:

$$\eta_{th.re} = \frac{UW}{HI} = \frac{C_p\left(T_3 - T_2\right)\cdot\left(1 - \dfrac{T_1}{T_2}\right)}{C_p T_3\left(1 - \dfrac{T_1}{T_2}\right)} \Rightarrow \eta_{th.re} = 1 - \frac{T_2}{T_3} \;\; (3\text{-}16) \Rightarrow \eta_{th.re} = 1 - \frac{\dfrac{T_2}{T_1}}{\dfrac{T_3}{T_1}}$$

$$\left(\frac{T_2}{T_1}\right) = \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} \Rightarrow \left(\frac{T_2}{T_1}\right) = R_C^{(\gamma-1)/\gamma} \qquad \Rightarrow$$

$$\Rightarrow \eta_{th.re} = 1 - \frac{T_1}{T_3}\cdot R_C^{(\gamma-1)/\gamma} \qquad (3\text{-}105)$$

Eq. (3-98) implies that the thermal efficiency depends on γ and on Rc (*Fig. 3.16b*). If $\dfrac{T_3}{T_1}$=const., i.e. T_3 is defined then when the ratio Rc increases, the efficiency $\eta_{th.re}$ reduces. This behavior is opposite than that of the simple cycle. Thus, the requirement of for reduced Rc, leads to an increase to the efficiency, while there is an optimum Rc value -significally greater- for which the SW becomes highest (like in the case of simple cycle). (*Fig.3.16*)

As the Rc increases, then the $\eta_{th.re\ reduces}$ until the condition $R_C^{(\gamma-1)/\gamma} = \sqrt{\dfrac{T_3}{T_1}} \Rightarrow \dfrac{T_2}{T_1} = \sqrt{\dfrac{T_3}{T_1}}$ is fulfilled. In that case: $T_4 = T_2$, $n_{th.re} = n_{th}$ and the UW becomes highest.

When Rc > $\left[\sqrt{\dfrac{T_3}{T_1}}\right]^{\gamma/(\gamma-1)} \Rightarrow \dfrac{T_2}{T_1} > \sqrt{\dfrac{T_3}{T_1}} \Rightarrow T_4 < T_2$ and then the heat exchanger operates reversely!

(a) (b)

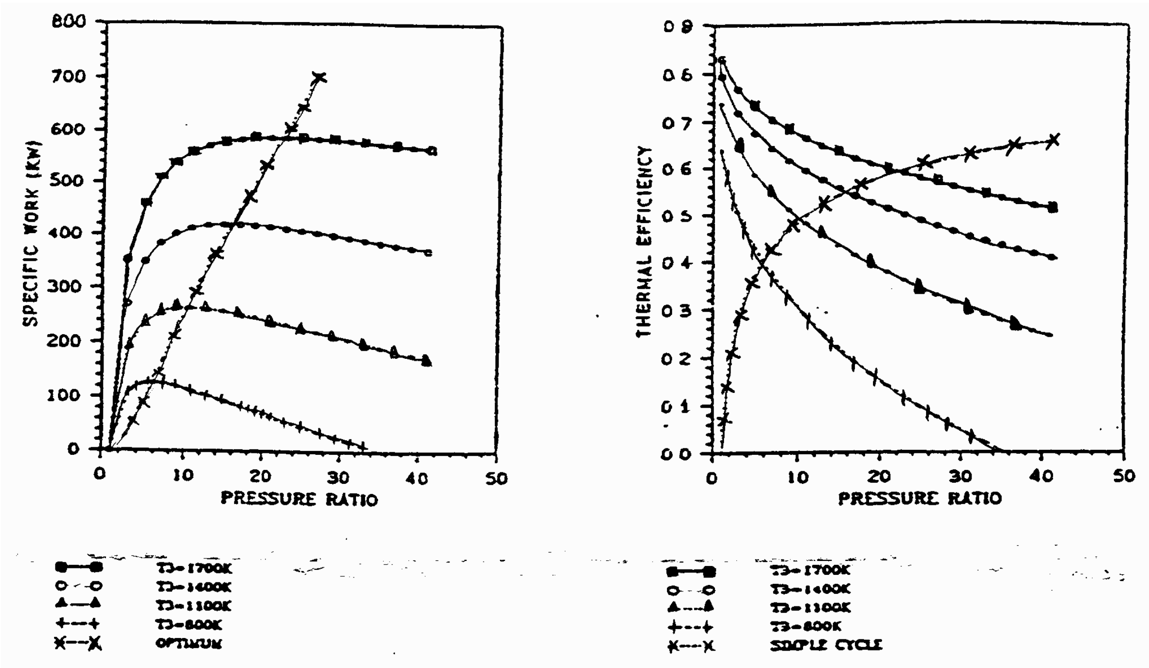

Fig. 3.16: (a) SW - Rc for different T3 (b) η_{th} – Rc, [16]

It has to be underlined, that this cycle configuration in order to operate sufficiently ($\eta_{th.re}$ increased), it is desirable that T_2 should to be kept relatively low, which means that Rc must be kept also low, resulting in efficiency increase. In other words, this means that a GT which operates with heat exchanger should have low pressure ratio.

Generally, the diagrams of *Figs 3.6* and *3.8* correspond at **cycle with heat exchanger;** have the same shape as *Figs 3.2* and *3.4.* However, there are some important differences.

In contrast to the simple cycle, when we travel on a hook curve clockwise the pressure ratio decreases.

As far as the specific work output is concerned, the addition of a heat exchanger merely causes a slight reduction due to the additional pressure losses.

The sfc on the contrary is substantially reduced, as shown in *Figs 3.6, 3.8.* Heat-exchange increases the efficiency substantially and markedly reduces the optimum pressure ratio for maximum efficiency. Thermal efficiency and specific power generally increase with TET. At optimum pressure ratios the thermal efficiency is approximately 10% better than that for the simple cycle, due to the heat recovery which reduces the fuel requirement. This difference decreases as TET increases because the corresponding simple cycle becomes more efficient. The optimum pressure ratio for thermal efficiency is comparatively low since the difference between the exhaust and compressor delivery temperatures is high, hence more heat may be recovered. This is the dominant effect of pressure ratio.

Unlike the corresponding efficiency curves for the ideal cycle, the actual efficiency curves do not rise to the Carnot value at R_C=1 but fall to zero at the pressure ratio at which the turbine provides just sufficient work to drive the compressor: at this point there will be positive heat input with zero net work output. The not proper performance, at the region where TET is low and the R_C is high, is because of the reverse operation of the heat exchanger.

The optimum R_C for SW depends on the heat exchanger effectiveness. It is very similar to that of a simple cycle, being only slightly reduced by the additional pressure losses in the heat exchanger. Effectiveness is one of the few component design parameters that have a strong effect upon the optimum pressure ratio for thermal efficiency. As effectiveness is increased the optimum pressure ratio decreases, since increased heat recovery offsets more easily the poor simple cycle efficiency.

3.5 Calculation Procedure of the Off-Design Performance

Focus was techno economics and not off-design programming. If required more detailed gas turbine codes could replace those used here. Nevertheless, the results are realistic.

The off-design point simulation is considered to be a substantially more complicated process than the design point one. The thermodynamic relationships used are similar in both cases but off design simulation requires additional considerations on the performance of the engine components at levels of operation which differ to those of design (part load, transient operation etc.)

In engine performance analysis, we consider the performance of an engine that was built (constructed physically or created mathematically) with a selected compressor pressure ratio and its corresponding turbine temperature ratio.

The off-design models attempted in this thesis are simplified versions of existing ones and are based on specific processes which have been discussed with the supervisor. It must be noted that for each different engine configuration, a different off-design model is required. Fundamental simplifications that have been used in this approach are considered to be the following:

- Instead of using a compressor map a simplified relationship is used.
- In the case of 1-shaft engines the turbine is assumed to be always choked.
- In the case of 2-shaft engines the compressor turbine is assumed to operate between choked nozzles, (the turbine temperature and pressure ratio remains essentially constant).

The off design performance of the following engines is simulated

1. 1-shaft GT
2. 2-shaft GT
3. 1-shaft GT with heat exchanger

The 2-shaft GT with heat exchanger was not simulated due to the problem caused by the contradictory restrictions of continuing operation of the compressor turbine between choked nozzle and continuous operation of the heat exchanger with positive temperature difference.

3.6 Off Design Performance Simulation

3.6.1 1-Shaft Simple Cycle Simulation Procedure

Referring to *Fig. 3.1*.

<u>***Input File (APPENDIX C.6)***</u>

Selection of the engine

There must be a selection of an engine from the design point performance. This selection should take into account several constraints and restrictions, depending on the specific application.

In order to test the correct response of the off-design simulation program, we choose an engine with the following parameters:

1. TET=1,300K
2. R_C=15

Variable Ambient Conditions

The simulation program includes three subroutines. Each subroutine simulates the off-design performance of the engine when one of the following parameters varies:

1. **Ambient temperature:** the operating temperatures of the engine are the average temperatures of each month of the region where the engine will operate.
2. **Ambient pressure:** the operating pressures of the engine are the average pressures of each month of the region where the engine will operate.
3. **Altitude:** the term altitude is used in order to indicate the simultaneous variation of both ambient temperature and pressure. The operating temperatures and pressures of the engine are the average values of each month of the region where the engine will operate.

Working fluid inputs

The mass flow $\dot{m}^{od}$ will be calculated from the program. The rest parameters are the same with those in paragraph 3.3.1.

Intake inputs

Is the same with those in paragraph 3.3.1.

Compressor inputs

Compressor pressure ratio R_{Cod}, will be calculated from the program.

Compressor isentropic efficiency, η_{iscod}, will be calculated from the program

Compressor degradation, P_{Cde} (%)=0

Premass inputs

Is the same with those in paragraph 3.3.1.

Combustion Chamber inputs

The fuel mass flow $\dot{m}_f^{od}$ will be calculated from the program. The rest parameters are the same with those in paragraph 3.3.1.

Mixer inputs

Is the same with those in paragraph 3.3.1.

Turbine inputs

Turbine entry temperature TET^{od} varies between TET-500K and TET+100K with step 50K. (Actually this variation corresponds to part load or overload performance)

Turbine degradation, P_{tde} (%)=0

Actually, turbine "work" is power and is represented as TW^{od} in MW.

Turbine isentropic efficiency η_{istod}, will be calculated from the program

Turbine degradation, P_{tde} (%)=0

Exhaust inputs

The exhaust temperature T_{o8}^{od} will be calculated from the program. The rest parameters are the same with those in paragraph 3.3.1.

Calculation procedure

Initially, the calculation procedure for the selected DP is carried out.
Assessment of the compressor and power turbine's stages, in order the turbine is choked.

$$Rt_{hyp}=\frac{R_c \cdot (100 - DPcclossv)}{100 \cdot 1.003} \tag{3-106}$$

$$\text{IF } (1.73 \leq Rt_{hyp} \leq 2.99) \Rightarrow s=1 \tag{3-107}$$

$$\text{IF } (2.99 \leq Rt_{hyp} \leq 5.198) \Rightarrow s=2 \tag{3-108}$$

$$\text{IF } (5.198 \leq Rt_{hyp} \leq 8.96) \Rightarrow s=3 \tag{3-109}$$

$$\text{IF } (8.96 \leq Rt_{hyp} \leq 15.5) \Rightarrow s=4 \tag{3-110}$$

$$\text{IF } (15.5 \leq Rt_{hyp} \leq 26.8) \Rightarrow s=5 \tag{3-111}$$

$$\text{IF } (Rt_{hyp} \geq 26.8) \Rightarrow s=6 \tag{3-112}$$

$$Rt_{ch}=1.73^{s} \tag{3-113}$$

$$R_t = \frac{P_{o6}}{P_{o7}} \tag{3-114}$$

Then the OD calculation begins taking into account the results of the DP.

1-2 INTAKE

$$P_{o1}^{od} = P_{\alpha}^{od} \cdot (1 + \frac{\gamma_c - 1}{2} \cdot M_{in}^2)^{\frac{\gamma_c}{\gamma_c - 1}} \tag{3-115}$$

$M_{in} = 0$, due to the fact that the GT is stationary.

$$P_{o2}^{od} = P_{o1}^{od} \cdot (1 - \frac{DP_{in_{loss}}}{100}) \tag{3-116}$$

$$T_{o1}^{od} = T_{\alpha}^{od} \cdot (1 + \frac{\gamma_c - 1}{2} \cdot M_{in}^2) \tag{3-117}$$

$$T_{o2}^{od} = T_{o1}^{od} \tag{3-118}$$

2-3 COMPRESSOR

$$G_2^{od} = \frac{\dot{m} \cdot \sqrt{T_{o2}}}{P_{o2}} \cdot \left(\frac{T_{o2}}{T_{o2}^{od}} \right) \tag{3-119}$$

$$\dot{m}^{od} = \frac{G_2^{od} \cdot P_{o2}^{od}}{\sqrt{T_{o2}}} \tag{3-120}$$

1-2 INTAKE

$$\dot{m}_1^{od} = \dot{m}_2^{od} = \dot{m}^{od} \tag{3-121}$$

7-8 EXHAUST

$$P_{o8}^{od} = P_{o1}^{od} \cdot 1.003 \tag{3-122}$$

$$P_{o7}^{od} = \frac{P_{o8}^{od} \cdot 100}{100 - DP_{exh_{loss}}} \tag{3-123}$$

4-5 COMBUSTION CHAMBER (BURNER)

$$T_{o5}^{od} = TET^{od} \tag{3-124}$$

5-6 MIXER

$$T_{o6}^{od} = T_{o5}^{od} - DT_{cooling}^{od} \tag{3-125}$$

$$P_{o6}^{od} = \frac{\dot{m}^{od} \cdot P_{o6}}{\dot{m}} \cdot \left(\frac{\sqrt{T_{o6}^{od}}}{\sqrt{T_{o6}}} \right) \tag{3-126}$$

$$R_t^{od} = \frac{P_{o6}^{od}}{P_{o7}^{od}} \tag{3-127}$$

If TET = T_{o5} is less than 1,300 then it is assumed that there is no need for by pass mass flow to cool the turbine section, thus

$$D_{mc}=0,\ DT_{cooling}=0 \tag{3-128}$$

On the contrary, if T_{o5} is greater or equal to 1,300 then there is a portion of mass flow (D_{mc}), which reduces the TET per $DT_{cooling}$:

$$D_{mc}=0.025*T_{o5}-25,\ DT_{cooling}=0.333*T_{o5}-333.333 \tag{3-129}$$

$$P_{o5}^{od} = P_{o6}^{od} \tag{3-130}$$

4-5 COMBUSTION CHAMBER (BURNER)

$$P_{o4}^{od} = \frac{P_{o5}^{od}}{1 - \dfrac{DP_{cc_{loss}}}{100}} \tag{3-131}$$

$$\eta_{cc}^{od} = \eta_{cc} \cdot \left(1 - \frac{DP_{cc_{loss}}}{100} \right) \tag{3-132}$$

3-4 PREMASS

$$P_{o3}^{od} = P_{o4}^{od} \tag{3-133}$$

2-3 COMPRESSOR

$$R_c^{od} = \frac{P_{o3}^{od}}{P_{o2}^{od}} \tag{3-134}$$

$$\eta_{is_c}^{od} = \eta_{is_c} \cdot \left(1 - \left| \frac{P_{c_{de}}}{100} \right| \right) \cdot \sqrt[0.1]{\frac{T_{o6}^{od}}{T_{o2}^{od}} \cdot \frac{T_{o2}}{T_{o6}}}\ [16] \tag{3-135}$$

$$T_{o3}^{od} = T_{o2}^{od} \cdot \left(\frac{\left(R_c^{od}\right)^{(\gamma_c - 1)/\gamma_c} - 1}{\eta_{is_c}^{od}} + 1 \right) \tag{3-136}$$

$$\dot{m}_3^{od} = \dot{m}^{od} \tag{3-137}$$

3-4 PREMASS

$$\dot{m}_4^{od} = \dot{m}^{od} \cdot \left(1 - \frac{Dm_c^{od}}{100} \right) \tag{3-138}$$

$$T_{o4}^{od} = T_{o3}^{od} \tag{3-139}$$

4-5 COMBUSTION CHAMBER (BURNER)

$$Q_{cc}^{od} = \frac{\dot{m}_4^{od} \cdot \left(C_{p_h} \cdot T_{o5}^{od} - C_{p_c} \cdot T_{o4}^{od} \right)}{1{,}000{,}000} \tag{3-140}$$

$$\dot{m}_f^{od} = \frac{Q_{cc}^{od}}{FCV^{od} \cdot \eta_{cc}^{od}} \tag{3-141}$$

$$\dot{m}_5^{od} = \dot{m}_4^{od} + \dot{m}_f^{od} \tag{3-142}$$

$$FAR_{45}^{od} = \frac{\dot{m}_f^{od}}{\dot{m}_4^{od}} \tag{3-143}$$

5-6 MIXER

$$\dot{m}_6^{od} = \dot{m}^{od} + \dot{m}_f^{od} \tag{3-144}$$

6-7 COMPRESSOR & POWER TURBINE

$$\eta_{is_t}^{od} = \eta_{is_t} \cdot \left(1 - \left|\frac{P_{t_{de}}}{100}\right|\right) \cdot 0.1\sqrt{\frac{T_{o6}^{od}}{T_{o2}^{od}} \cdot \frac{T_{o2}}{T_{o6}}} \quad [16] \tag{3-145}$$

$$\dot{m}_7^{od} = \dot{m}_6^{od} \tag{3-146}$$

$$T_{o7}^{od} = T_{o6}^{od} \cdot \left[1 - \left[1 - \left(\frac{P_{o7}^{od}}{P_{o6}^{od}}\right)^{\gamma_h - 1/\gamma_h}\right] \cdot \eta_{is_t}^{od}\right] \tag{3-147}$$

7-8 EXHAUST

$$T_{o8}^{od} = T_{o7}^{od} \tag{3-148}$$

$$\dot{m}_8^{od} = \dot{m}_6^{od} \tag{3-149}$$

If R_t^{od} is less or equal to Rt_{ch} then, the turbine is unchoked so the results are unreliable.

On the contrary, if R_t^{od} is greater than Rt_{ch} then the performance calculation can continue.

PERFORMANCE

$$CW^{od} = \frac{\dot{m}^{od} \cdot C_{p_c} \cdot \left(T_{o3}^{od} - T_{o2}^{od}\right)}{1,000,000} \quad (MW) \tag{3-150}$$

$$TW^{od} = \frac{\dot{m}^{od} \cdot C_{p_h} \cdot \left(T_{o6}^{od} - T_{o7}^{od}\right)}{1,000,000} \quad (MW) \tag{3-151}$$

$$HI^{od} = Q_{cc}^{od} \quad (MW) \tag{3-152}$$

$$UW^{od} = TW^{od} - CW^{od} \quad (MW) \tag{3-153}$$

$$SW^{od} = \frac{UW^{od}}{\dot{m}^{od}} \quad (MJ/kgr) \tag{3-154}$$

$$\eta_{th}^{od} = \frac{UW^{od}}{HI^{od}} \tag{3-155}$$

$$sfc^{od} = \frac{1}{\eta_{th}^{od} \cdot FCV^{od}} \quad (kgr/(MW*sec)=kgr/MJ) \tag{3-156}$$

$$Q_{out}^{od} = \frac{\dot{m}_8^{od} \cdot C_{p_h} \cdot \left(T_{o8}^{od} - T_{\alpha}^{od}\right)}{1,000,000} \tag{3-157}$$

1. When ambient temperature T_α varies, ambient pressure P_α remains constant, and then in the equations above we substitute $P_\alpha^{od} = P_\alpha$.
2. When ambient pressure P_α varies, ambient temperature T_α remains constant, and then in the equations above we substitute $T_\alpha^{od} = T_\alpha$.
3. When altitude varies, then P_α^{od} and T_α^{od} remain as they are, but they change according to the following equations:

$$T_\alpha^{od} = 288.15 - 0.0065 \cdot A^{od} \tag{3-158}$$

$$P_\alpha^{od} = 101.325 - \left(\frac{288.15}{T_\alpha^{od}} \right)^{-5.25588} \tag{3-159}$$

The results of the above calculation procedure for the three variables T_α, P_α and altitude -using the FORTRAN program developed by the author- are represented in the following *Figs*.

The above calculation method is simplified but realistic. If required a more detailed model it can easily replace this one, due to module-construction of the overall simulation program developed by the author.

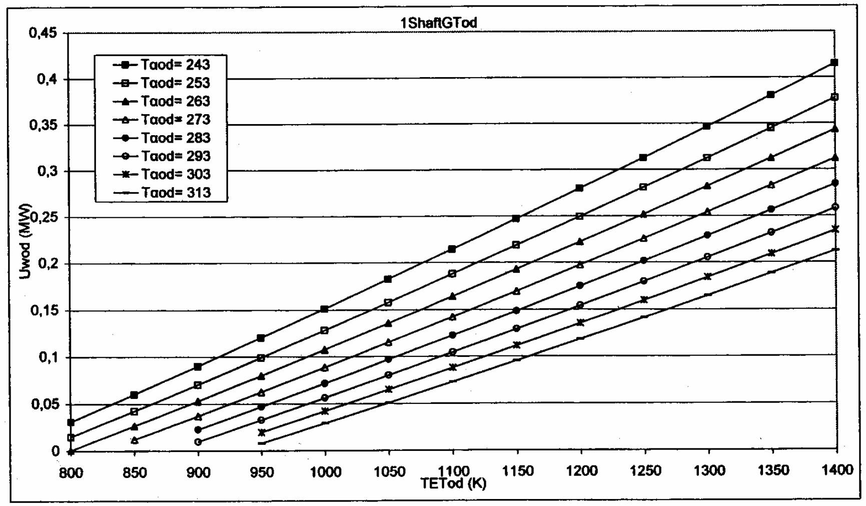

Fig. 3.17: 1Shaft GT, UW vs TET (parameter Tα)

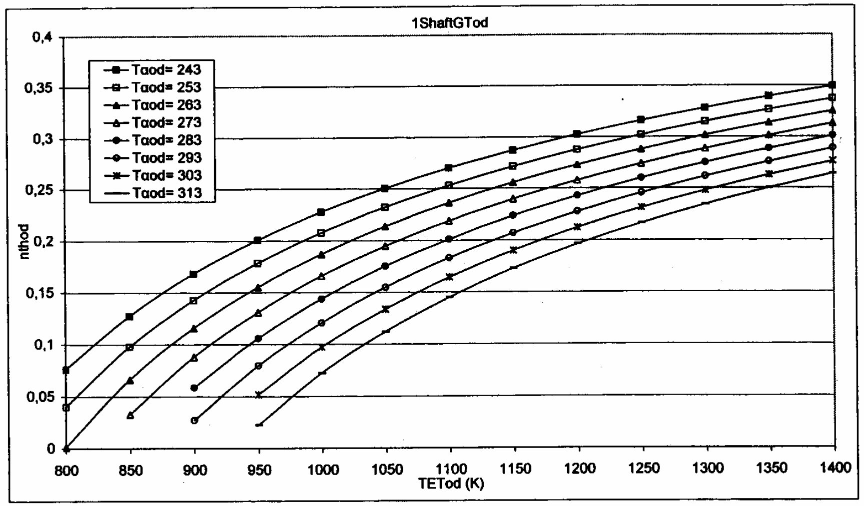

Fig. 3.18: 1Shaft GT, η_{th} vs TET (parameter Tα)

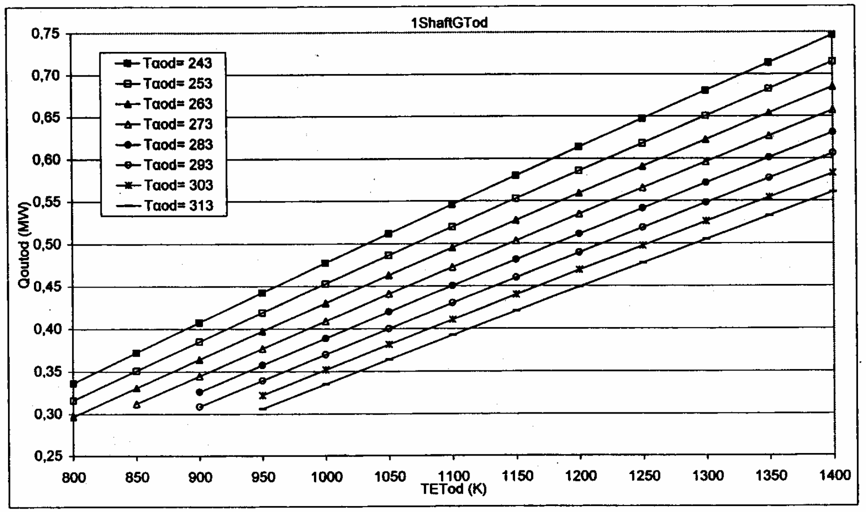

Fig. 3.19: 1Shaft GT, Qout vs TET (parameter Tα)

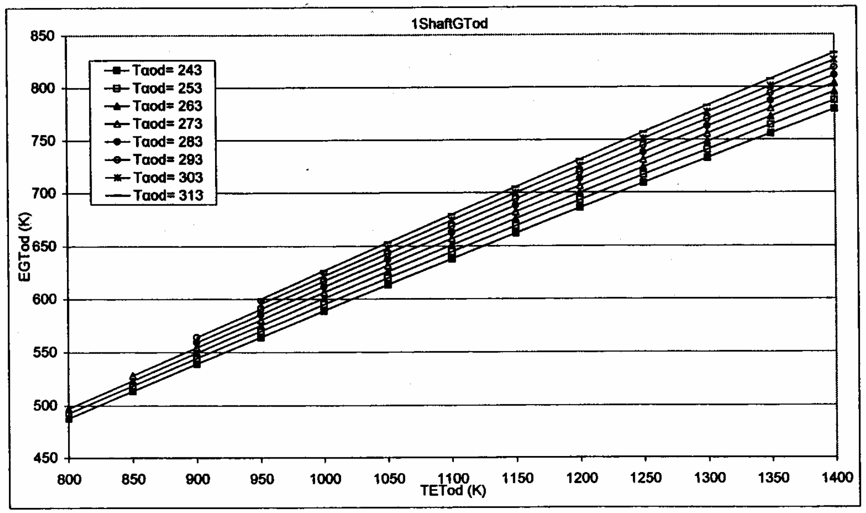

Fig. 3.20: 1Shaft GT, EGT vs TET (parameter Tα)

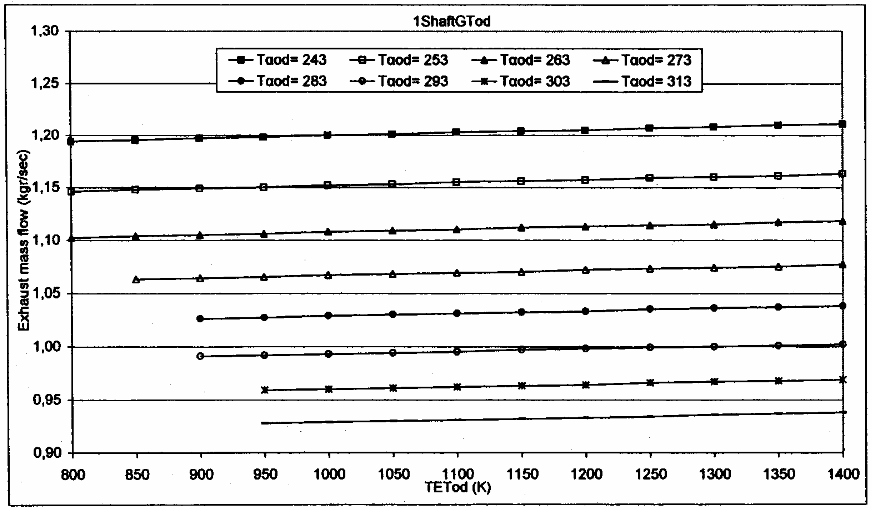

Fig. 3.21: 1Shaft GT, Exhaust mass flow vs TET (parameter Tα)

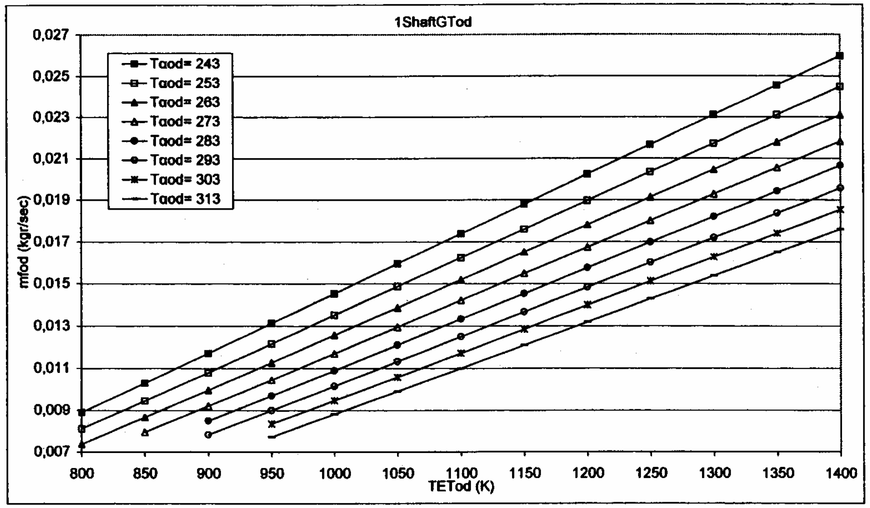

Fig. 3.22: 1Shaft GT, m_f vs TET (parameter Tα)

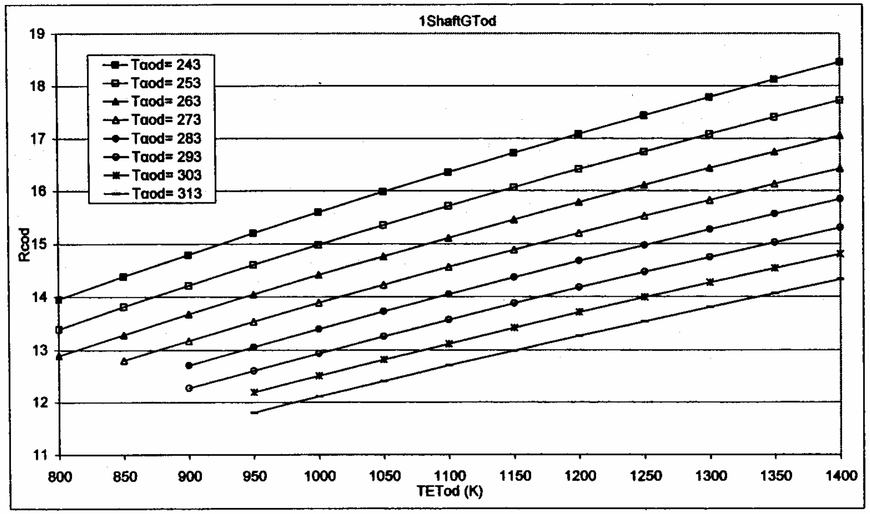

Fig. 3.23: 1Shaft GT, R_c vs TET (parameter Tα)

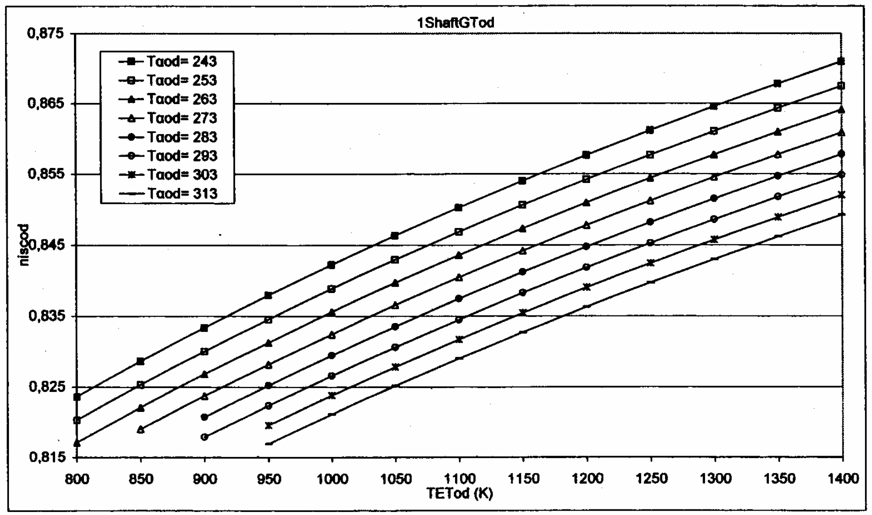

Fig. 3.24: 1Shaft GT, η_{isc} vs TET (parameter Tα)

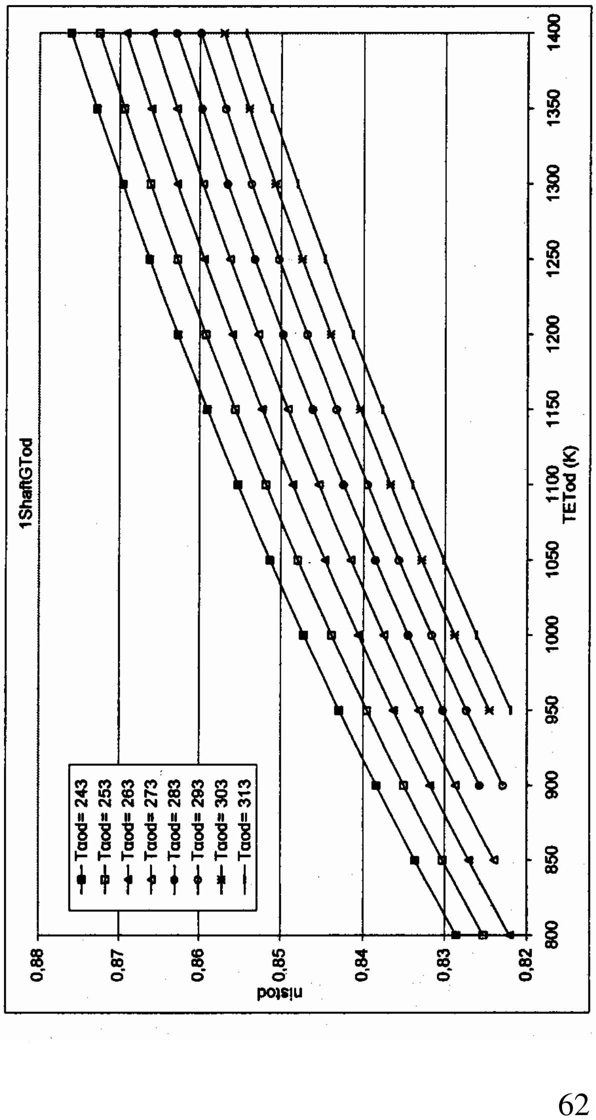

Fig. 3.25: 1Shaft GT, η_{ist} vs TET (parameter $T\alpha$)

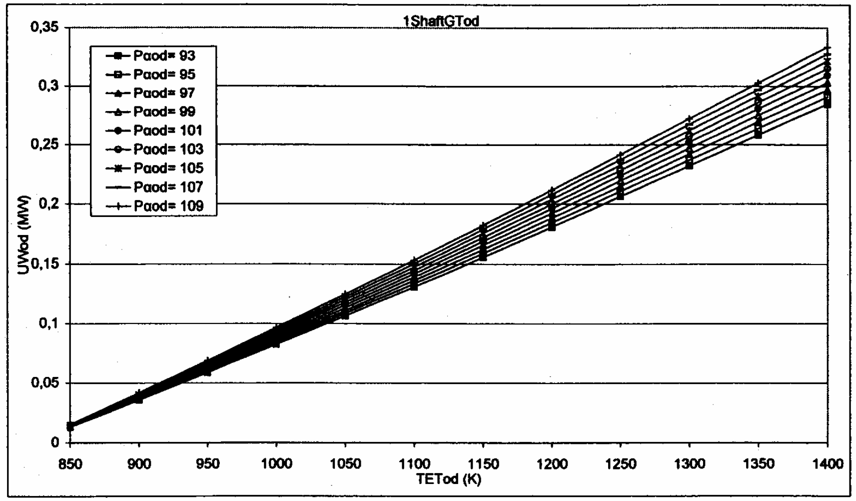

Fig. 3.26: 1Shaft GT, UW vs TET (parameter Pα)

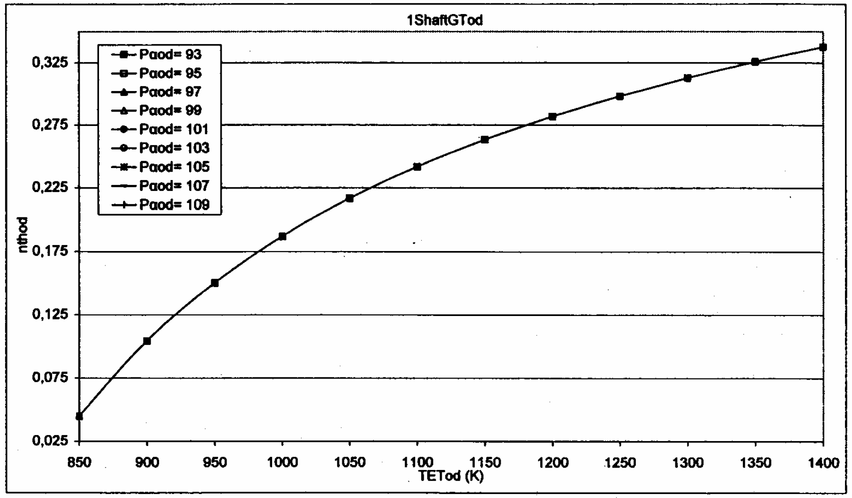

Fig. 3.27: 1Shaft GT, η_{th} vs TET (parameter Pα)

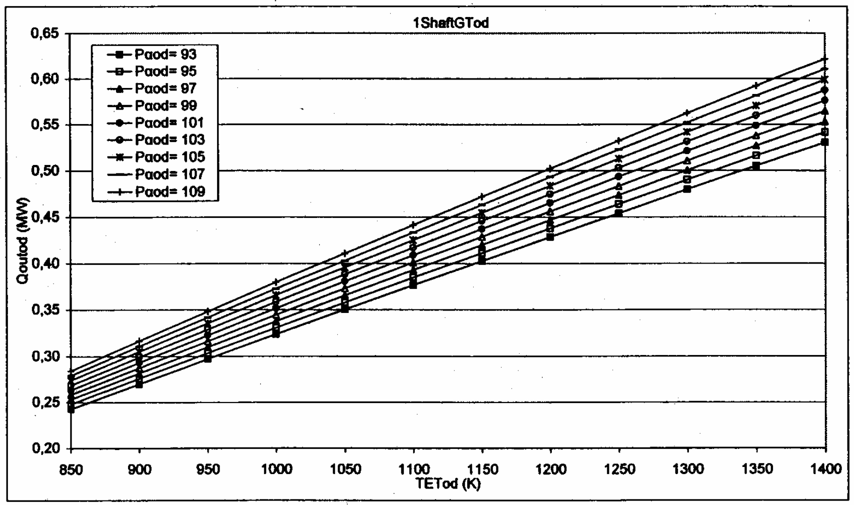

Fig. 3.28: 1Shaft GT, Qout vs TET (parameter Pα)

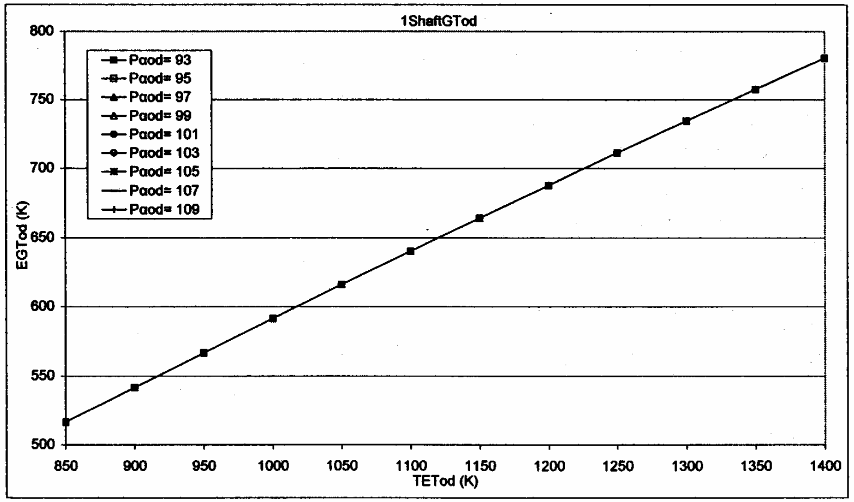

Fig. 3.29: 1Shaft GT, EGT vs TET (parameter Pα)

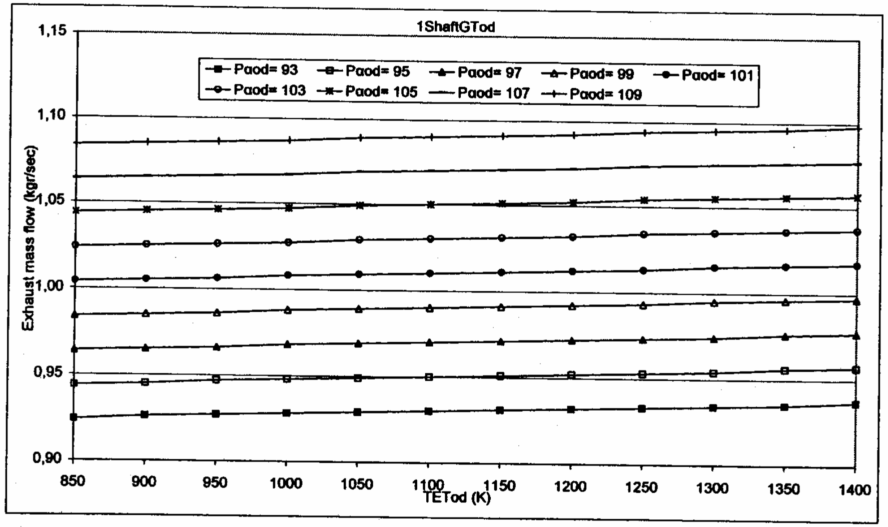

Fig. 3.30: 1Shaft GT, Exhaust mass flow vs TET (parameter Pα)

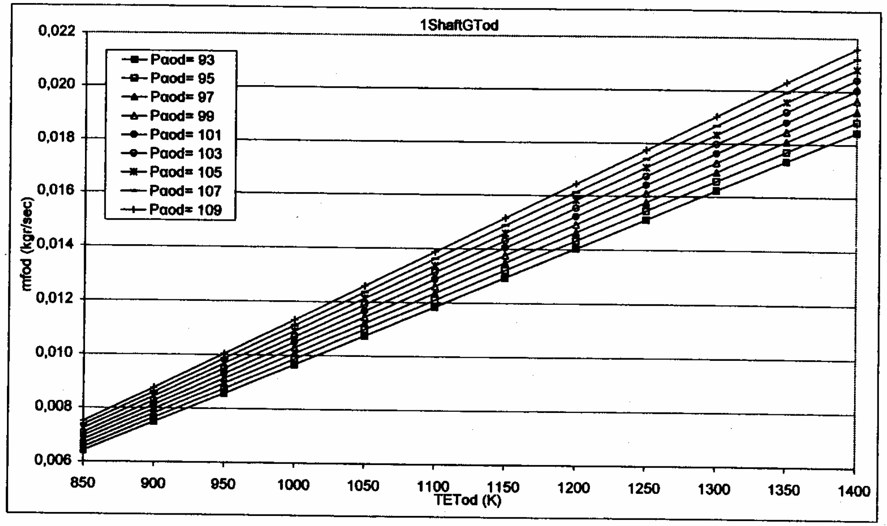

Fig. 3.31: 1Shaft GT, m_f vs TET (parameter Pα)

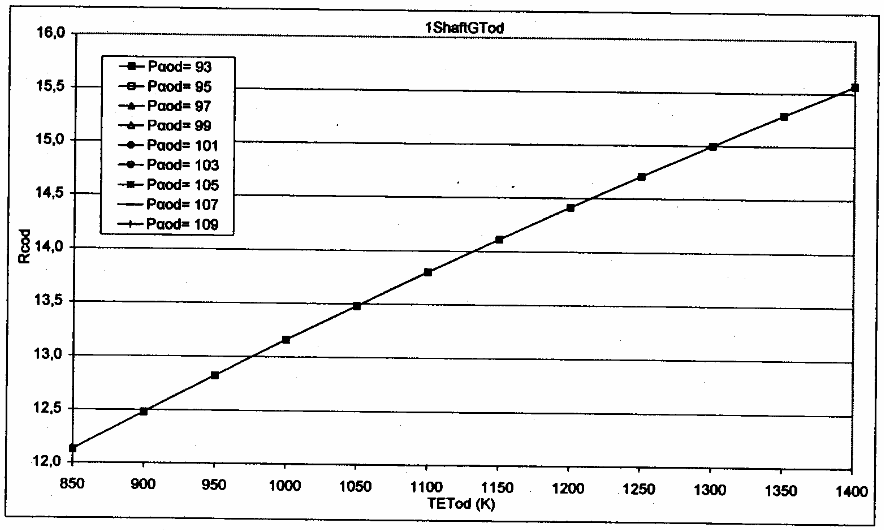

Fig. 3.32: 1Shaft GT, R_c vs TET (parameter Pα)

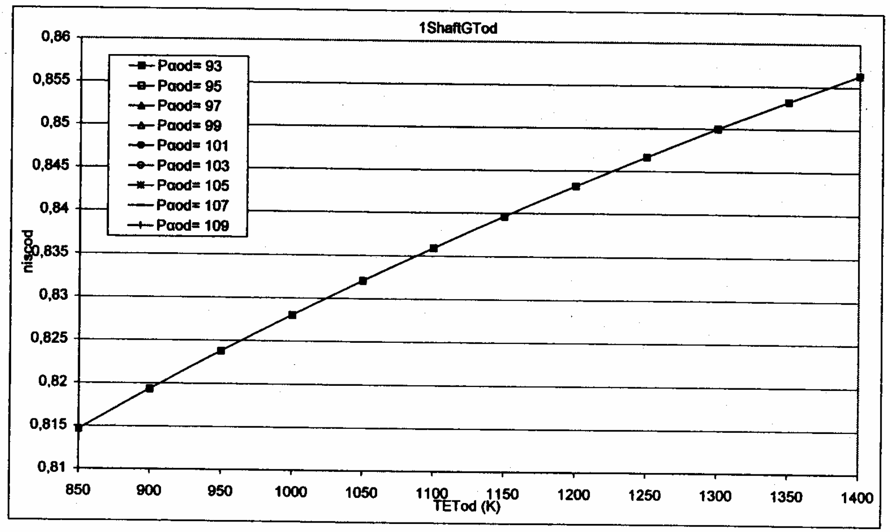

Fig. 3.33: 1Shaft GT, η_{isc} vs TET (parameter Pα)

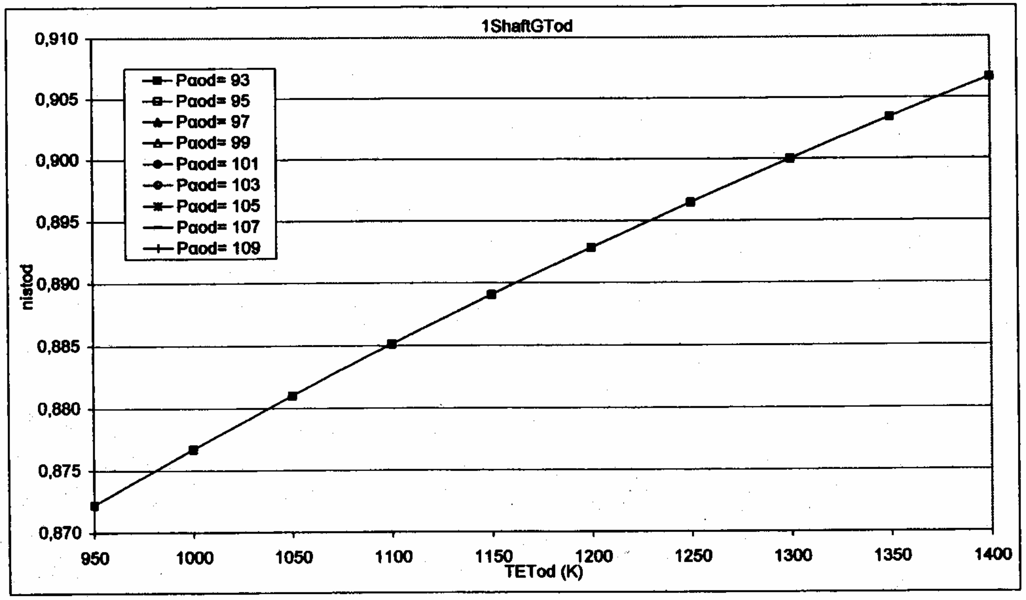

Fig. 3.34: 1Shaft GT, η_{ist} vs TET (parameter Pα)

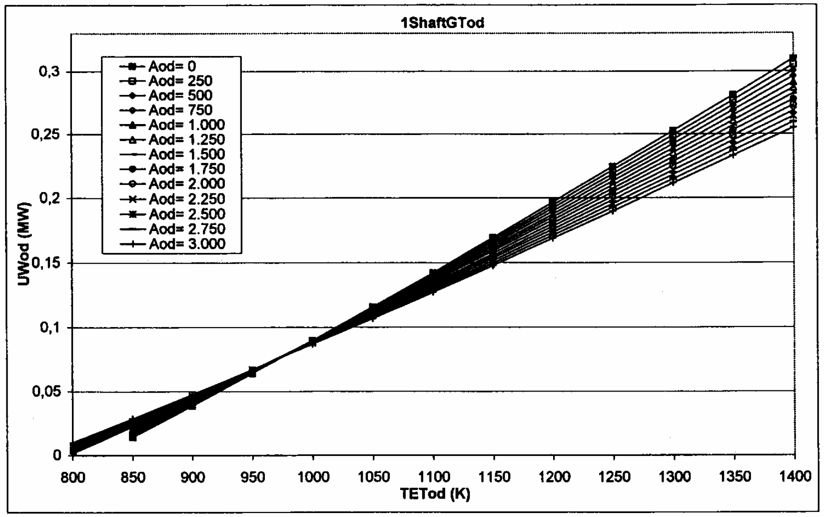

Fig. 3.35: 1Shaft GT, UW vs TET (parameter A)

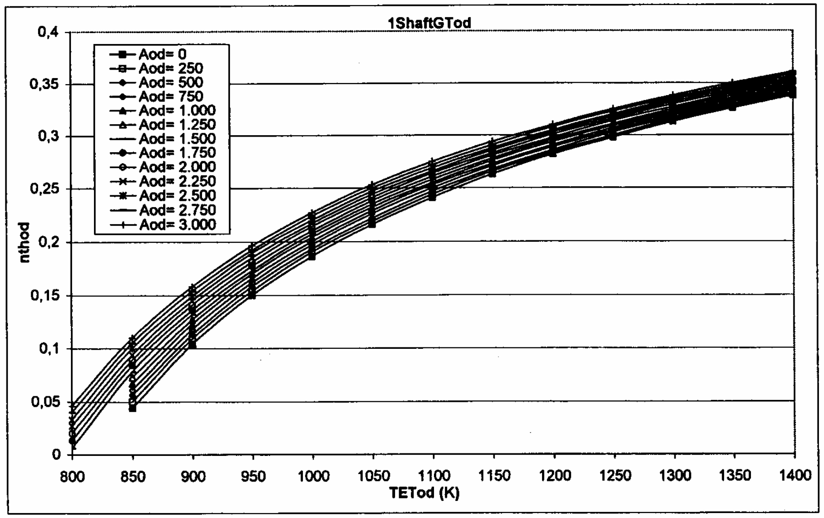

Fig. 3.36: 1Shaft GT, η_{th} vs TET (parameter A)

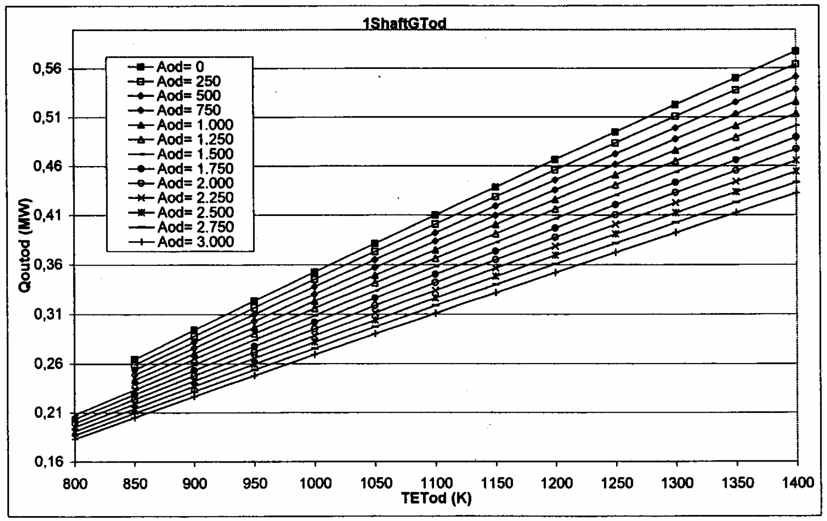

Fig. 3.37: 1Shaft GT, Qout vs TET (parameter A)

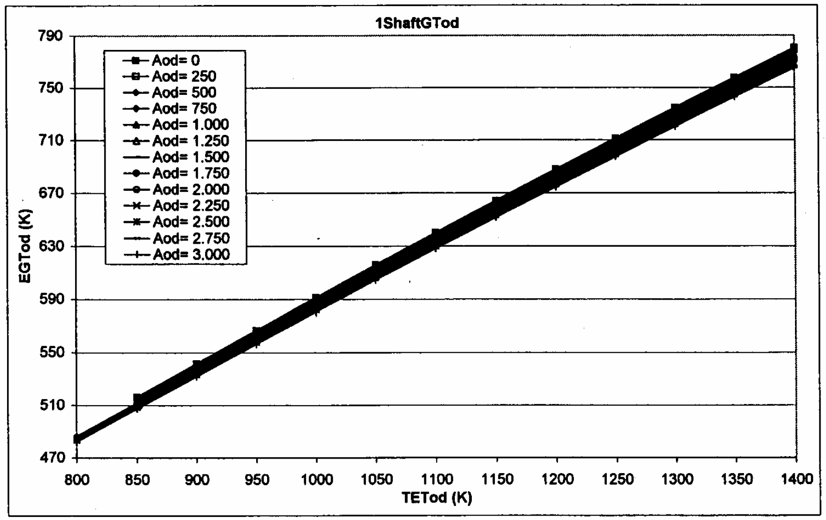

Fig. 3.38: 1Shaft GT, EGT vs TET (parameter A)

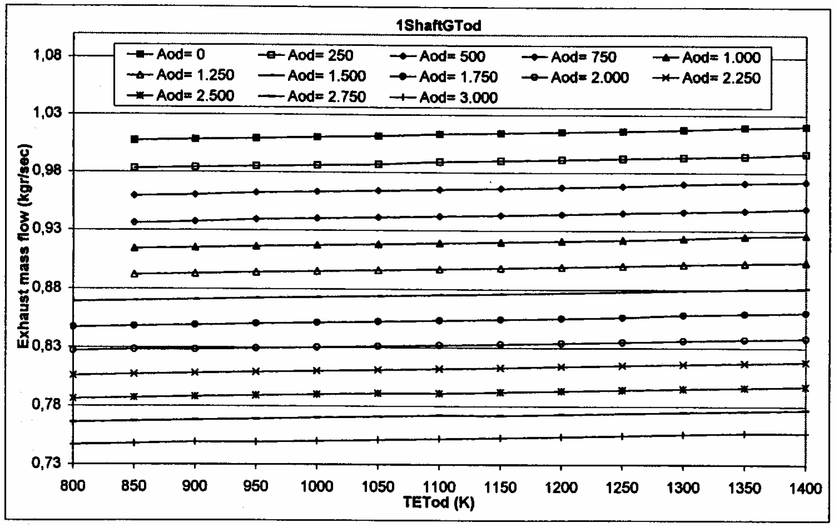

Fig. 3.39: 1Shaft GT, Exhaust mass flow vs TET (parameter A)

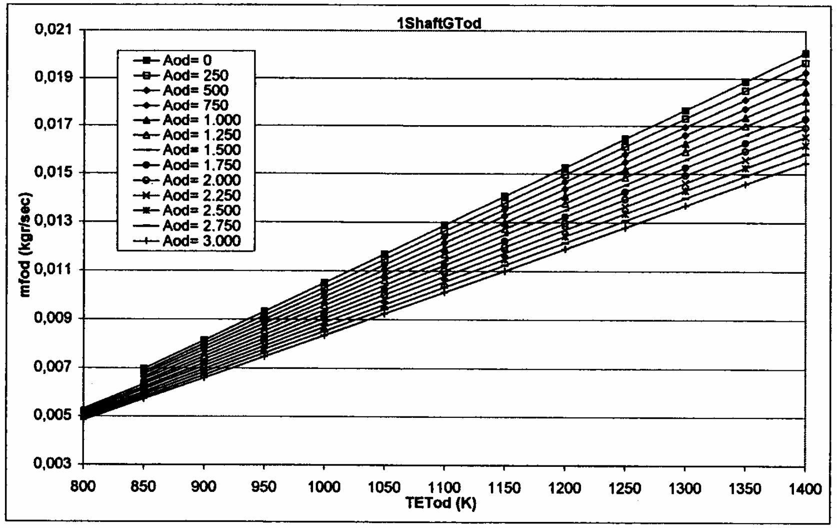

Fig. 3.40: 1Shaft GT, m_f vs TET (parameter A)

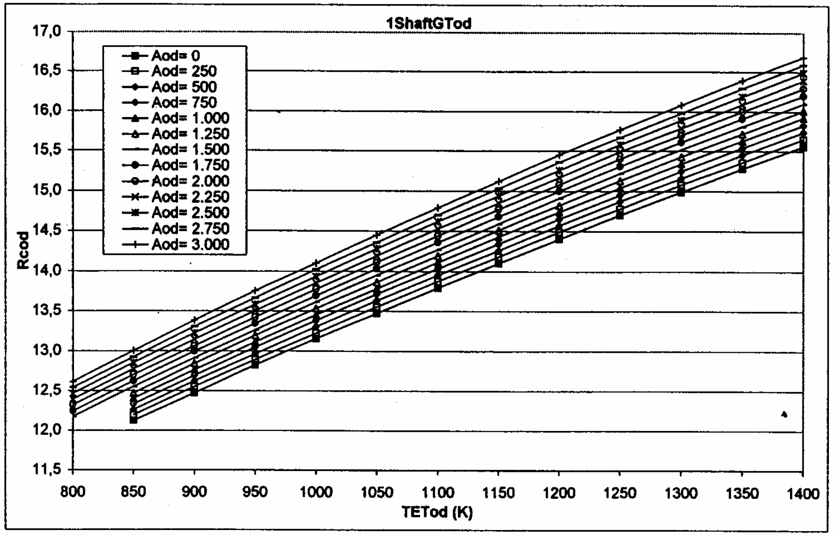

Fig. 3.41: 1Shaft GT, R_c vs TET (parameter A)

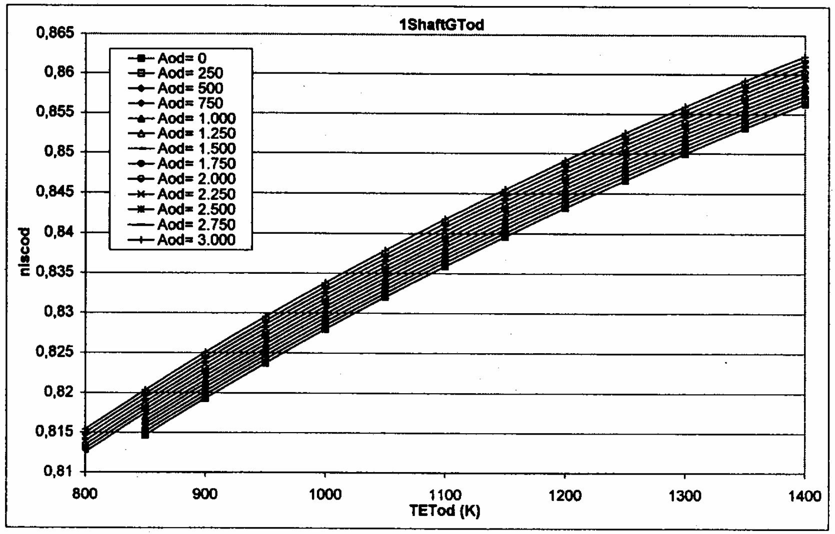

Fig. 3.42: 1Shaft GT, η_{isc} vs TET (parameter A)

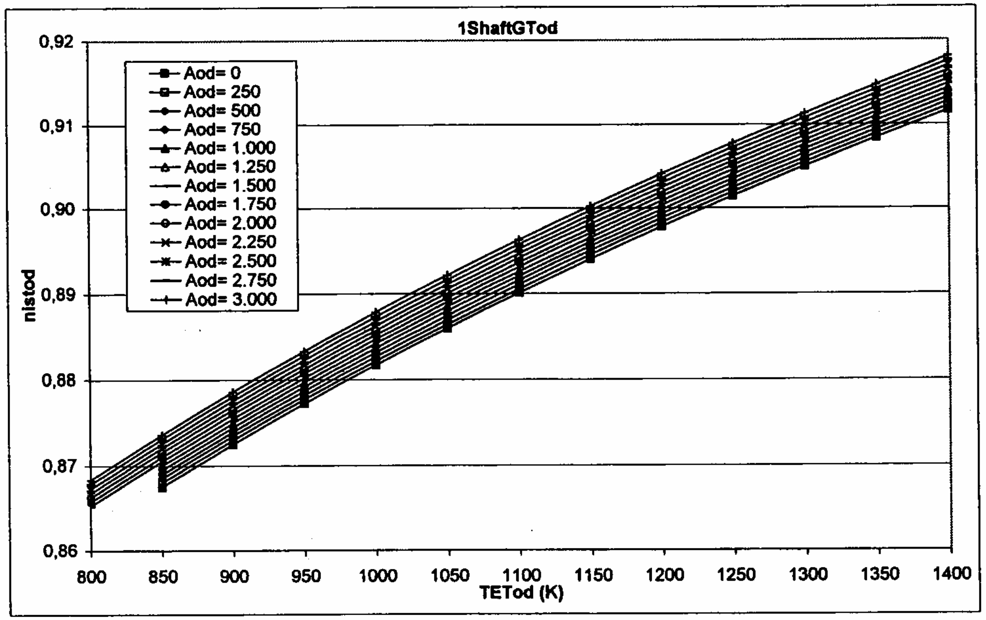

Fig. 3.43: 1Shaft GT, η_{ist} vs TET (parameter A)

3.6.2 2-Shaft Simple Cycle Simulation Procedure

Referring to *Fig. 3.3*.

<u>Input File (APPENDIX C.8)</u>

Selection of the engine

In order to test the correct response of the off-design simulation program, we choose an engine with the following parameters:

3. TET=1,300K
4. R_C=15

Variable Ambient Conditions

Same as in paragraph 3.6.1

Working fluid inputs

Same as in paragraph 3.6.1

Intake inputs

Is the same with those in paragraph 3.3.1.

Compressor inputs

Same as in paragraph 3.6.1

Premass inputs

Is the same with those in paragraph 3.3.1.

Combustion Chamber inputs

Same as in paragraph 3.6.1

Mixer inputs

Is the same with those in paragraph 3.3.1.

Compressor Turbine inputs

Turbine entry temperature TET^{od} varies between TET-600K and TET+100K with step 50K. (Actually this variation corresponds to part load or overload performance)

The rest are as in paragraph 3.3.1 and 3.3.2

Power Turbine

It is the same with those in paragraph 3.3.1 and 3.3.2

Exhaust inputs

It is the same with those in paragraph 3.6.1.

Calculation procedure

Initially, the calculation procedure for the selected DP is carried out.
Assessment of the compressor turbine and power turbine's stages, in order the turbine is choked.

$$R_{t_1} = \frac{P_{o6}}{P_{o7}} \tag{3-160}$$

$$\text{IF } (1.73 \le R_{t_1} \le 2.99) \Rightarrow s_1=1 \tag{3-161}$$

$$\text{IF } (2.99 \le R_{t_1} \le 5.198) \Rightarrow s_1=2 \tag{3-162}$$

$$\text{IF } (5.198 \le R_{t_1} \le 8.96) \Rightarrow s_1=3 \tag{3-163}$$

$$\text{IF } (8.96 \le R_{t_1} \le 15.5) \Rightarrow s_1=4 \tag{3-164}$$

$$\text{IF } (15.5 \le R_{t_1} \le 26.8) \Rightarrow s_1=5 \tag{3-165}$$

$$\text{IF } (R_{t_1} \ge 26.8) \Rightarrow s_1=6 \tag{3-166}$$

$$R_{t_2} = \frac{P_{o7}}{P_{o8}} \tag{3-167}$$

$$\text{IF } (1.73 \le R_{t_2} \le 2.99) \Rightarrow s_2=1 \tag{3-168}$$

$$\text{IF } (2.99 \le R_{t_2} \le 5.198) \Rightarrow s_2=2 \tag{3-169}$$

$$\text{IF } (5.198 \le R_{t_2} \le 8.96) \Rightarrow s_2=3 \tag{3-170}$$

$$\text{IF } (8.96 \le R_{t_2} \le 15.5) \Rightarrow s_2=4 \tag{3-171}$$

$$\text{IF } (15.5 \le R_{t_2} \le 26.8) \Rightarrow s_2=5 \tag{3-172}$$

$$\text{IF } (R_{t_2} \ge 26.8) \Rightarrow s_2=6 \tag{3-173}$$

$$Rt_{ch_1} = 1.73^{S1} \tag{3-174}$$

$$Rt_{ch_2} = 1.73^{S2} \tag{3-175}$$

Then the OD calculation begins taking into account the results of the DP.

1-2 INTAKE
Equations (3-115) - (3-118)
4-5 COMBUSTION CHAMBER (BURNER)
Equation (3-124)
5-6 MIXER
Equation (3-125)
6-7 COMPRESSOR TURBINE

$$\eta_{pol_{Ct}}^{od} = \eta_{pol_{Ct}} \cdot \left(1 - \left|\frac{P_{t_{de}}}{100}\right|\right) \cdot 0.1\sqrt{\frac{T_{o6}^{od}}{T_{o2}^{od}} \cdot \frac{T_{o2}}{T_{o6}}} \tag{3-176}$$

$$e = \frac{\gamma_h}{\eta_{pol_{Ct}}^{od} \cdot (\gamma_h - 1)}$$

$$T_{o7}^{od} = T_{o6}^{od} \cdot \left(\frac{\dot{m}_6 \cdot \sqrt{T_{o6}} \cdot P_{o7}}{\dot{m}_7 \cdot \sqrt{T_{o7}} \cdot P_{o6}} \right)^{\frac{2}{2 \cdot e - 1}} \tag{3-177}$$

2-3 COMPRESSOR

$$T_{o3}^{od} = \frac{C_{p_h} \cdot \left(T_{o6}^{od} - T_{o7}^{od} \right) + C_{p_c} \cdot T_{o2}^{od}}{C_{p_c}} \tag{3-178}$$

Equation (3-139), (3-135)

$$R_c^{od} = \left(\frac{\eta_{is_c}^{od} \cdot (T_{o3}^{od} - T_{o2}^{od})}{T_{o2}^{od}} + 1 \right)^{\frac{\gamma_c}{\gamma_c - 1}} \tag{3-179}$$

3-4 PREMASS

$$P_{o3}^{od} = R_c^{od} \cdot P_{o2}^{od} \tag{3-180}$$

$$P_{o4}^{od} = P_{o3}^{od} \tag{3-181}$$

4-5 COMBUSTION CHAMBER (BURNER)

$$P_{o5}^{od} = P_{o4}^{od} \cdot \left(\frac{1 - DP_{cc_{loss}}}{100} \right) \tag{3-182}$$

5-6 MIXER

$$P_{o6}^{od} = P_{o5}^{od} \tag{3-183}$$

6-7 COMPRESSOR TURBINE

$$P_{o7}^{od} = \frac{P_{o6}^{od}}{\left(\frac{T_{o6}^{od}}{T_{o7}^{od}} \right)^{\frac{\gamma_h}{(\gamma_h - 1) \cdot \eta_{pol_{Ct}}^{od}}}} \tag{3-184}$$

8-9 EXHAUST

$$P_{o9}^{od} = P_{o1}^{od} \cdot 1.003 \tag{3-185}$$

$$P_{o8}^{od} = \frac{P_{o9}^{od} \cdot 100}{100 - DP_{exh_{loss}}} \tag{3-186}$$

6-7 COMPRESSOR TURBINE

$$\dot{m}_6^{od} = \frac{P_{o6}^{od}}{\sqrt{T_{o6}^{od}}} \cdot \frac{\dot{m}_6 \cdot \sqrt{T_{o6}}}{P_{o6}} \tag{3-187}$$

4-5 COMBUSTION CHAMBER (BURNER)

Equation (3-132)

$$\dot{m}_{o5}^{od} = \dot{m}_{o6}^{od} \tag{3-188}$$

Equations (3-140), (3-141), (3-143).

5-6 MIXER

$$\dot{m}^{od} = \dot{m}_{o5}^{od} - \dot{m}_f^{od} \tag{3-189}$$

1-2 INTAKE
Equation (3-121)
2-3 COMPRESSOR
Equation (3-137)
3-4 PREMASS
Equation (3-138)
6-7 COMPRESSOR TURBINE
Equation (3-146)

$$R_{t_1}^{od} = \frac{P_{o6}^{od}}{P_{o7}^{od}} \tag{3-190}$$

$$\eta_{is_{Ct}}^{od} = \frac{\left[1 - \left(\frac{P_{o7}^{od}}{P_{o6}^{od}}\right)^{\frac{\eta_{pol_{Ct}}^{od} \cdot (\gamma_h - 1)}{\gamma_h}}\right]}{\left[1 - \left(\frac{P_{o7}^{od}}{P_{o6}^{od}}\right)^{\frac{\gamma_h - 1}{\gamma_h}}\right]} \tag{3-191}$$

7-8 POWER TURBINE

$$R_{t_2}^{od} = \frac{P_{o7}^{od}}{P_{o8}^{od}} \tag{3-192}$$

$$n_{is_{Pt}}^{od} = n_{is_{Pt}} \cdot \left(1 - \left|\frac{P_{t_{de}}}{100}\right|\right) \cdot 0{,}1 \sqrt{\frac{T_{o6}^{od}}{T_{o2}^{od}} \cdot \frac{T_{o2}}{T_{o6}}} \tag{3-193}$$

$$T_{o8}^{od} = T_{o7}^{od} \cdot \left[1 - \eta_{is_{Pt}}^{od} \cdot \left(1 - \left(\frac{P_{o8}^{od}}{P_{o7}^{od}}\right)^{\frac{\gamma_h - 1}{\gamma_h}}\right)\right] \tag{3-194}$$

Equation (3-149)
8-9 EXHAUST

$$T_{o9}^{od} = T_{o8}^{od} \tag{3-195}$$

$$\dot{m}_9^{od} = \dot{m}_6^{od} \tag{3-196}$$

If $R_{t_1}^{od}$ is less or equal to Rt_{ch_1} then, the turbine is unchoked, so the results are unreliable, else performance calculation can continue.
If $R_{t_2}^{od}$ is less or equal to Rt_{ch_2} then, the turbine is unchoked, so the results are unreliable, else performance calculation can continue.
Equation (3-150)

PERFORMANCE

$$PTW^{od} = \frac{\dot{m}_7^{od} \cdot C_{p_h} \cdot \left(T_{o7}^{od} - T_{o8}^{od}\right)}{1.000.000} \quad \text{(MW)} \qquad (3\text{-}197)$$

$$UW^{od} = PTW^{od} \quad \text{(MW)} \qquad (3\text{-}198)$$

Equations (3-152), (3-154) - (3-156)

$$Q_{out}^{od} = \frac{\dot{m}_9^{od} \cdot C_{p_h} \cdot \left(T_{o9}^{od} - T_{\alpha}^{od}\right)}{1.000.000} \quad \text{(MW)} \qquad (3\text{-}199)$$

1. When ambient temperature T_α varies, ambient pressure P_α remains constant, and then in the equations above we substitute $P_\alpha^{od} = P_\alpha$.
2. When ambient pressure P_α varies, ambient temperature T_α remains constant, and then in the equations above we substitute $T_\alpha^{od} = T_\alpha$.
3. When altitude varies, then P_α^{od} and T_α^{od} remain as they are, but they change according to the equations: (3-158), (3-159)

The results of the above calculation procedure for the three variables T_α, P_α and altitude -using the FORTRAN program developed by the author- are represented in the following *Figs*.

The above calculation method is simplified but realistic. If required a more detailed model it can easily replace this one, due to module-construction of the overall simulation program developed by the author.

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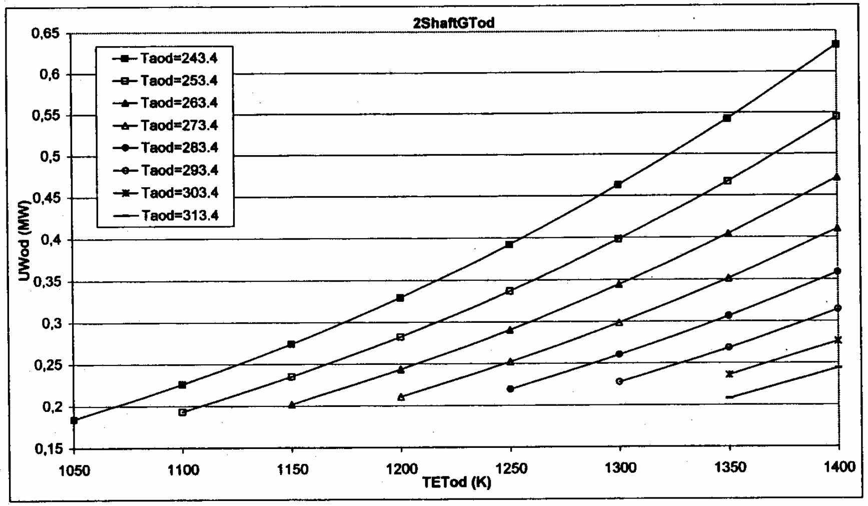


Fig. 3.44: 2Shaft GT, UW vs TET (parameter Tα)

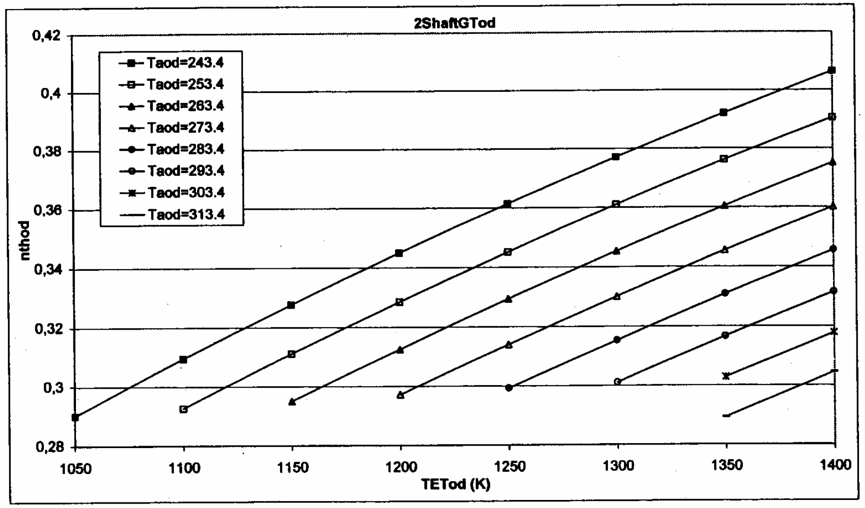


Fig. 3.45: 2Shaft GT, η_{th} vs TET (parameter Tα)

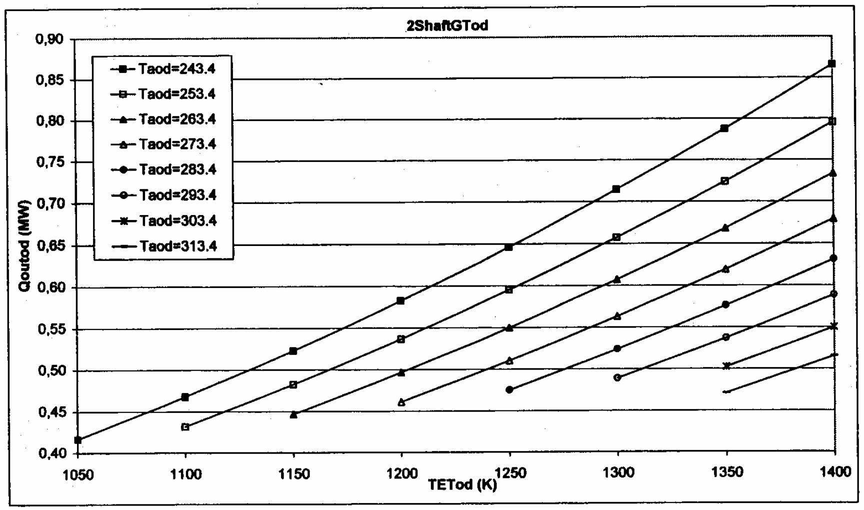


Fig. 3.46: 2Shaft GT, Qout vs TET (parameter Tα)

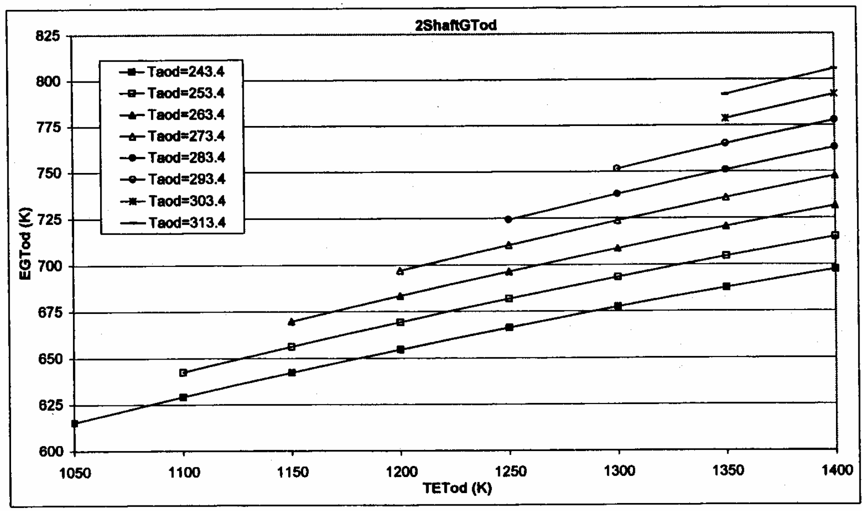


Fig. 3.47: 2Shaft GT, EGT vs TET (parameter Tα)

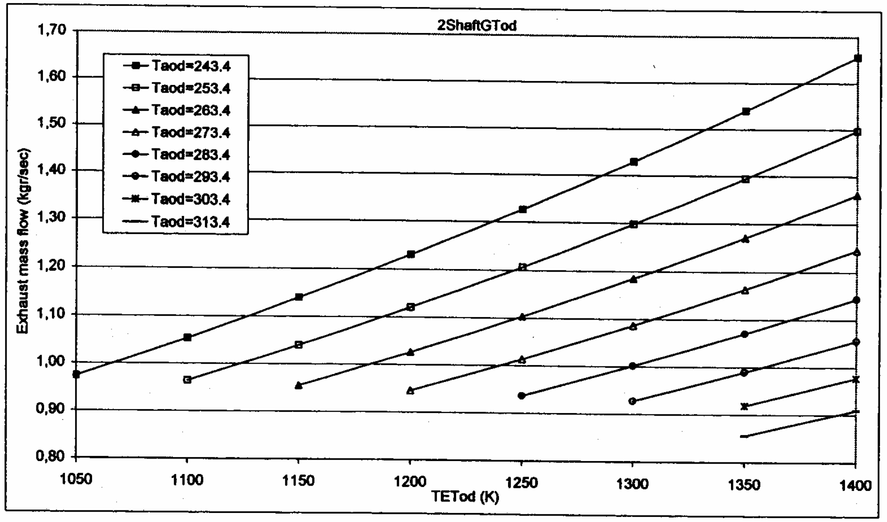


Fig. 3.48: 2Shaft GT, Exhaust mass flow vs TET (parameter Tα)

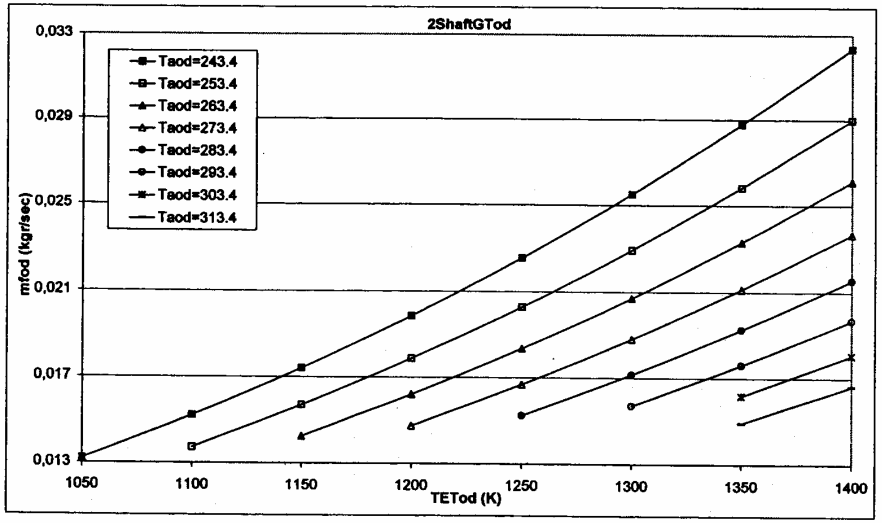


Fig. 3.49: 2Shaft GT, mf vs TET (parameter Tα)

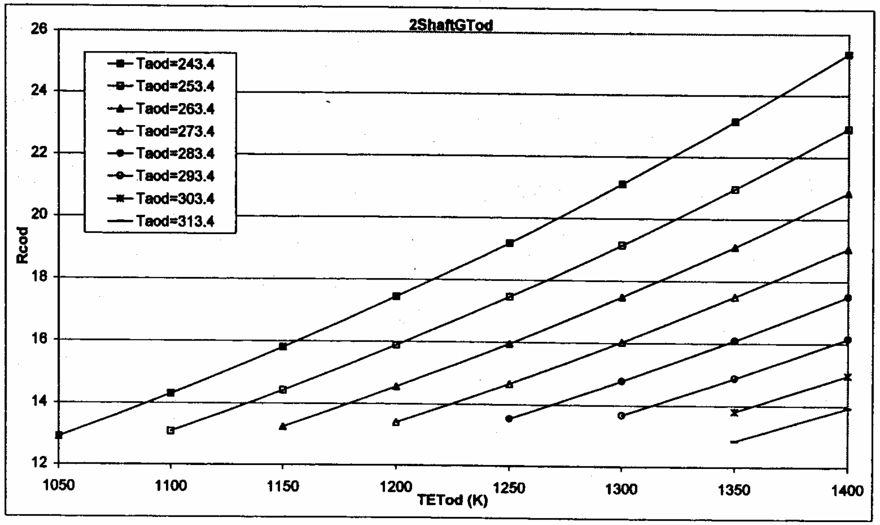


Fig. 3.50: 2Shaft GT, R_c vs TET (parameter Tα)

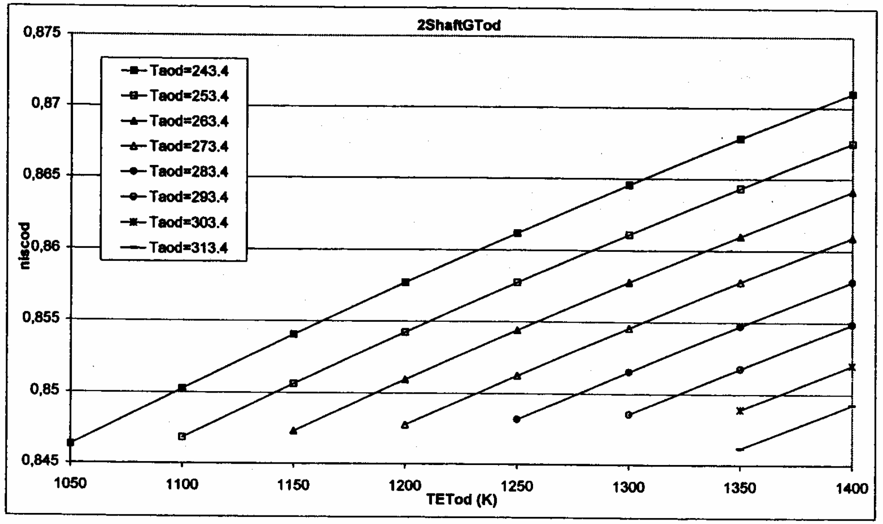


Fig. 3.51: 2Shaft GT, η_{isc} vs TET (parameter Tα)

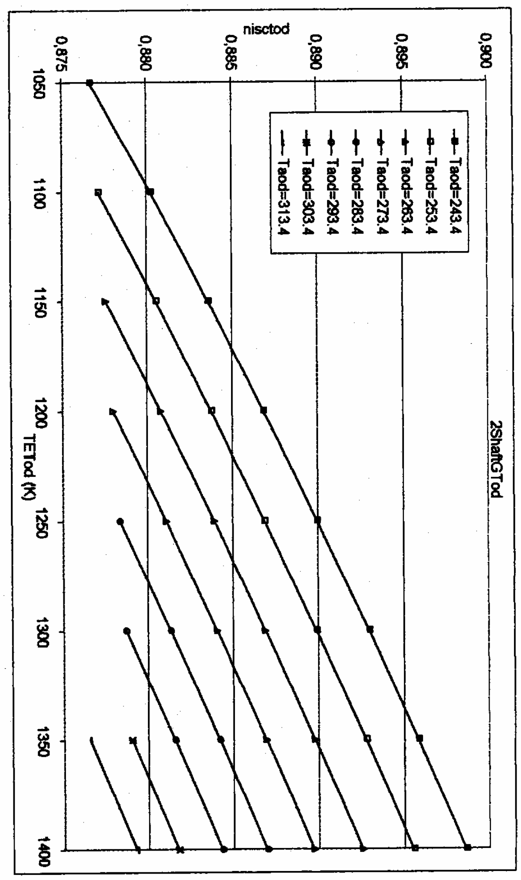


Fig. 3.52: 2Shaft GT, η_{isct} vs TET (parameter $T\alpha$)

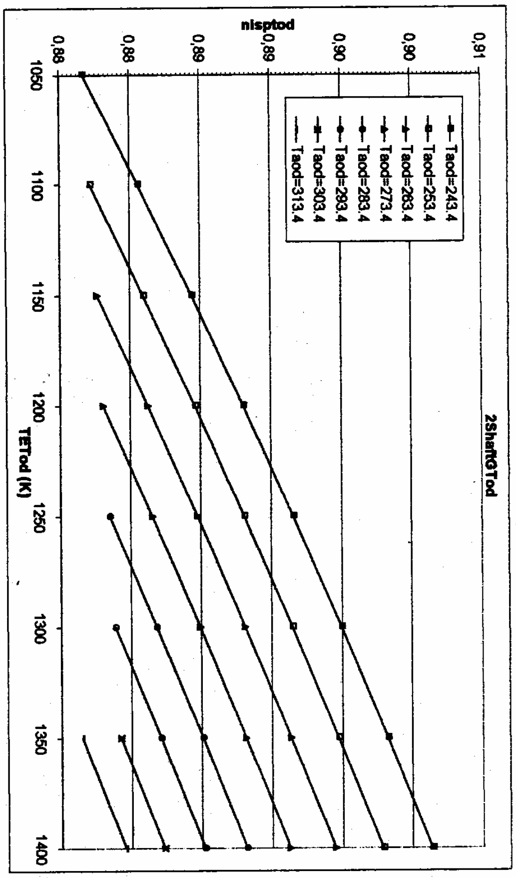


Fig. 3.53: 2Shaft GT, η_{ispt} vs TET (parameter $T\alpha$)

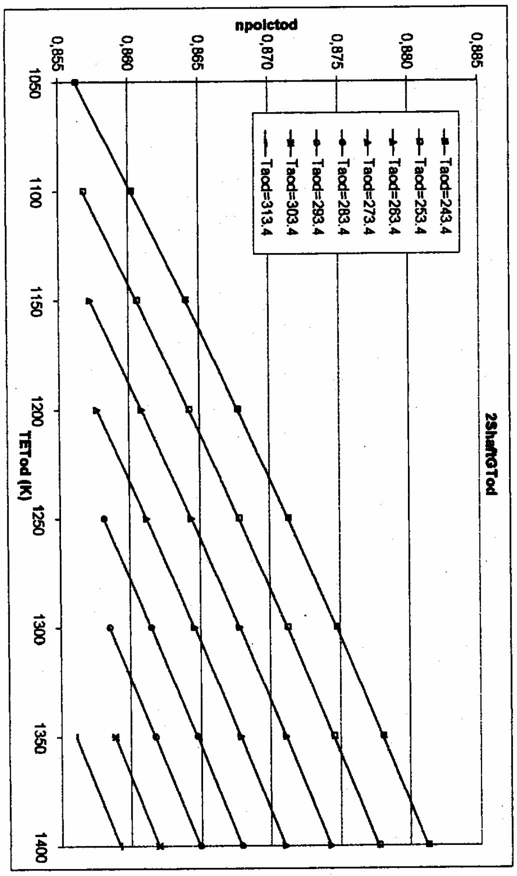


Fig. 3.54: 2Shaft GT, η_{polct} vs TET (parameter $T\alpha$)

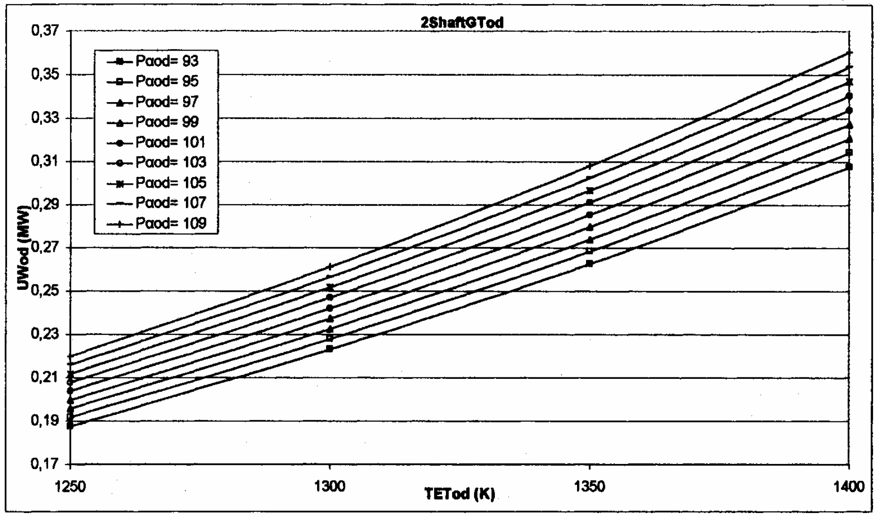


Fig. 3.55: 2Shaft GT, UW vs TET (parameter Pα)

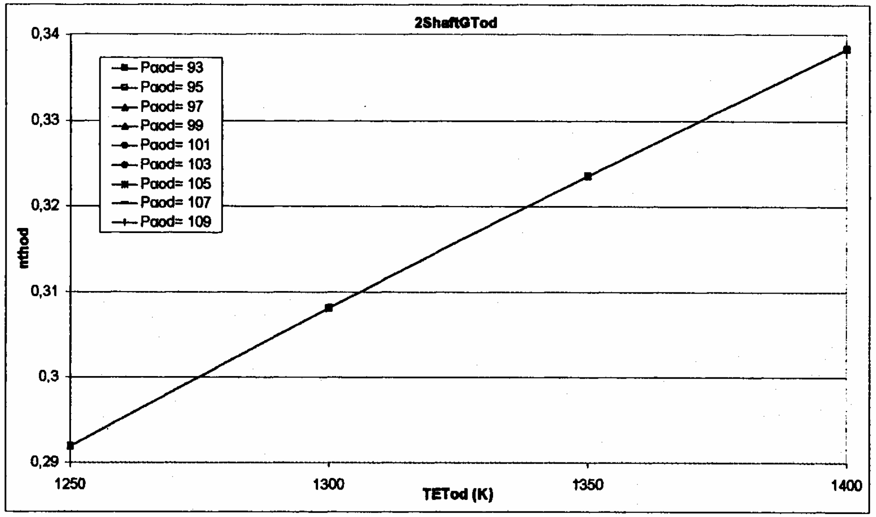


Fig. 3.56: 2Shaft GT, η_{th} vs TET (parameter Pα)

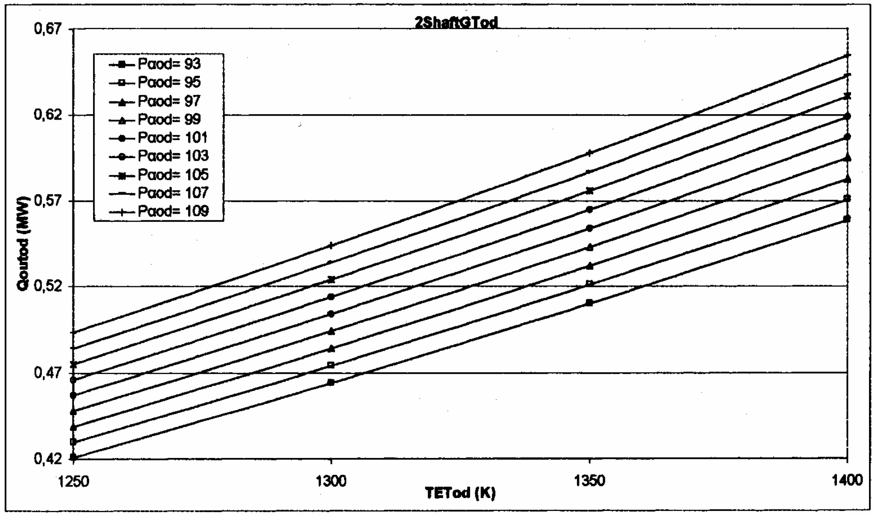


Fig. 3.57: 2Shaft GT, Qout vs TET (parameter Pα)

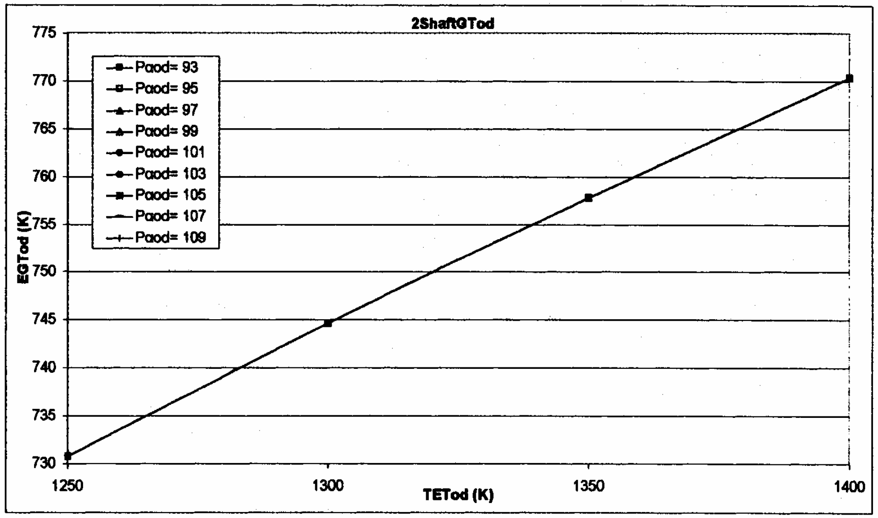


Fig. 3.58: 2Shaft GT, EGT vs TET (parameter Pα)

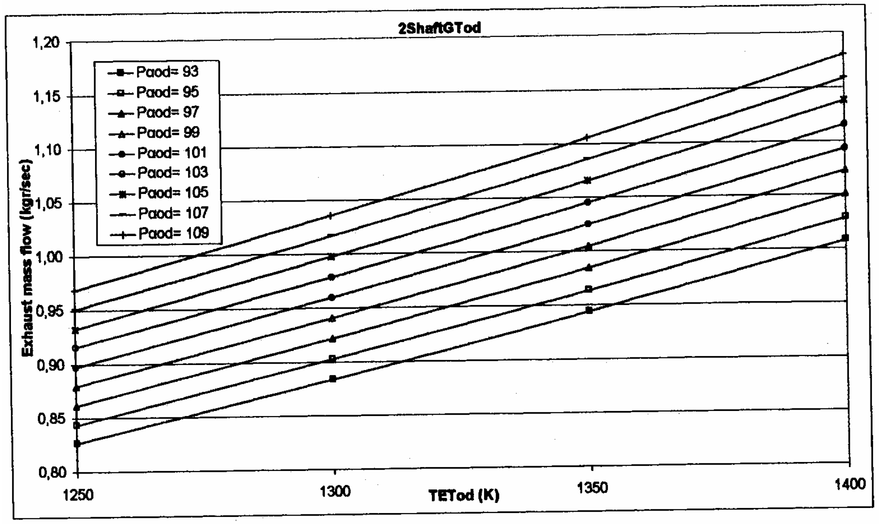


Fig. 3.59: 2Shaft GT, Exhaust mass flow vs TET (parameter Pα)

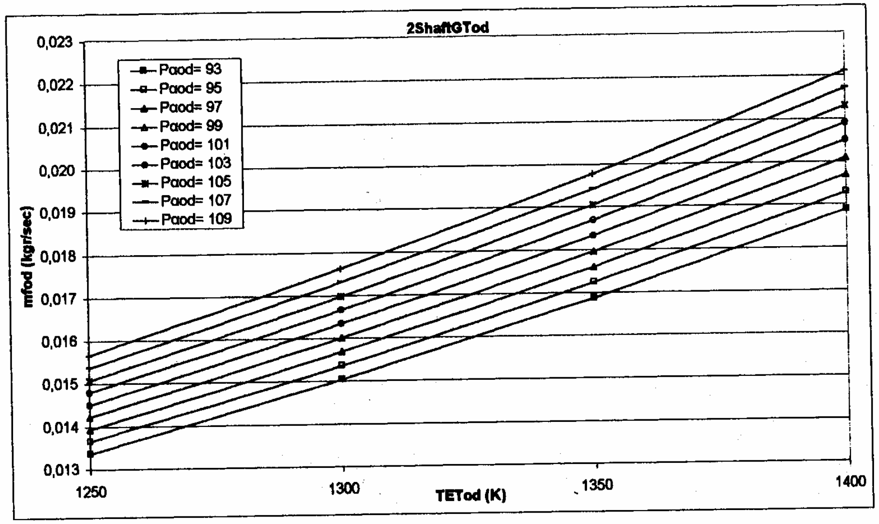


Fig. 3.60: 2Shaft GT, mf vs TET (parameter Pα)

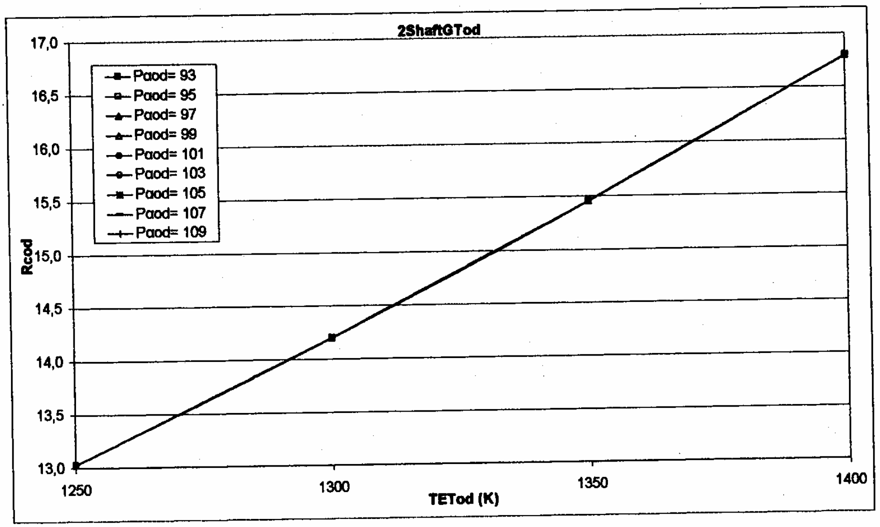


Fig. 3.61: 2Shaft GT, Rc vs TET (parameter Pα)

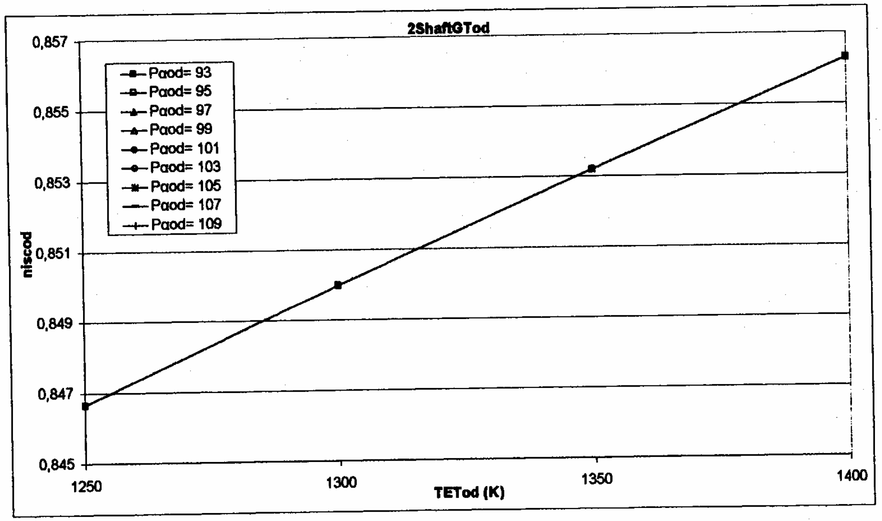


Fig. 3.62: 2Shaft GT, η_{isc} vs TET (parameter Pα)

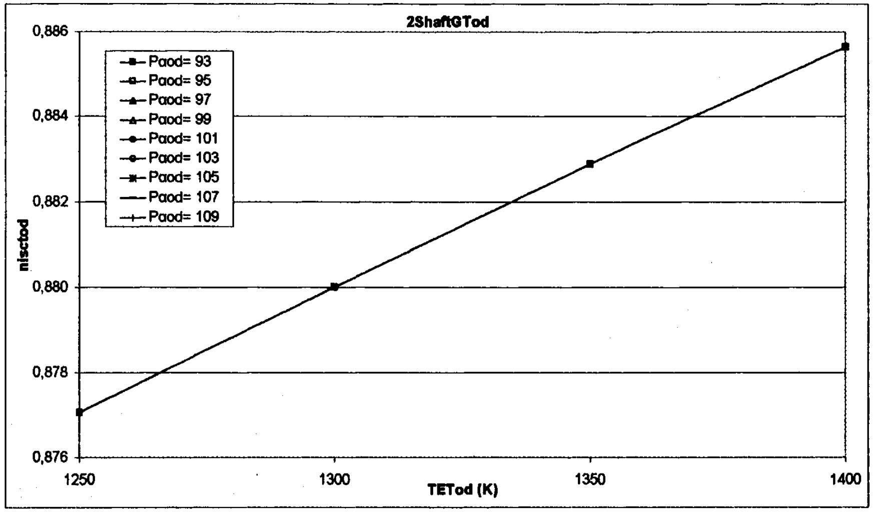


Fig. 3.63: 2Shaft GT, η_{isct} vs TET (parameter Pα)

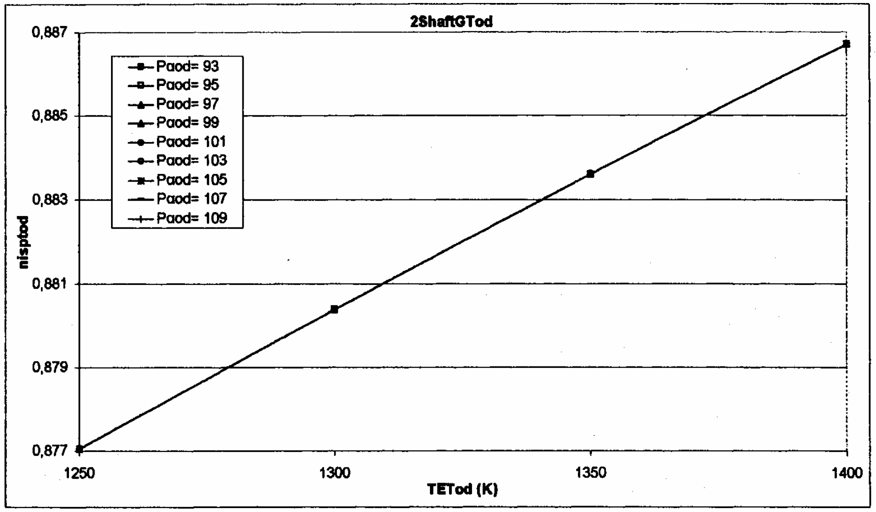


Fig. 3.64: 2Shaft GT, η_{ispt} vs TET (parameter Pα)

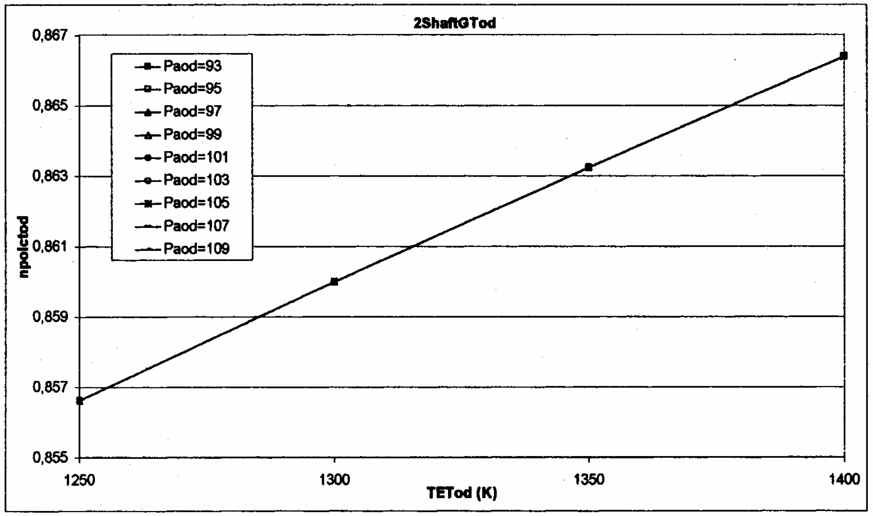


Fig. 3.65: 2Shaft GT, η_{polct} vs TET (parameter Pα)

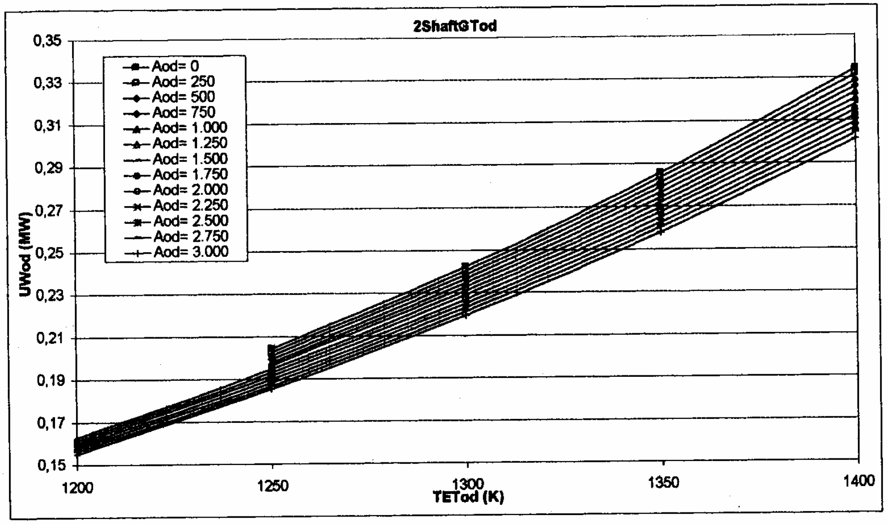


Fig. 3.66: 2Shaft GT, UW vs TET (parameter A)

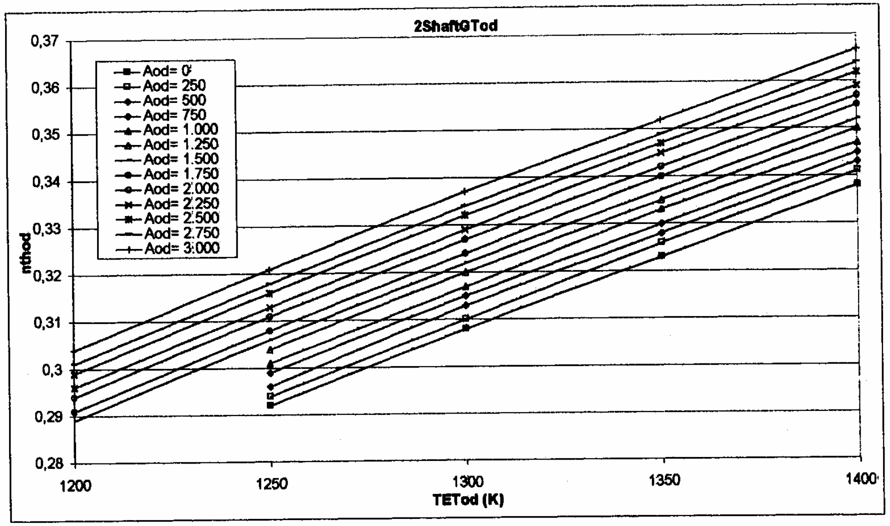


Fig. 3.67: 2Shaft GT, η_{th} vs TET (parameter A)

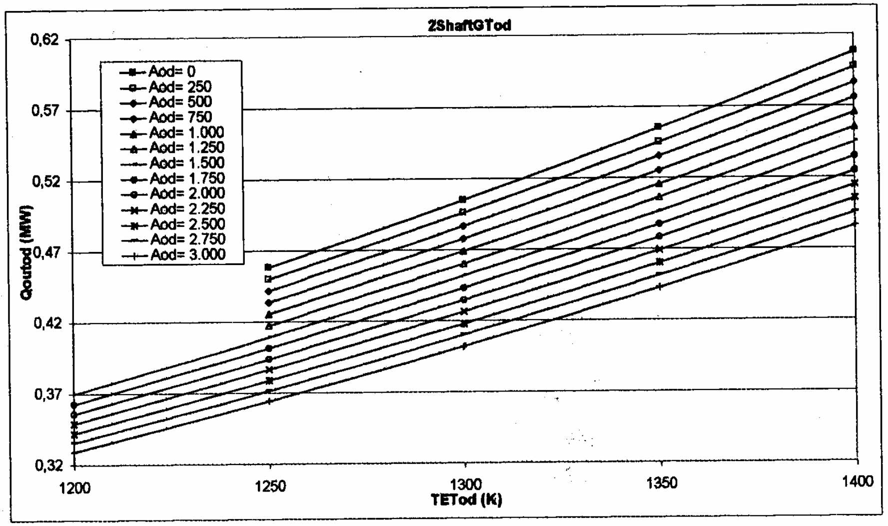


Fig. 3.68: 2Shaft GT, Qout vs TET (parameter A)

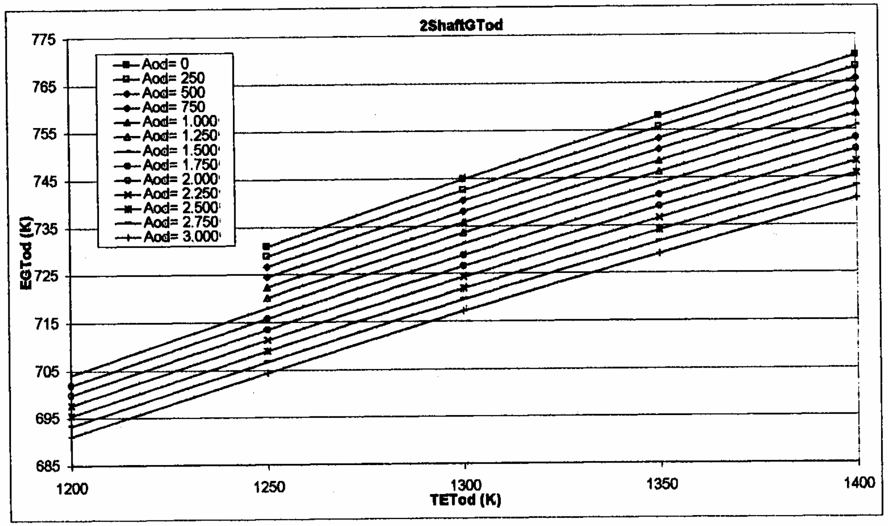


Fig. 3.69: 2Shaft GT, EGT vs TET (parameter A)

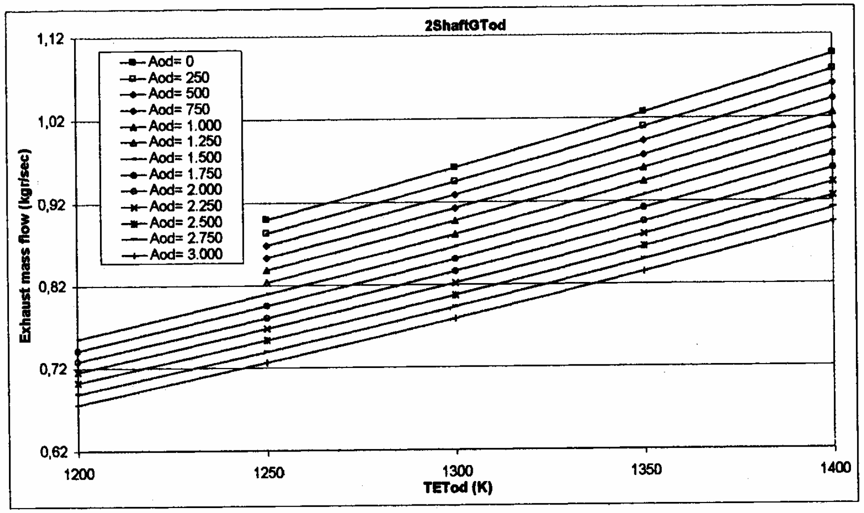


Fig. 3.70: 2Shaft GT, Exhaust mass flow vs TET (parameter A)

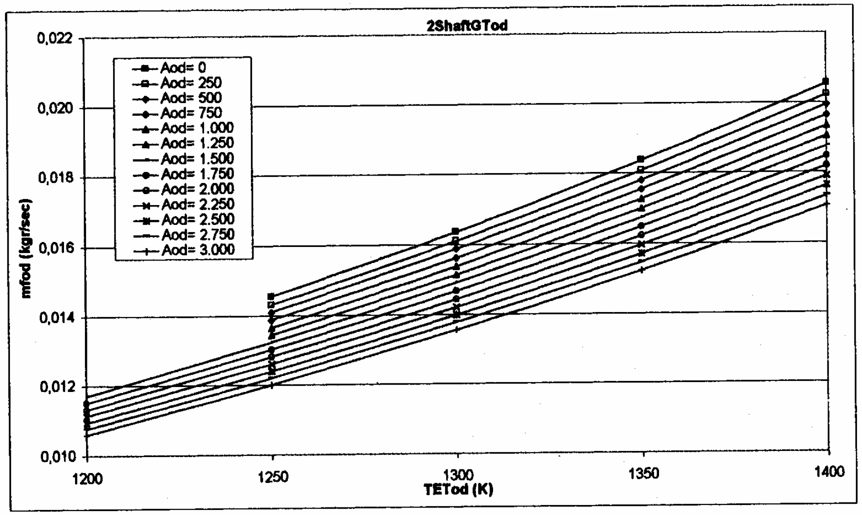


Fig. 3.71: 2Shaft GT, m_f vs TET (parameter A)

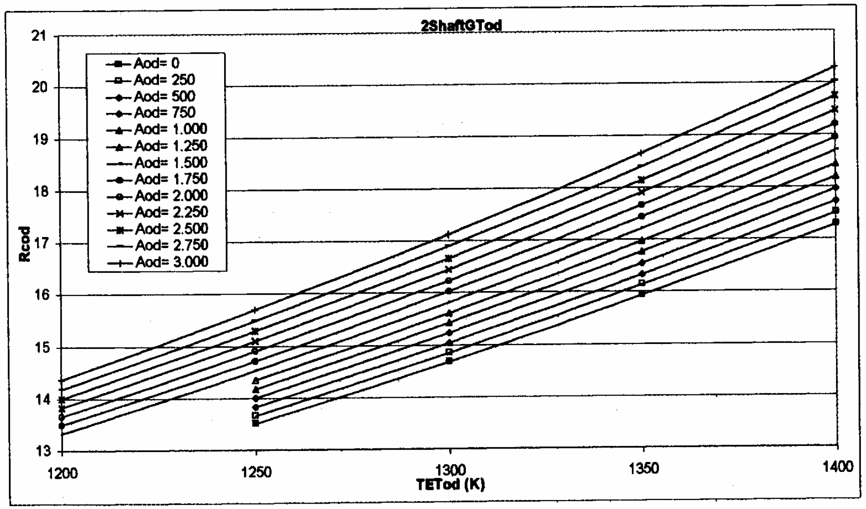


Fig. 3.72: 2Shaft GT, R_c vs TET (parameter A)

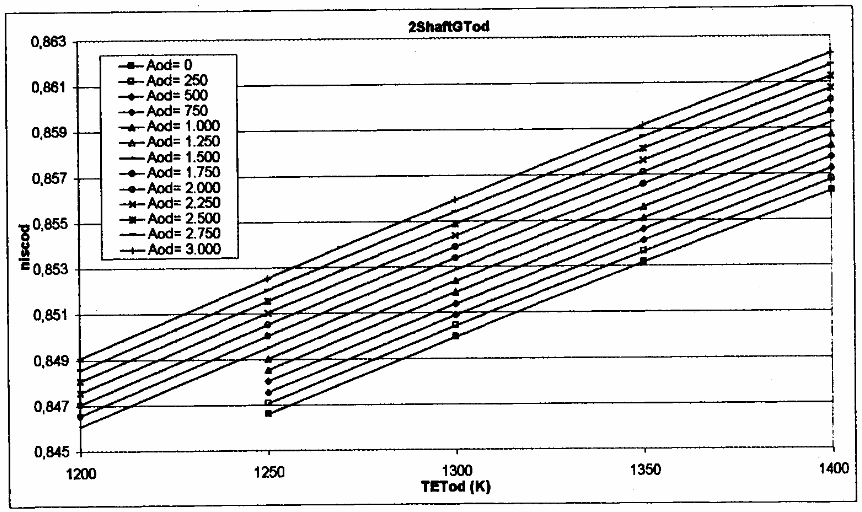


Fig. 3.73: 2Shaft GT, η_{isc} vs TET (parameter A)

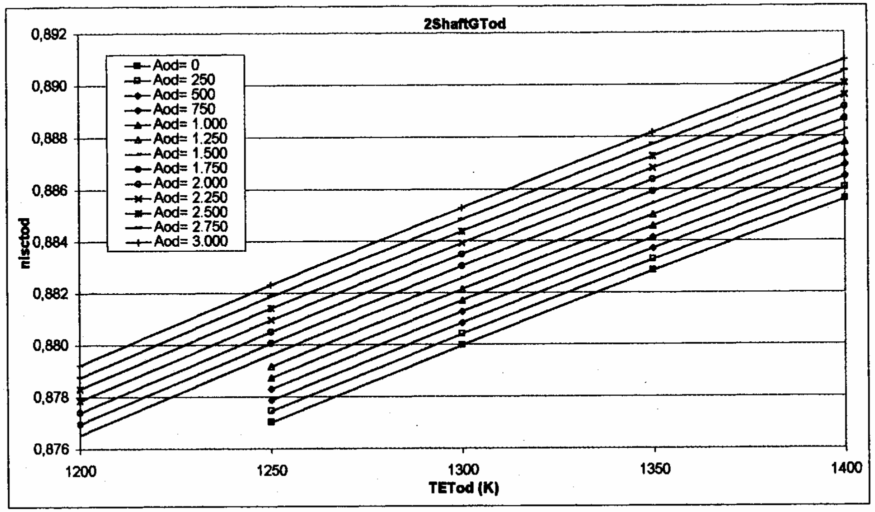


Fig. 3.74: 2Shaft GT, η_{isct} vs TET (parameter A)

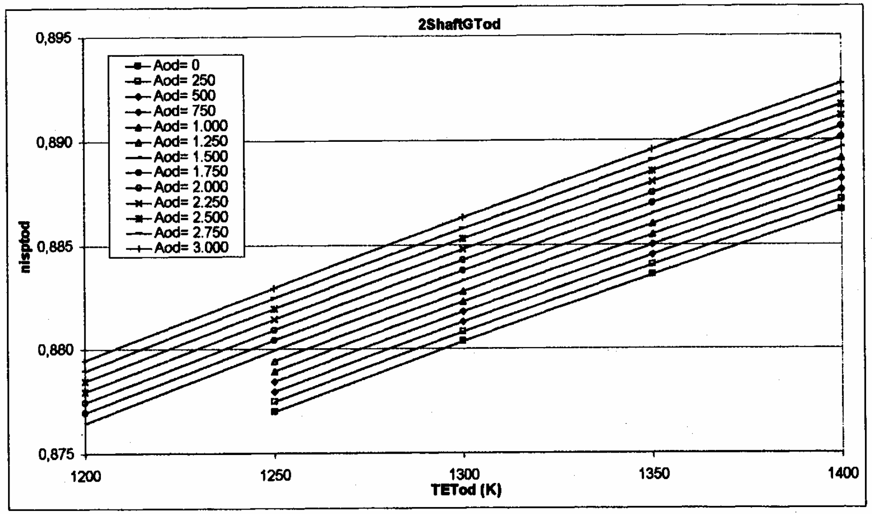


Fig. 3.75: 2Shaft GT, η_{ispt} vs TET (parameter A)

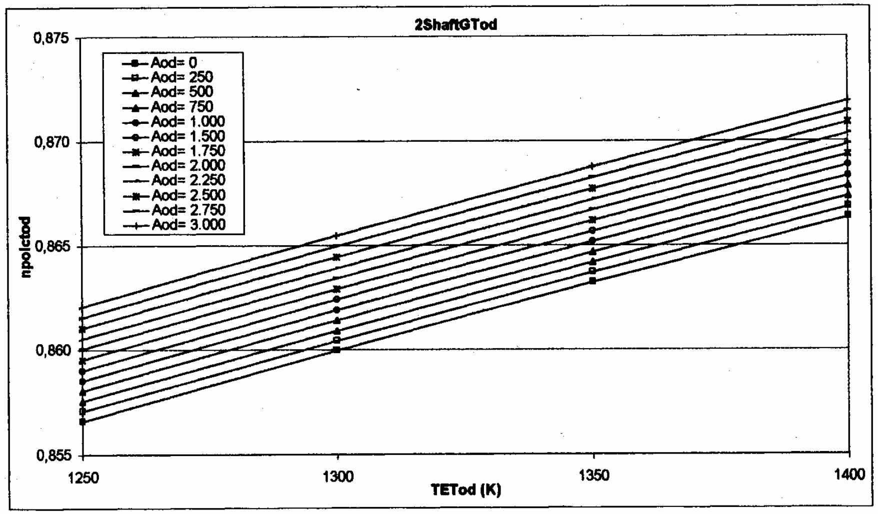


Fig. 3.76: 2Shaft GT, η_{polct} vs TET (parameter A)

3.6.3 1-Shaft with Heat Exchanger Cycle Simulation Procedure

Referring to *Fig. 3.5*.

<u>Input File (APPENDIX C.8)</u>

It is the same with those in paragraph 3.6.1 with the addition of the inputs for the two more components heat exchanger cold (HEC), heat exchanger hot (HET), simulating the heat exchanger.

Heat Exchanger inputs

The heat exchanger has effectiveness about 0.7÷0.85. So we assume efficiencies for the two simulation components HEC and HET: $\eta_{che} = \eta_{hhe}$=0.84÷0.92 ($\eta_{che} = \eta_{hhe}$=0.9).
Pressure losses in the two components HEC and HET: DPcheloss=DPhheloss=1-4%

Selection of the engine

In order to test the correct response of the off-design simulation program, we choose an engine with the following parameters:

1. TET=1,100K
2. R_C=5

<u>Calculation procedure</u>

Equations (3-106) – (3-113)

$$R_t = \frac{P_{o7}}{P_{o8}} \tag{3-200}$$

Then the OD calculation begins taking into account the results of the DP.

1-2 INTAKE

Equations (3-115) - (3-118)

2-3 COMPRESSOR

Equations (3-119), (3-120)

1-2 INTAKE

Equation (3-121)

9-10 EXHAUST

$$P_{o10}^{od} = P_{o1}^{od} \cdot 1.003 \tag{3-201}$$

$$P_{o9}^{od} = P_{o10}^{od} \tag{3-202}$$

8-9 HEAT EXCHANGER HOT

$$P_{o8}^{od} = \frac{P_{o9}^{od} \cdot 100}{100 - DP_{exh_{loss}}} \tag{3-203}$$

6-7 MIXER

$$T_{o6}^{od} = TET^{od} \tag{3-204}$$

$$T_{o7}^{od} = T_{o6}^{od} - DT_{cooling}^{od} \tag{3-205}$$

7-8 COMPRESSOR & POWER TURBINE

$$P_{o7}^{od} = \frac{\dot{m}^{od} \cdot P_{o7}}{\dot{m}} \cdot \left(\frac{\sqrt{T_{o7}^{od}}}{\sqrt{T_{o7}}} \right) \tag{3-206}$$

6-7 MIXER

$$P_{o6}^{od} = P_{o7}^{od} \tag{3-207}$$

5-6 COMBUSTION CHAMBER (BURNER)

$$P_{o5}^{od} = \frac{P_{o6}^{od}}{1 - \frac{DP_{cc_{loss}}}{100}} \tag{3-208}$$

3-4 PREMASS

$$P_{o4}^{od} = P_{o5}^{od} \tag{3-209}$$

6-7 MIXER

If TET = T_{o6} is less than 1,300 then it is assumed that there is no need for by pass mass flow to cool the turbine section, thus

$$D_{mc}=0,\ DT_{cooling}=0 \tag{3-210}$$

On the contrary, if T_{o6} greater or equal to 1,300 then there is a portion of mass flow (D_{mc}), which reduces the TET per $DT_{cooling}$:

$$D_{mc}=0.025*T_{o6}-25,\ DT_{cooling}=0.333*T_{o6}-333.333 \tag{3-211}$$

3-4 PREMASS

Equations (3-138), (3-133)

5-6 COMBUSTION CHAMBER (BURNER)

Equation (3-132)

2-3 COMPRESSOR

Equation (3-134)

$$\eta_{is_c}^{od} = \eta_{is_c} \cdot \left(1 - \left|\frac{P_{c_{de}}}{100}\right|\right) \cdot \sqrt[0.1]{\frac{T_{o7}^{od}}{T_{o2}^{od}} \cdot \frac{T_{o2}}{T_{o7}}} \tag{3-212}$$

Equations (3-136), (3-137)

3-4 PREMASS

Equation (3-139)

4-5 HEAT EXCHANGER COLD

$$T_{o5}^{od} = T_{o4}^{od} \tag{3-213}$$

$$\dot{m}_5^{od} = \dot{m}_4^{od} \tag{3-214}$$

5-6 COMBUSTION CHAMBER (BURNER)

$$Q_{cc}^{od} = \frac{\dot{m}_5^{od} \cdot \left(C_{p_h} \cdot T_{o6}^{od} - C_{p_c} \cdot T_{o5}^{od}\right)}{1{,}000{,}000} \tag{3-215}$$

Equation (3-141)

$$FAR_{56}^{od} = \frac{\dot{m}_f^{od}}{\dot{m}_5^{od}} \tag{3-216}$$

$$\dot{m}_6^{od} = \dot{m}_5^{od} + \dot{m}_f^{od} \tag{3-217}$$

6-7 MIXER

$$\dot{m}_7^{od} = \dot{m}^{od} + \dot{m}_f^{od} \tag{3-218}$$

7-8 COMPRESSOR & POWER TURBINE

$$\eta_{is_t}^{od} = \eta_{is_t} \cdot \left(1 - \left|\frac{P_{t_{de}}}{100}\right|\right) \cdot \sqrt[0.1]{\frac{T_{o7}^{od}}{T_{o2}^{od}} \cdot \frac{T_{o2}}{T_{o7}}} \tag{3-219}$$

$$T_{o8}^{od} = T_{o7}^{od} \cdot \left[1 - \left[1 - \left(\frac{P_{o8}^{od}}{P_{o7}^{od}}\right)^{(\gamma_h - 1)/\gamma_h}\right] \cdot \eta_{is_t}^{od}\right] \tag{3-220}$$

$$\dot{m}_8^{od} = \dot{m}_7^{od} \tag{3-221}$$

8-9 HEAT EXCHANGER HOT

$$R_t^{od} = \frac{P_{o7}^{od}}{P_{o8}^{od}} \tag{3-222}$$

If R_t^{od} is less or equal to R_{tch} then, the turbine is unchoked, so the results are unreliable, else performance calculation can continue.

If T_{o8}^{od} is less or equal to T_{o4}^{od} then, the heat exchanger goes to inversed operation mode, so the results are unreliable, else performance calculation can continue.

$$T_{o9}^{od} = T_{o8}^{od} - \left[\eta_{h_{he}} \cdot \left(T_{o8}^{od} - T_{o4}^{od}\right)\right] \tag{3-223}$$

$$\dot{m}_9^{od} = \dot{m}_7^{od} \tag{3-224}$$

9-10 EXHAUST

$$T_{o10}^{od} = T_{o9}^{od} \tag{3-225}$$

$$\dot{m}_{10}^{od} = \dot{m}_7^{od} \tag{3-226}$$

PERFORMANCE

Equation (3-150)

$$TW^{od} = \frac{\dot{m}^{od} \cdot C_{p_h} \cdot \left(T_{o7}^{od} - T_{o8}^{od}\right)}{1{,}000{,}000} \tag{3-227}$$

Equations (3-152) - (3-156)

$$Q_{out}^{od} = \frac{\dot{m}_{10}^{od} \cdot C_{p_h} \cdot \left(T_{o10}^{od} - T_{\alpha}^{od}\right)}{1{,}000{,}000} \tag{3-228}$$

1. When ambient temperature T_α varies, ambient pressure P_α remains constant, and then in the equations above we substitute $P_\alpha^{od} = P_\alpha$.
2. When ambient pressure P_α varies, ambient temperature T_α remains constant, and then in the equations above we substitute $T_\alpha^{od} = T_\alpha$.
3. When altitude varies, then P_α^{od} and T_α^{od} remain as they are, but they change according to the equations: (3-158), (3-159)

The results of the above calculation procedure for the three variables T_α, P_α and altitude -using the FORTRAN program developed by the author- are represented in the following *Figs*.

The above calculation method is simplified but realistic. If required o more detailed model it can easily replace this one, due to module-construction of the overall simulation program developed by the author.

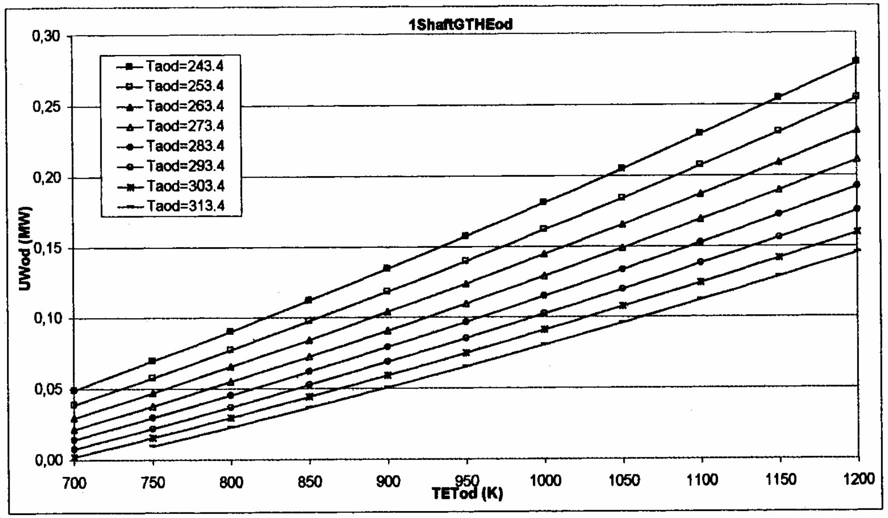


Fig. 3.77: 1Shaft GT HE, UW vs TET (parameter Tα)

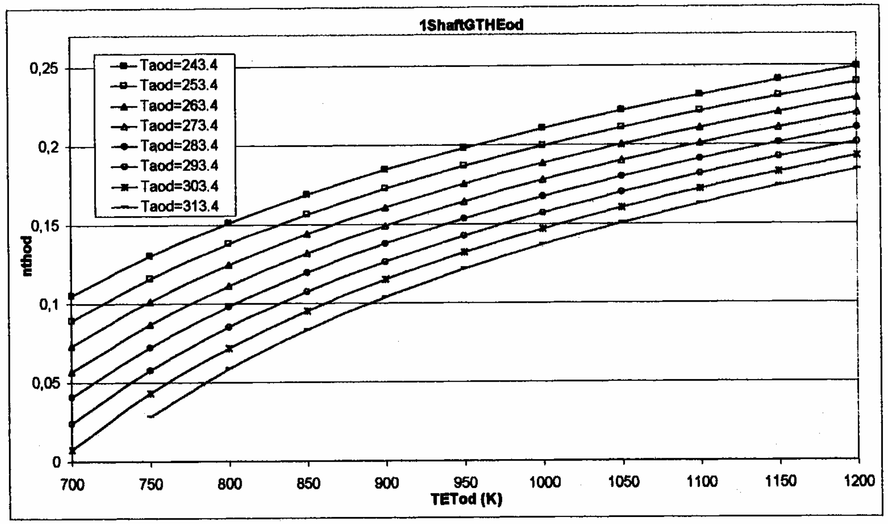


Fig. 3.78: 1Shaft GT HE, η_{th} vs TET (parameter Tα)

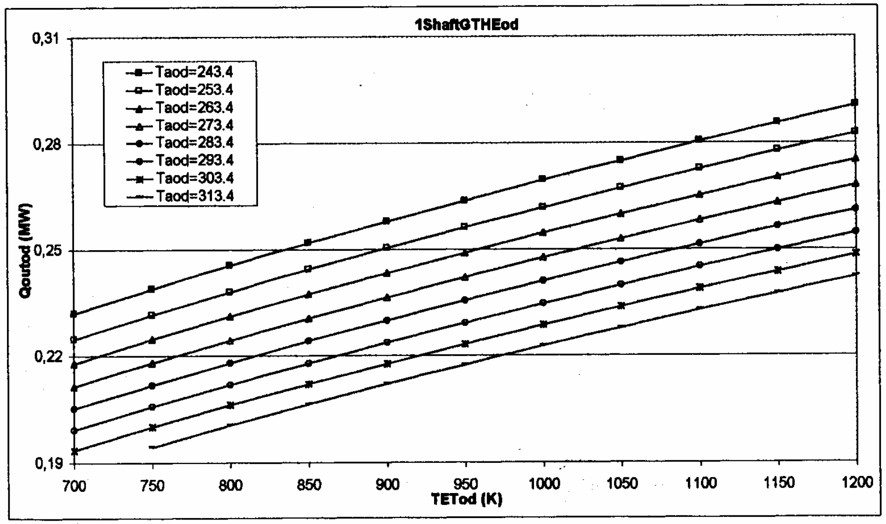


Fig. 3.79: 1Shaft GT HE, Qout vs TET (parameter Tα)

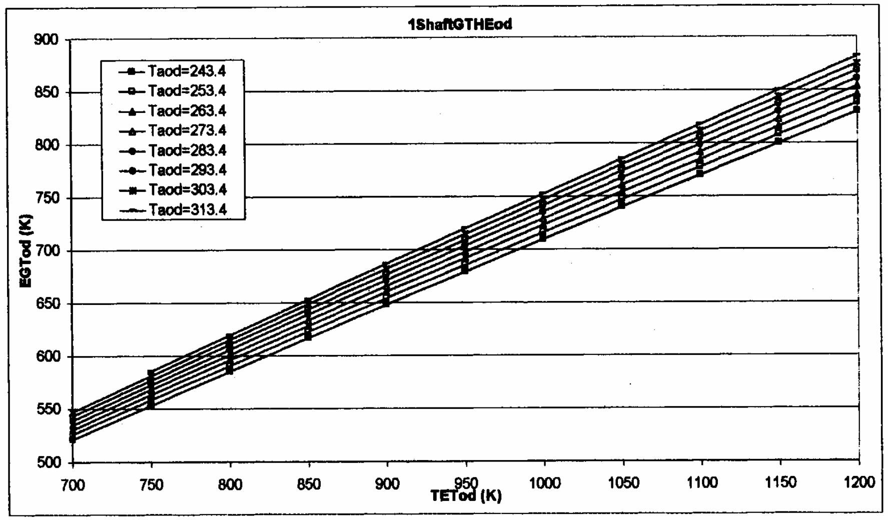


Fig. 3.80: 1Shaft GT HE, EGT vs TET (parameter Tα)

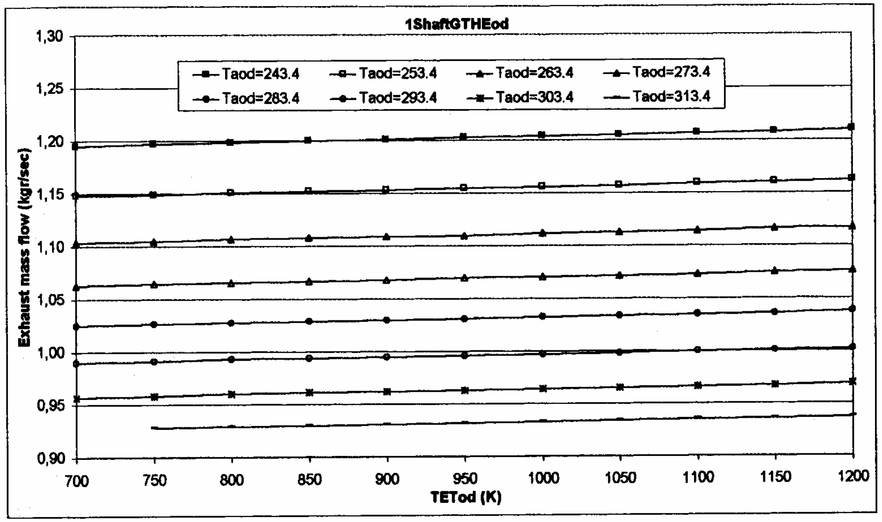


Fig. 3.81: 1Shaft GT HE, Exhaust mass flow vs TET (parameter Tα)

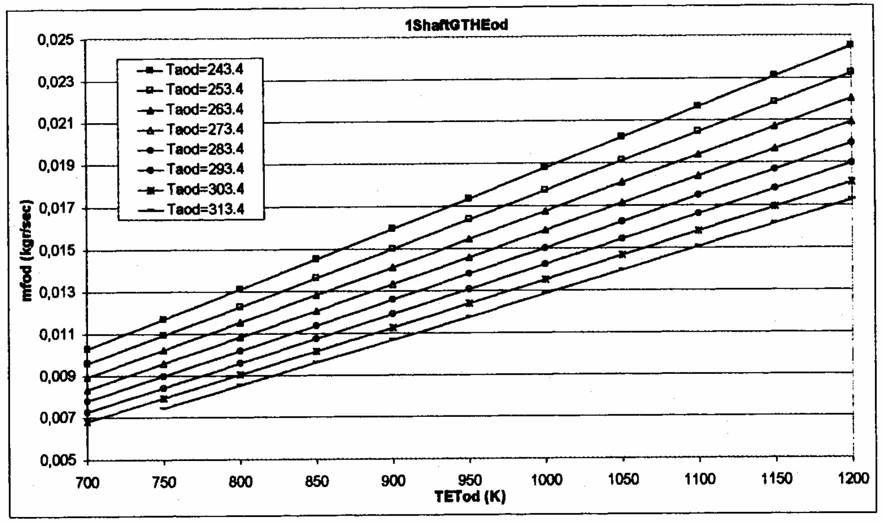


Fig. 3.82: 1Shaft GT HE, m_f vs TET (parameter Tα)

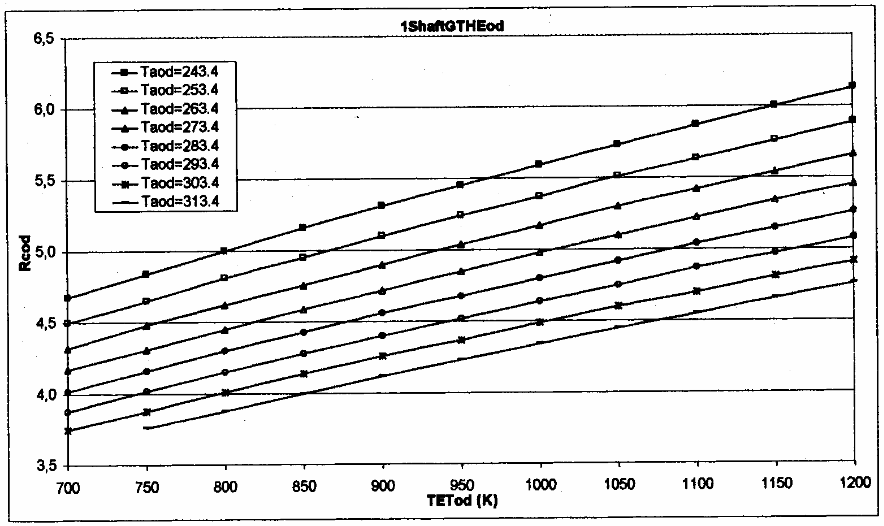


Fig. 3.83: 1Shaft GT HE, R_c vs TET (parameter Tα)

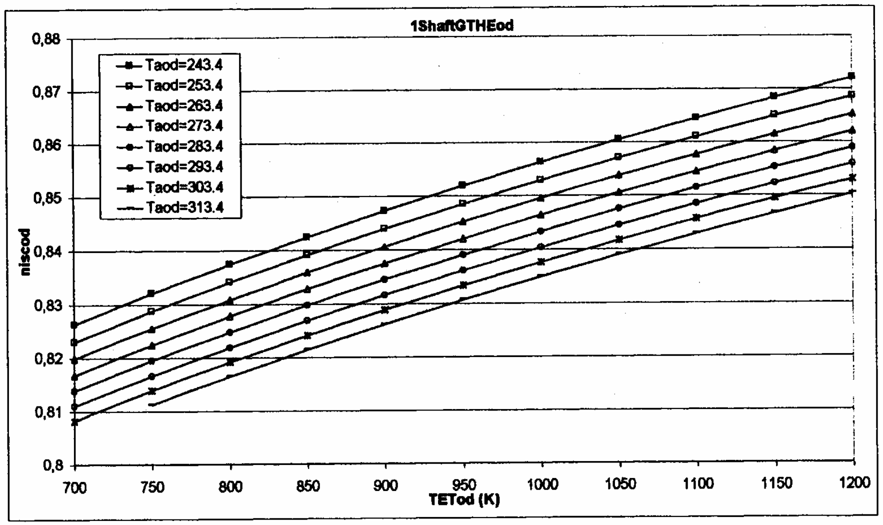


Fig. 3.84: 1Shaft GT HE, η_{isc} vs TET (parameter Tα)

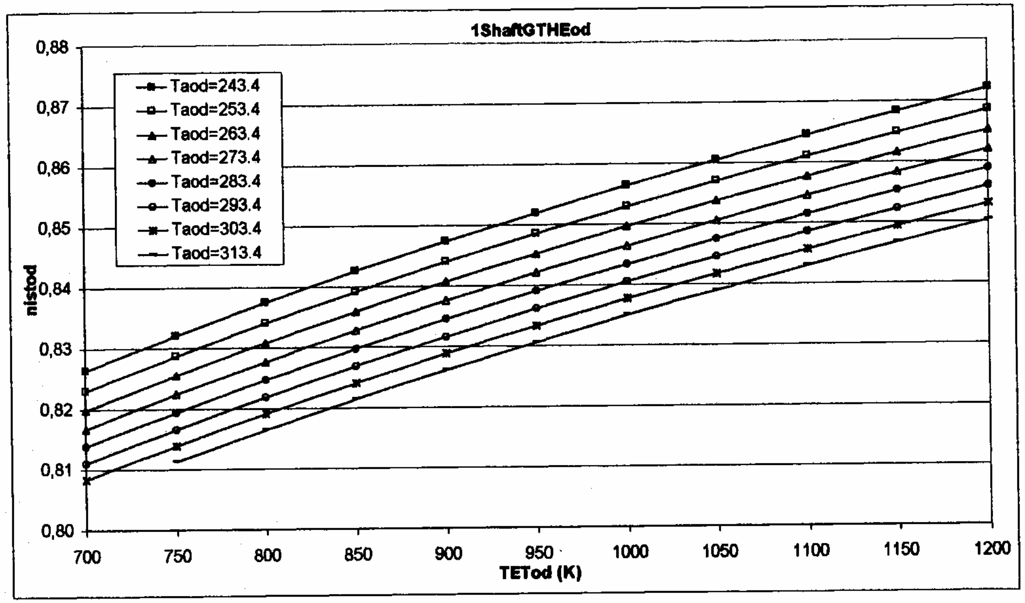


Fig. 3.85: 1Shaft GT HE, η_{ist} vs TET (parameter $T\alpha$)

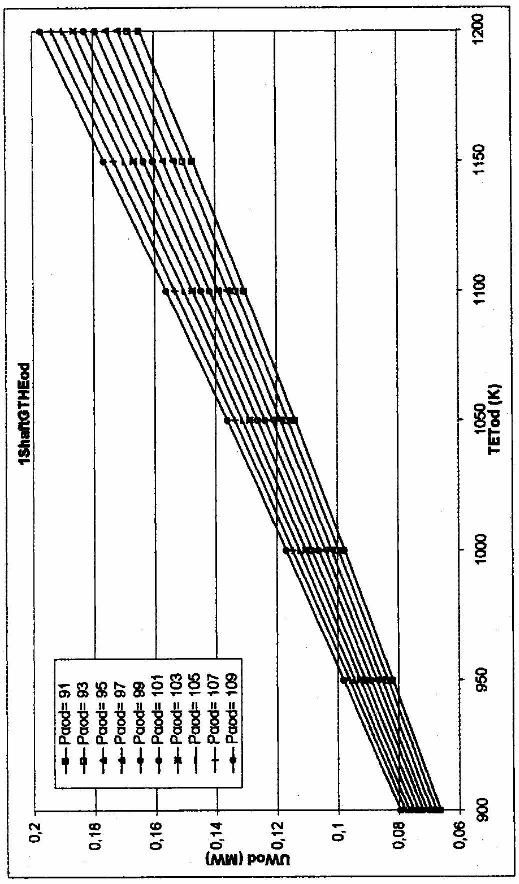


Fig. 3.86: 1Shaft GT HE, UW vs TET (parameter Pα)

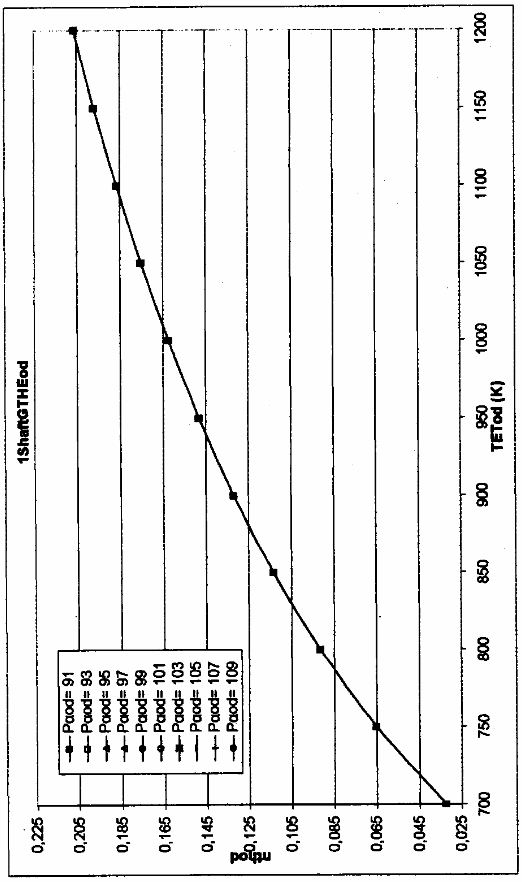


Fig. 3.87: 1Shaft GT HE, ηth vs TET (parameter Pα)

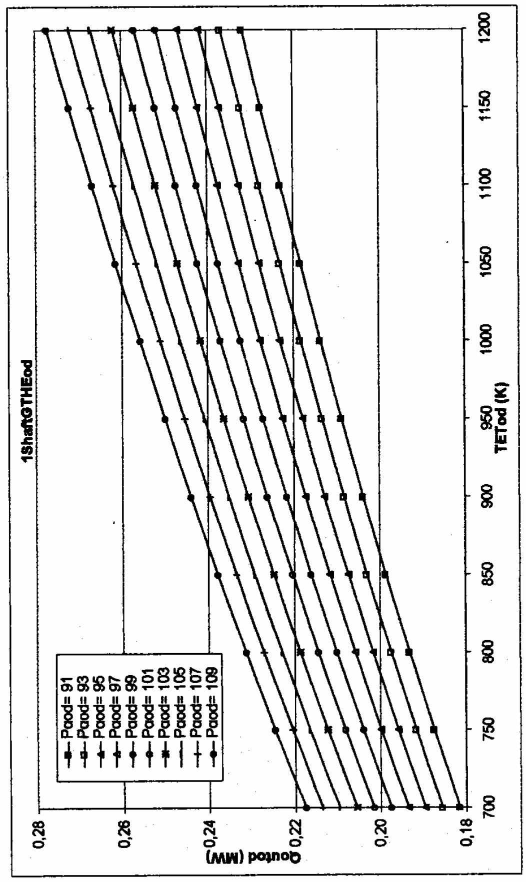


Fig. 3.88: 1Shaft GT HE, Qout vs TET (parameter Pα)

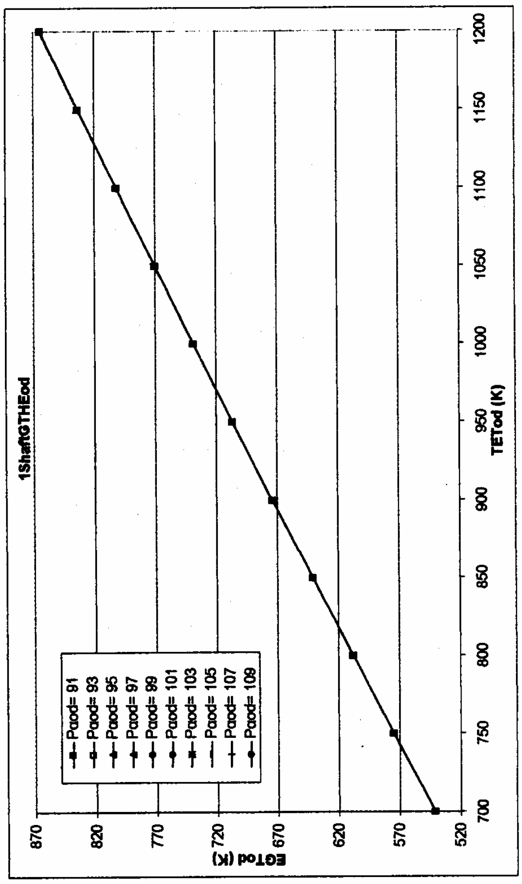


Fig. 3.89: 1Shaft GT HE, EGT vs TET (parameter Pα)

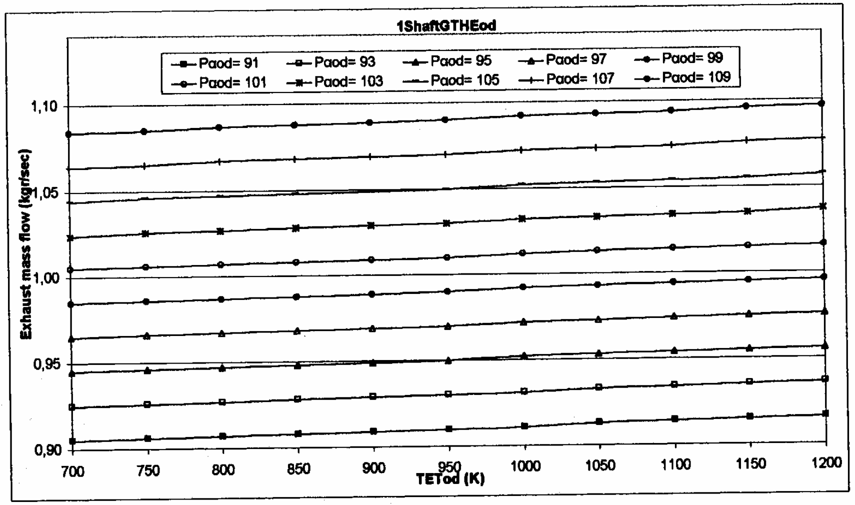


Fig. 3.90: 1Shaft GT HE, Exhaust mass flow vs TET (parameter Pα)

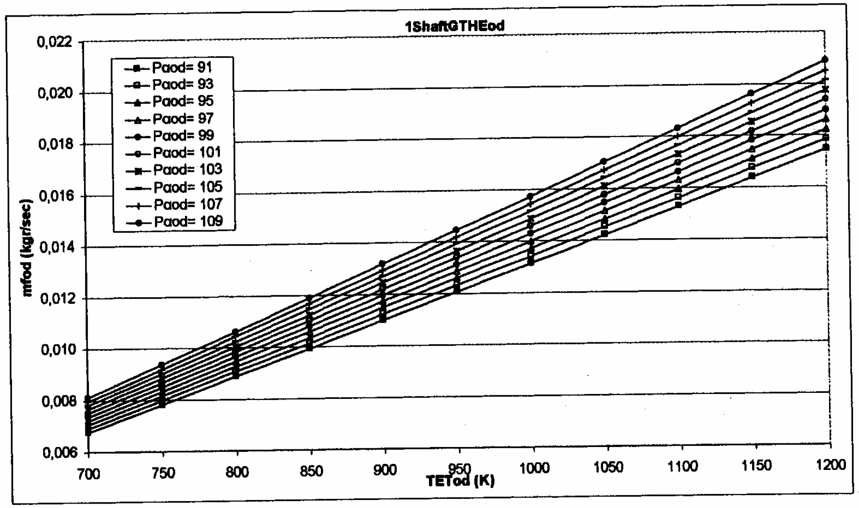


Fig. 3.91: 1Shaft GT HE, mf vs TET (parameter Pα)

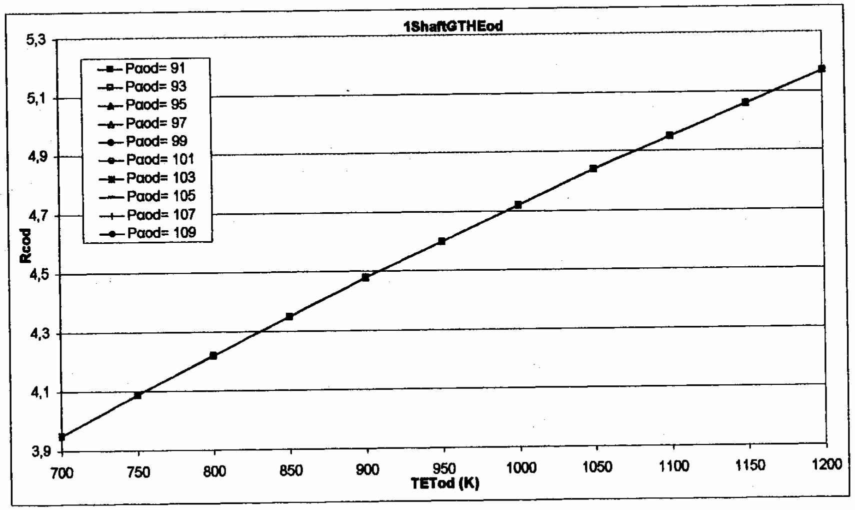


Fig. 3.92: 1Shaft GT HE, Rc vs TET (parameter Pα)

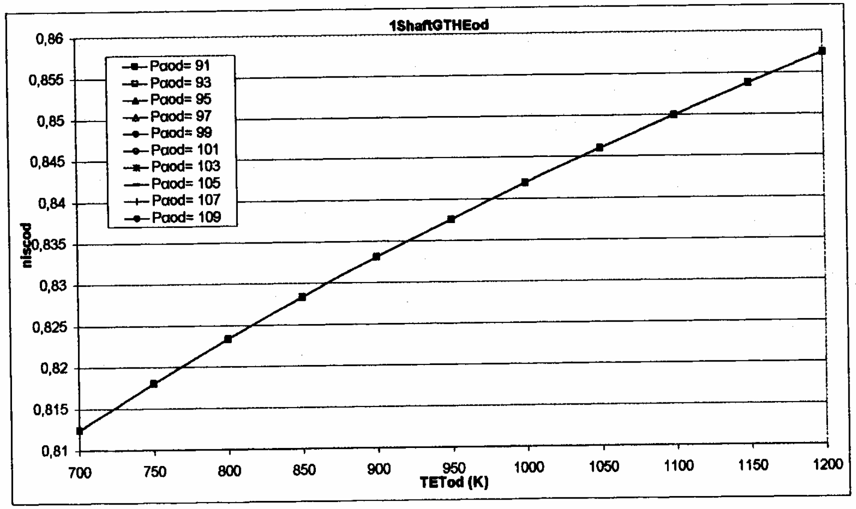


Fig. 3.93: 1Shaft GT HE, η_{isc} vs TET (parameter Pα)

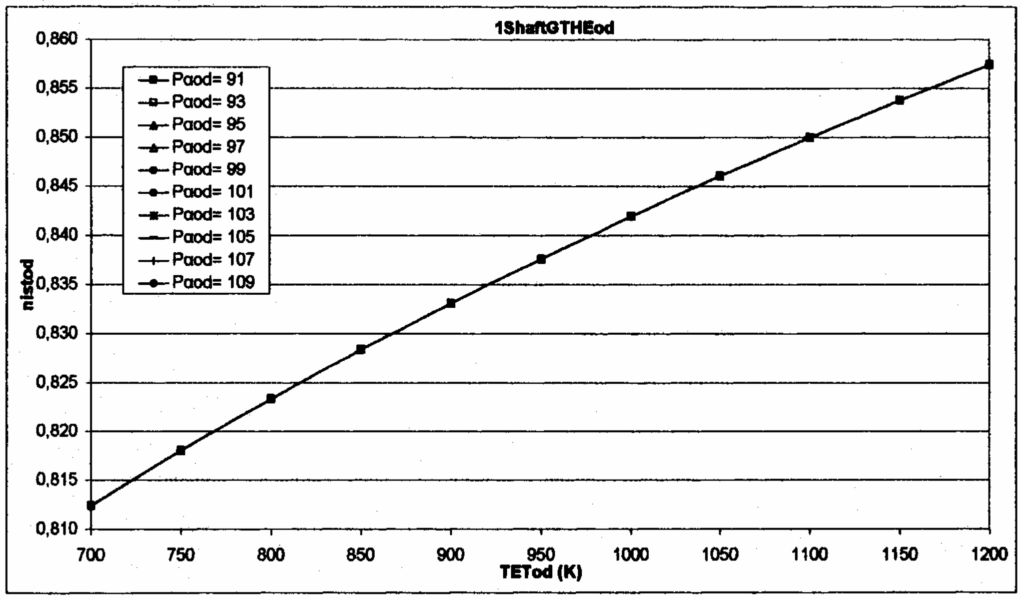


Fig. 3.94: 1Shaft GT HE, η_{ist} vs TET (parameter Pα)

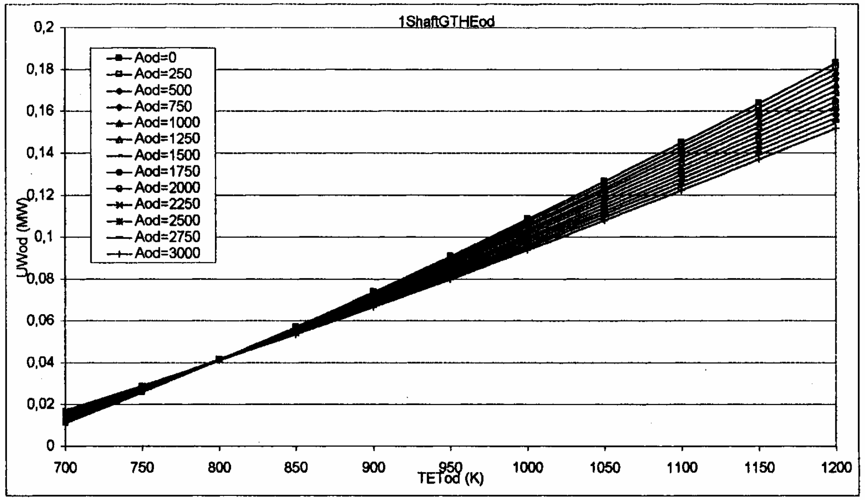


Fig. 3.95: 1Shaft GT HE, UW vs TET (parameter A)

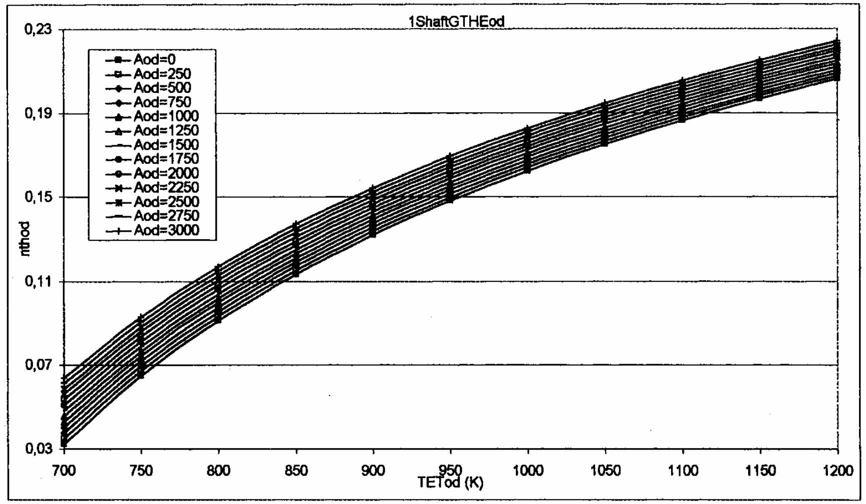


Fig. 3.96: 1Shaft GT HE, η_{th} vs TET (parameter A)

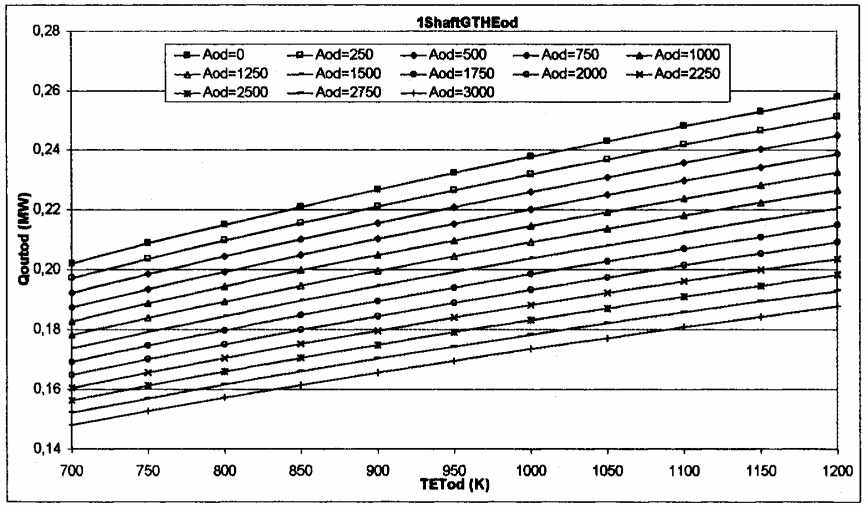


Fig. 3.97: 1Shaft GT HE, Qout vs TET (parameter A)

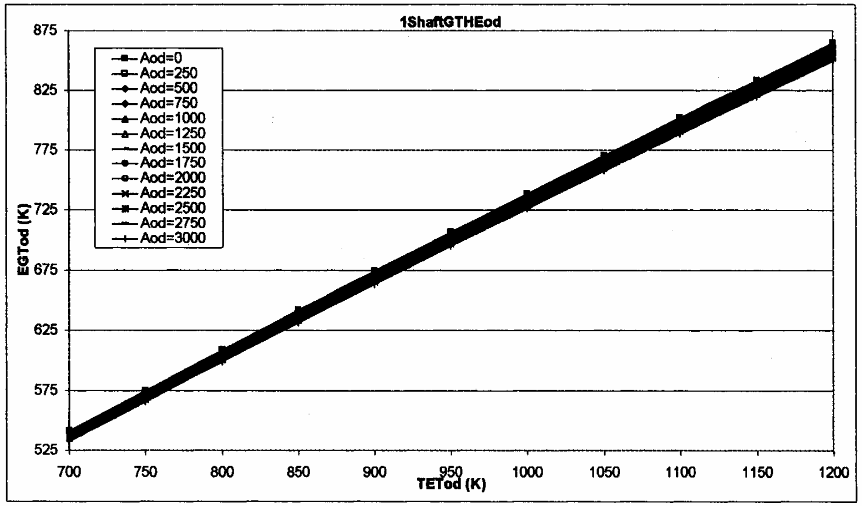


Fig. 3.98: 1Shaft GT HE, EGT vs TET (parameter A)

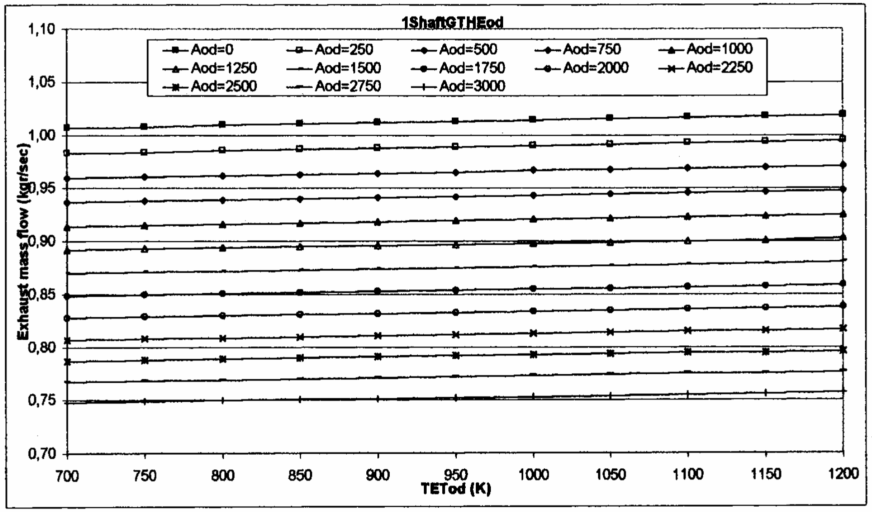


Fig. 3.99: 1Shaft GT HE, Exhaust mass flow vs TET (parameter A)

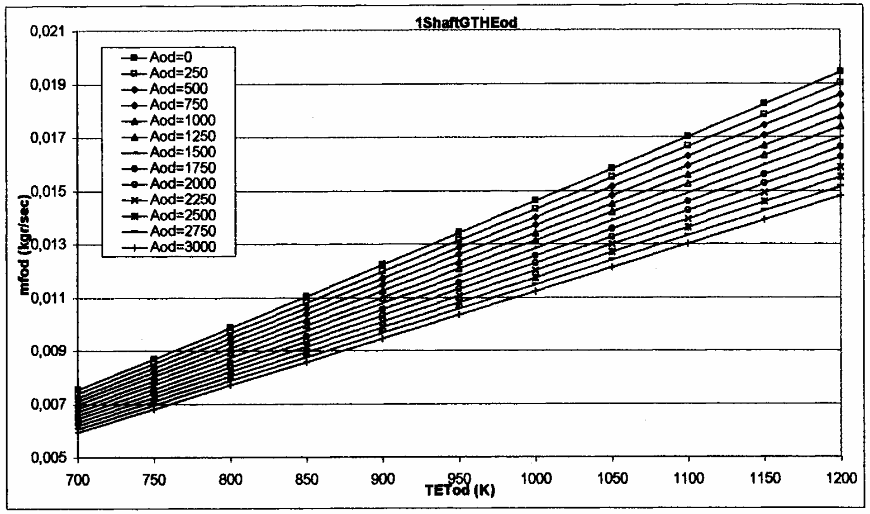


Fig. 3.100: 1Shaft GT HE, mf vs TET (parameter A)

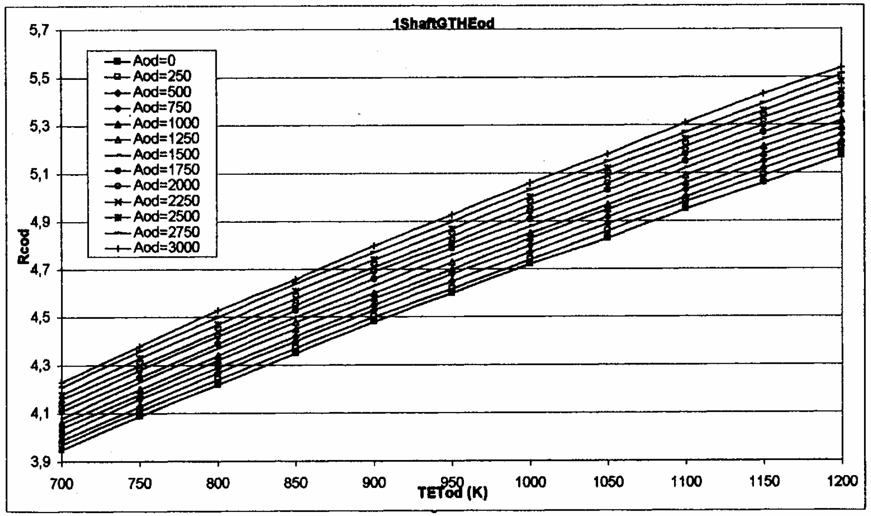


Fig. 3.101: 1Shaft GT HE, Rc vs TET (parameter A)

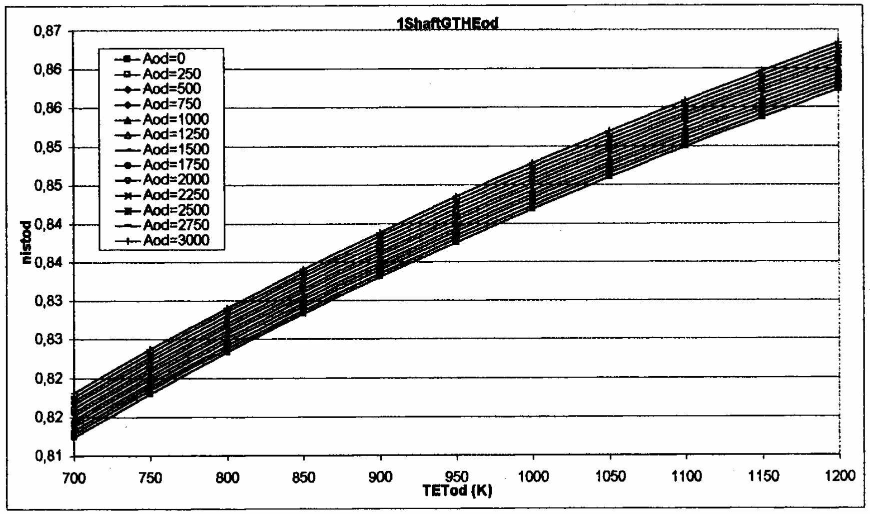


Fig. 3.102: 1Shaft GT HE, η_{isc} vs TET (parameter A)

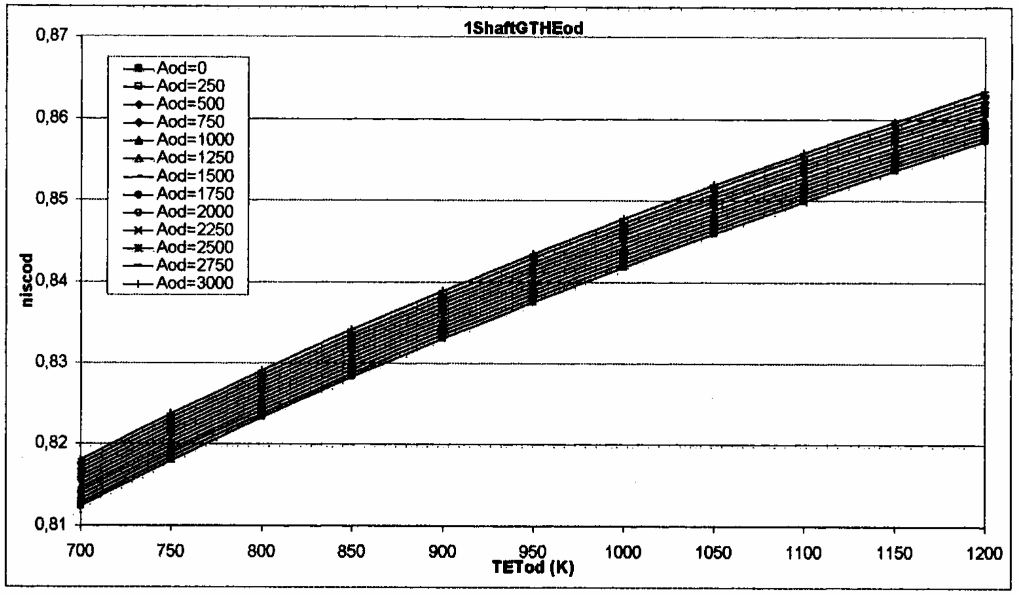


Fig. 3.103: 1Shaft GT HE, η_{ist} vs TET (parameter A)

3.7 Off-Design Performance Results Analysis

Effect of altitude

An important aspect that an engineer has to consider is the behavior of the gas turbine at different altitudes. Usually, this investigation takes place for aero-engines but due to increasing demand of supplying industrial gas turbines to areas of high altitude, e.g. Mexico, South America and Central Europe, the above investigation is necessary.

As is well known, the ambient static temperature falls linearly with increasing altitude from the sea level to –56.5°C at the tropopause (at 11km). The ambient static pressure and density also fall but more rapidly (exponentially).

Let us consider the case of the **1-shaft engine**. Because density falls and the engine runs with N=constant the mass flow $\dot{m}$ will drop, which has a significant negative impact on the UW. This reduction in $\dot{m}$ is independent of the TET value. **A B C D**

However, since the compressor entry total temperature T_{o2} is proportional to the ambient static value T_α, and since the rotational speed N is constant, the $\frac{N}{\sqrt{T_{o2}}}$ increases with altitude. Let us now consider *Fig. 3.104*.

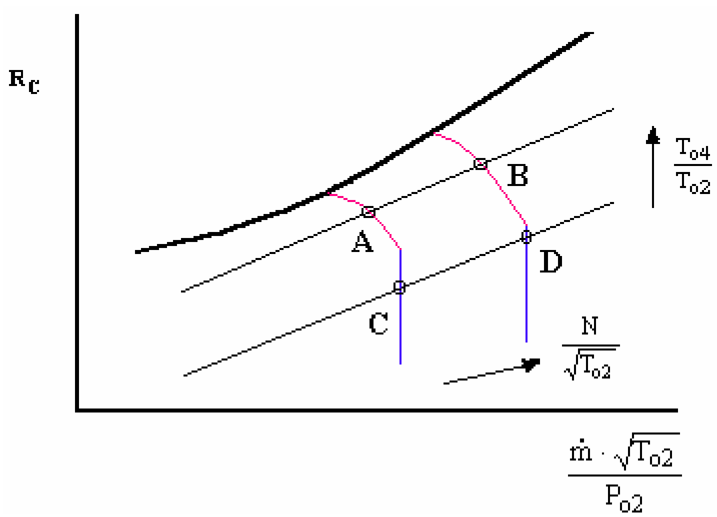


Fig. 3.104: Effect of rising altitude at different TETs

The curves of constant $\frac{N}{\sqrt{T_{o2}}}$ are generally divided in two parts the upper (with a slope) and the lower (vertical). If the operating point is **A** and it is located at the upper part (corresponds to relatively high TET=T_{o4}), then an increase in altitude will send the operating point to the right to a higher constant $\frac{N}{\sqrt{T_{o2}}}$ line and slightly up to increased R_C, (point **B**).

On the other hand, if the operating point is **C** and it is located at the lower part (corresponds to relatively lower TET=T_{o4}), then an increase in altitude, will sent the operating point again to the right on a higher constant $\frac{N}{\sqrt{T_{o2}}}$ line and up, to higher R_C (more than the previous case, point **D**). We know that in the case of 1-shaft engines $R_C \approx R_T$, thus in the second situation (TET relatively low), the pressure ratio and consequently the temperature ratio of the turbine will be relatively higher. This implies that the turbine work is higher. In the end, there will be TET, below which the increase of turbine temperature ratio will overweigh the reduction of the mass flow, thus the UW will rise slightly when the altitude rises. (*Figs 3.35, 3.104*)

Thermal efficiency increases with altitude increases. This is due to, as previously explain, the pressure ratio increase with altitude. Thermal efficiency depends on pressure ratio, thus thermal efficiency increases with altitude for constant TET.

In the case of **2-shaft engines** the situation is more straightforward, due to the fact that the running line has a slope which is approximate to the slope of different TETs. Thus, the reduction of mass flow $\dot{m}$ is always the dominant effect. (*Fig. 3.107*)

Effect of temperature

It is well known that the performance of a gas turbine is dependent on the ambient temperature (at fixed ambient pressure). During hot summer days, as ambient temperature increases, the air density falls and hence so does the air mass flow, $\dot{m}$.

In the ideal cycle when ambient temperature increases the compression work increases (since compressor entry total temperature is proportional to ambient temperature) but the expansion work remains constant. The useful work is the difference between them (UW = EW - CW), so it decreases. (*Fig. 3.105*) Therefore, the UW decreases. The pressure ratio of the cycle has not been affected, thus thermal efficiency is not affected because it is a function of pressure ratio only.

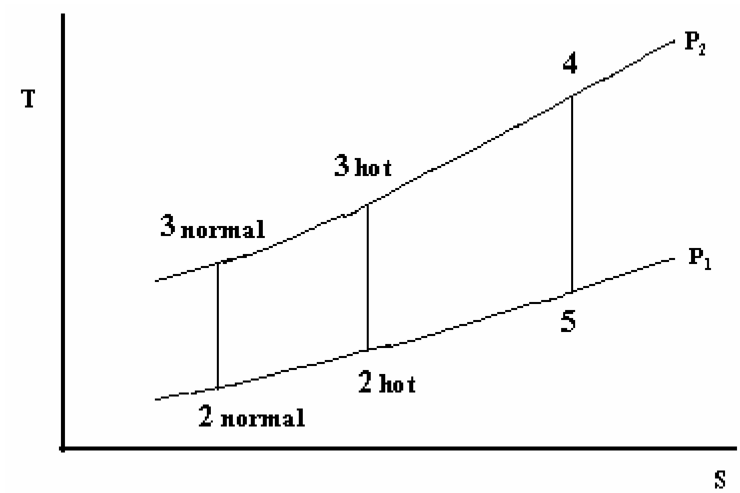


Fig. 3.105: Effect of increased T_α on the ideal cycle [16]

In a real cycle, thermal efficiency depends, also, on the temperature ratio ($\frac{T_{o4}}{T_{o2}}$), so the thermal efficiency will be affected, along with the UW.

When the **engine operates at constant shaft speed** on a hot day, the $\frac{N}{\sqrt{T_{o2}}}$ will be lower than during a normal day, as well as the compressor pressure and temperature ratio. (*Fig. 3.104* **B** $\rightarrow$ **A**) Thus, the UW and thermal efficiency will be reduced. (*Fig. 3.106*).

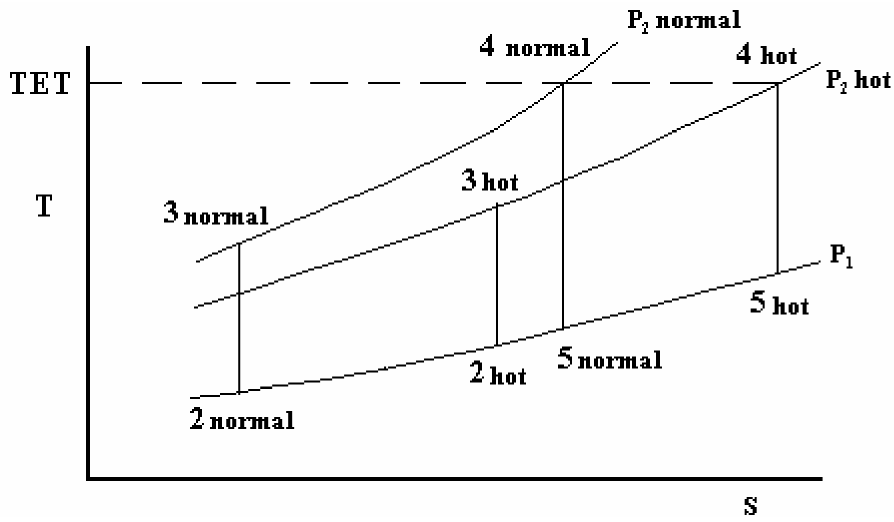


Fig. 3.106: Change of the ideal cycle due to T_a [16]

For a **constant TET**, the increase of ambient temperature results in a drop of fuel flow. Fuel flow depends on ratio HI/FCV; specifically, heat input (HI) divided by fuel calorific value (FCV). FCV is constant and HI is equal to the specific heat of gas at constant pressure (Cp) mu1tiplied by the difference between TET and combustor inlet temperature (HI=TET-T_{o2hot}), Thus, if ambient temperature increases, T_{o2hot} increases. On the other hand, the reduction in R_C is not enough so T_{o3hot}>$T_{o3normal}$. This results in falling HI, fuel flow and mass flow also.

It is obvious that hot days can dramatically affect gas turbine performance by decreasing the inlet mass flow and the pressure ratio. This explains why "**chilling**" plants based on a cooled inlet mass flow, provide better efficiency and power output than normal plants.

The ambient temperature affects gas turbine exhaust conditions, which determine the performance of the steam cycle or absorption cooling system. Thus, even a small change in ambient temperature modifies both exhaust mass flow and temperature and thus makes the steam cycle run off-design. However, due to the fact that both parameters behave in an opposite way overall **CC plant** efficiency is not as sharply affected as plant UW is. In conclusion, hot days mean less UW for the plant and perhaps a poorer efficiency.

Effect of altering the TET (part load, overload)
The increase in TET generally has a positive effect on all the engine parameters. This is more obvious at higher TETs because the component losses become relatively less important. Obviously, care must always be taken in order not to exceed the thermal limitations of the turbine working components (blades, rotors etc.). **Above 1.300K a cooling system is necessary.**

The choice of whether to use a 1-shaft or twin-shaft (free turbine) power plant is largely determined by the characteristics of the driven load. An electric generator requires a constant rotational speed and an engine designed specifically for this application would make use of a 1-shaft configuration. An alternative, however, is the use of an aircraft derivative with a free power turbine in the place of the propelling nozzle. With this arrangement it is possible to design a power turbine of substantially larger diameter than the gas generator, using an elongated duct between the gas generator and the power turbine; this then permits the power turbine to operate at the required electric generator speed without the need for a reduction gearbox.

The running lines for 1-shaft and twin-shaft units are shown in *Fig. 3.107*.

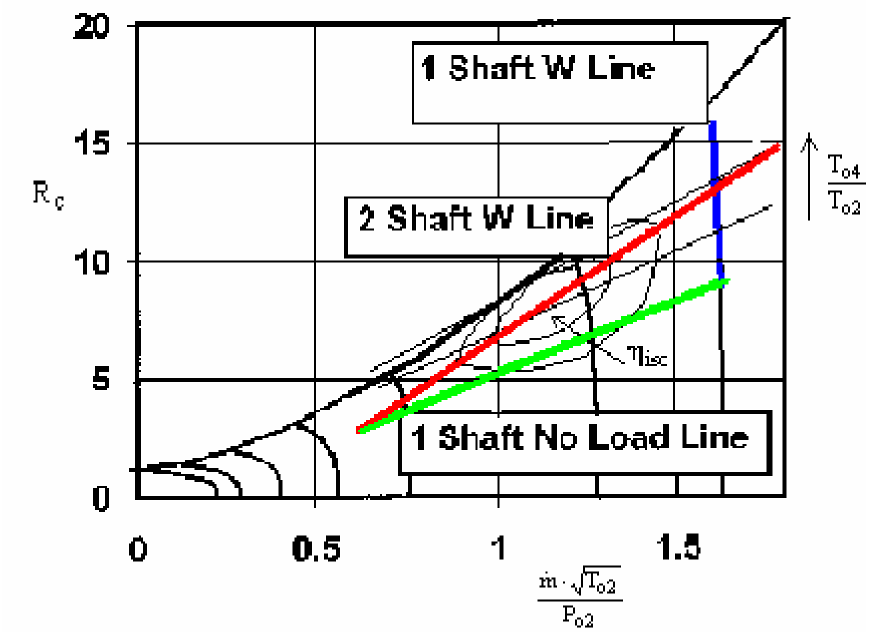


Fig. 3.107: Running lines of 1-Shaft and 2-Shaft GT$_S$ [16]

It should be noted that in the case of the **1-shaft** engine driving a generator, reduction in output power results in a slight increase in compressor mass flow. Although there is some reduction in compressor pressure ratio, there is little change in compressor temperature rise because the efficiency is also reduced. This means that the compressor power remains essentially fixed. With a **2-shaft** engine, however, reducing UW involves a reduction in compressor speed and hence in airflow, pressure ratio and temperature rise. The compressor power needed is therefore appreciably lower than for the 1-shaft engine. Also, it is evident from *Fig. 3.107* that the compressor operates over a smaller range of efficiency in a 2-shaft engine. For these reasons the part-load fuel consumption of a 2-shaft engine is superior when driving a constant speed load.

The two types also have different characteristics regarding the supply of waste heat to a cogeneration plant, primarily due to the differences in exhaust flow as load is reduced. The essentially constant airflow and compressor power in a 1-shaft unit results in a larger decrease of exhaust temperature for a given reduction in power, which might necessitate the burning of supplementary fuel in the waste heat boiler under operating conditions. This would be unnecessary with a 2-shaft.

In both cases, the exhaust temperature may be increased by the use of variable inlet guide vanes.

The better performance of the 1-shaft two spools engine when ambient temperature is varied or when the plant is installed at altitudes other than 0m above sea level san be observed from the curves. In part load, not surprisingly the scheme including a free power turbine behaved better. It is well known that in the case of the 1 shaft machine, because a fixed speed is determined by the electrica1 generator, reduction in power output results in no appreciable reduction in the compressor work, and hence therma1 efficiency decreases rapidly. In the case of the free power turbine however, a reduction in power output involves a reduction in the speed of the compressor of the gas generator and thus mass flow, pressure ratio and temperature rise. The compressor work is therefore reduced. This leads to a softer reduction of thermal efficiency than in the case of the 1-shaft engine.

The smaller range of TET in the diagrams of 2-shaft GT is due to the restriction of the simulation program, which results from with the assumption that the turbines are choked. Nevertheless, the above conclusions are obvious even in that range of TETs.

For **recuperated gas turbines**, comparative part load performance analyses have been carried out considering different engine configurations and various operation strategies. The main findings of this work are summarized as follows:

- The part load efficiency for a simple operation (fuel only control) is far lower than that of the simple cycle having equivalent design efficiency.
- Design with a higher pressure ratio at a given TET leads to higher part load efficiency and provides additiona1 options for operation strategy, maintaining the recuperator material capability.
- As the design TET increases, the relative part load efficiency becomes higher. The design pressure ratio for advanced high temperature recuperated gas turbines must be considerably higher than that of optima1 therma1 efficiency. This leads to several advantages such as lower TET (low-temperature recuperator material), much larger specific power and far less efficiency degradation at part load operation.

Effect of Pressure

The reasons varying the ambient pressure P_α, could be either the weather changes or the inlet pressure drop caused by inlet air filters or/and ducts. The latter is referred here, because it can be simulated in the same way as the first. A common value for inlet loss is between 1 and 2 %. But monitored plant performance often shows that dirty filters can add unexpected performance losses.

In the case of constant TET and T_α, when P_α is reduced then the following occur:

- density reduces
- mass flow reduces
- R_C remains same
- UW reduces, because of the reduced mass flow
- m_f decreases, because the compression process moves to the right on the T-S diagram, to higher entropy while the Ta, TET and Rc are remaining constant. So the HI decreases.
- η_{th} is constant due to mutual reduction in UW and HI.

4. ABSORPTION COOLING MODELLING

4.1 Introduction

During last decade there has been an enormous increase in the demand of cooling systems. Most of them (almost 90%, 2003,[63]) are based on electricity power supply. Recently due to the fact that electricity becomes more and more expensive power source to use, the market turned to others cooling technologies, which are using as primary source, instead of electric power, heat. These systems have high reliability and in some cases are very economically competitive to the classic electric power supply systems.[73], [74], [75]

The classic refrigerant cycle (see paragraph 4.2) is based in the vapour compression (compressor), driven by electric mover. In some cases though, there is an excess of heat power (coming from waste heat from the use of a Gas Turbine or a Steam Turbine, or a large internal combustion engine), or simply, there is available cheap fuel (for example natural gas), which can be burn producing heat. The first technology is called indirect and is proposed for large cooling installations, while the second is called direct fired and suits smaller, mainly domestic systems.

Let us now consider some thermodynamics principles. The Clausius statement of the Second Law of Thermodynamics, state that it is impossible to construct a device, that operates in a cycle, which simply transfers heat from a low temperature heat reservoir to a higher temperature reservoir. In other words, the statement means that it is impossible to transfer heat from cold to hot area, without any outside assistance (work input). **Refrigeration systems** and **heat pumps** provide the work necessary transferring the heat. The difference between the refrigerator and heat pump is one of definition more than the science behind them. The refrigerator system transfer heat from cold to hot region and so doing cooling the cold region (*Fig.4.1a*). The heat pump transfer heat to a high temperature region, from the heat taken from the low temperature region (*Fig.4.1b*).

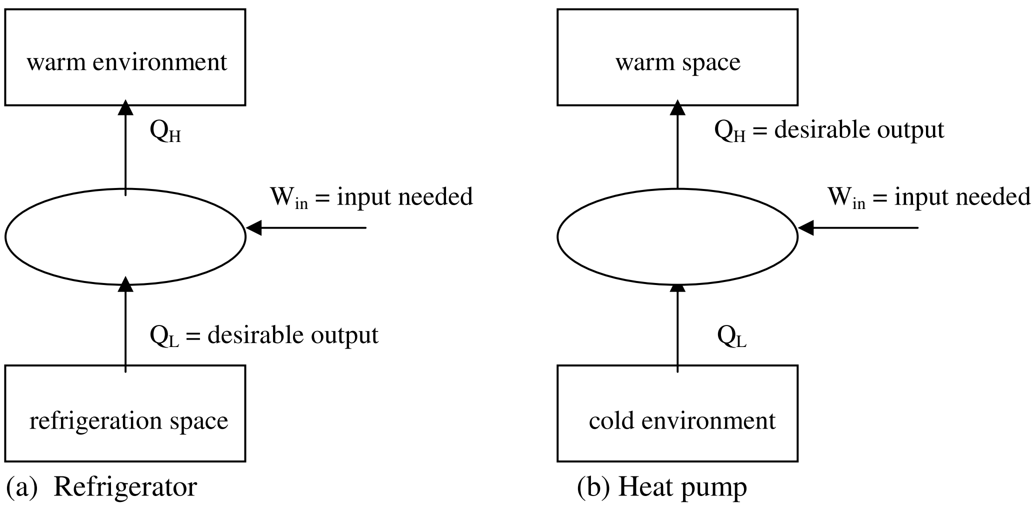


Fig.4.1: Block diagrams showing the operation principal of the refrigerator (a) and the heat pump (b)

The measure of performance of a **refrigeration** system is given in terms of the **coefficient of performance COP_R**, which is defined as

$$COP_R = \frac{Q_L}{W_{in}} \tag{4-1}$$

where Q_L, is the heat t(he refrigeration power) coming out of the space which must be refrigerated, and W_{in}, is the work (the power) which must be added into the cycle.

The refrigeration power usually is defined in **kW** and sometimes in **Tons of Refrigeration (RT)**, which is the ability of the refrigeration system to freeze 1 ton of water (with temperature 0^oC) to ice (with temperature 0^oC) in 24h.

Notation
Power units: 1RT= =12,000BTU/hr=3.5172kW,
Energy units: 1BTU=1.055kJ, 1BTU=0.252kcal,
Enthalpy units: 1BTU/lb=2.3259kJ/kgr.

In the same way the **coefficient of performance COP_{HP} of the heat pump**, is defined as

$$COP_{HP} = \frac{Q_H}{W_{in}} \tag{4-2}$$

where Q_H, is the heat (the heating power) coming into the space which must be heated, and W_{in}, is the work (the power) which must be added into the cycle.

These to coefficients can take prices over unity. A high coefficient of performance is attractive because it shows that a given amount of refrigeration requires only a small amount of work input. [49], [50], [51].

4.2 Refrigeration Carnot cycle

The reversed refrigeration Carnot cycle is showed in *Fig.4.2*. It is called reversed, because it operates anti-clockwise. It takes place inside the two-phase liquid-vapour dome.

The **working fluid (refrigerant or cooling fluid**) absorbs, isothermally, heat Q_L from the space with low temperature T_L (process 1-2), compressed isentropically to stage 3 with an temperature increase to T_H, (process 2-3), rejects isothermally heat Q_H to the space with high temperature T_L (process 3-4) and finally discharged isentropically to the stage 1 with a temperature decrease in T_L. Note that during the process 3-4, the refrigerant is condensed from stage of saturated steam to the stage of saturated liquid.

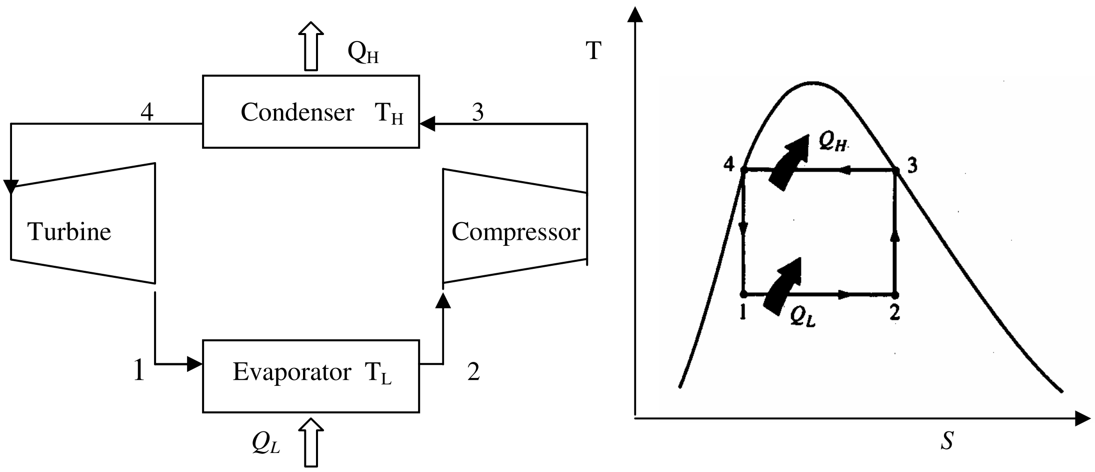


Fig.4.2: The Refrigeration Carnot cycle

The coefficient of performance of the Carnot cycle **$COP_{R,Carnot}$**, is defined as: [50]

$$COP_{R,Carnot} = \frac{Q_L}{W_{in}} = \frac{1}{\frac{T_H}{T_L} - 1} \tag{4-3}$$

It is obvious that the $COP_{R,Carnot}$ is increasing when the temperature deference is decreasing in other works when T_L is increasing or/and T_H is decreasing. The refrigeration Carnot cycle is the most efficient refrigeration cycle, which operates between two temperatures. That is the reason why it is usually used as referring cycle. So it has been defined a coefficient called **refrigeration efficiency**:

$$\eta_{re} = \frac{COP_R}{COP_{R,Carnot}} \tag{4-4}$$

showing, how well the refrigeration cycle is approaching the refrigeration Carnot cycle.

The refrigeration Carnot cycle is not an actual refrigeration cycle. The reason is that the processes 2-3 and 4-1 cannot be approach in reality. And that is so, because in these processes a mixture of liquid-steam is compressed and discharged respectively. [50]

4.3 Vapour compression refrigeration theoretical cycle

Refrigeration could be explained as the cooling of substances or enclosed area to lower temperature than the environment. In order to maintain this constant low temperature, heat must be removed from the refrigerated space. The space, in refrigeration term, where by the low temperature is being maintained is called evaporator. A mixture of liquid and vapour, the refrigerant, evaporates and absorbs heat in its heat of vaporization. Vapour compression refrigeration system is the most generally used and most widely understood system. The schematic of the cycle equipment is depicted in *Fig.4.3.*

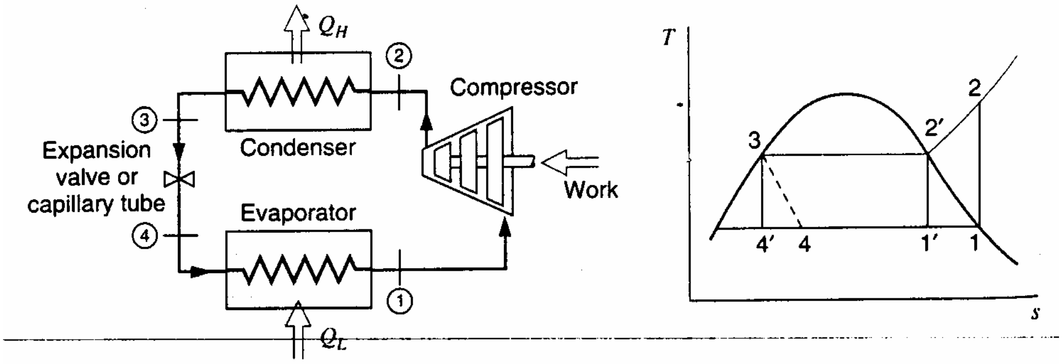


Fig.4.3: The ideal vapour compression refrigeration cycle

In this section, we consider the ideal refrigeration cycle for a working substance that changes phase during the cycle, in a manner equivalent to that done with the Rankine power cycle. In doing so, we note that state **3** in *Fig.4.3* is saturated liquid at the condenser temperature and state **1** is saturated vapour at the evaporator temperature. This means that the expansion process from **3-4** will be in the two-phase region, and mostly liquid. As a consequence, there will be very little work output from this process, such that it is not worth the cost of including this piece of equipment in the system. We therefore replace the turbine with a throttling device, usually a valve or a length of small-diameter tubing, by which the working fluid is throttled from the high- pressure to the low-pressure side. The resulting cycle becomes the ideal model for a vapour-compression refrigeration system, which is shown in *Fig.4.3*. Saturated vapour at low pressure enters the compressor and undergoes a reversible adiabatic compression, process **1-2**. Heat is then rejected at constant pressure in process **2-3**, and the working fluid exits the condenser as saturated liquid. An adiabatic throttling process, **3-4**, follows, and the working fluid is then evaporated at constant pressure, process **4-1**, to complete the cycle.

We also note the difference between this cycle and the ideal Carnot cycle, in which the working fluid always remains inside the two-phase region, 1'-2'-3-4'-1'. It is much more expedient to have a compressor handle only vapour, than a mixture of liquid and vapour, as would be required in process 1'-2' of the Carnot cycle. It is virtually impossible to compress, at a reasonable rate, a mixture such as that represented by state 1' and still maintain equilibrium between liquid and vapour. The other difference is that of replacing the turbine by the throttling process has already been discussed.

Generally there are four main basic components in vapour compression refrigeration; an evaporator, a compressor, a condenser and an expansion valve. In few words, the functions of each of these basic components are:

Compressor: to draw vapour from the evaporator, thus causing a low pressure in the evaporator so that the refrigerant could boil to give the required temperature. This pumping action of the compressor also raises the vapour pressure and delivers it to a condenser where the vapour is being condensed by the cooling fluid (water or air). The compressor in a small and medium-sized refrigeration system is usually the positive displacement machine such as reciprocating, rotary screw etc. The centrifugal compressors are mostly used in larger systems refrigeration; these centrifugal compressors are mostly known in refrigeration industries as 'turbo-compressors'.

Evaporator: to vaporize the working fluid with best possible heat transfer coefficient and also prevent carrying over the liquid droplets through the suction valve to the compressors. This is the part where by the useful cooling is accomplished by lowering the temperature of the refrigerant to below the surrounding temperature. In home refrigerators the evaporator section is the food compartment and its surrounding coils, which contain the working fluid.

Condenser: the main function of the condenser is to transfer heat from the refrigerant to the coolant fluid (water, air, etc.). The refrigerant leaves the compressor with heat from the evaporator, heat of cooling from the condenser to the evaporator, compression heat and heat in the suction line and chambers. The water-cooled condenser is mainly used in the large refrigeration cycle while the air-cooled one is used in smaller units such as in the household.

Expansion Valve: to maintain the pressure difference between the condenser and the evaporator. Also, it is used to control the flow rate from the condenser to the evaporator.

Referring to the *Fig.4.3*, the space under the curve 4-1 is representing the absorbed heat of the refrigerant in the evaporator, while the space under the curve 2-3 is representing the rejected heat in the condenser. It is known that, the coefficient of performance is improved at about 2-4% for every degree of the evaporation temperature that increases or the condenser's temperature that decreases. [51], [54].

4.4 Vapour compression refrigeration actual cycle

The actual refrigeration cycle deviates from the ideal cycle primarily because of pressure drops associated with fluid flow and heat transfer to or from the surroundings. The actual cycle might approach the one shown in *Fig.4.4*.

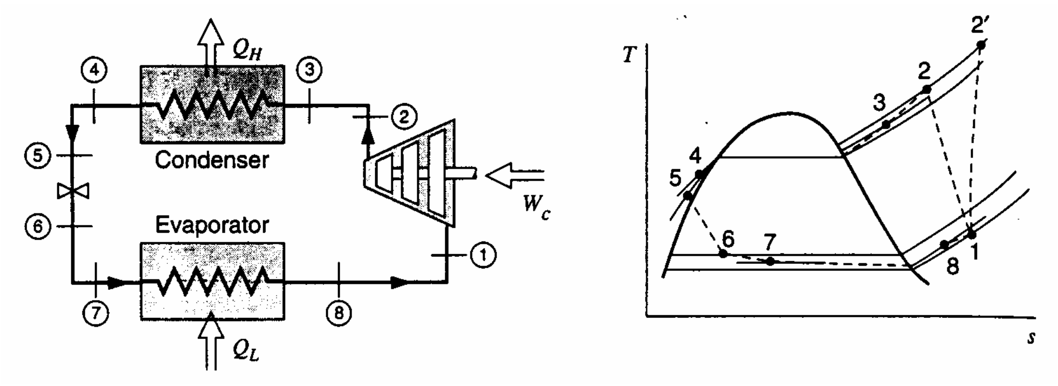


Fig.4.4: The actual vapour refrigeration cycle

The vapour entering the compressor will probably be superheated. During the compression process there are irreversibilities and heat transfer either to or from the surroundings, depending on the temperature of the refrigerant and the surroundings. Therefore, the entropy might increase or decrease during this process, for the irreversibility and the heat transferred to the refrigerant cause an increase in entropy, and the heat transferred from the refrigerant causes a decrease in entropy. The two dashed lines 1-2 and 1-2' represent these possibilities. The pressure of the liquid leaving the condenser will be less than the pressure of the vapour entering, and the temperature of the refrigerant in the condenser will be somewhat higher than that of the surroundings to which heat is being transferred. Usually the temperature of the liquid leaving the condenser is lower than the saturation temperature. It might drop somewhat more in the piping between the condenser and expansion valve. This represents a gain, however, because as a result of this heat transfer the refrigerant enters the evaporator with a lower enthalpy, which permits more heat to be transferred to the refrigerant in the evaporator.

There is some drop in pressure as the refrigerant flows through the evaporator. It may be slightly superheated as it leaves the evaporator, and through heat transferred from the surroundings its temperature will increase in the piping between the evaporator and the compressor. This heat transfer represents a loss, because it increases the work of the compressor, since the fluid entering it has an increased specific volume. [54]

4.5 Working fluids for vapour compression refrigeration systems

A large number of different working fluids (refrigerants) is utilized in vapour-compression refrigeration systems than in vapour power cycles. **Ammonia** and **sulfur dioxide** were important in the early days of vapour - compression refrigeration, but both are highly toxic and therefore dangerous substances. For many years now, the principal refrigerants have been the halogenated hydrocarbons, which are marketed under the trade names of **Freon** and **Genatron**. For example, dichlorodifluoromethane (CCl_2F_2) is known as Freon-12 and Genatron-12, and therefore as refrigerant-12 or R-12. This group of substances, known commonly as **chlorofluorocarbons or CFCs**, is chemically very stable at ambient temperature, especially those lacking any hydrogen atoms. This characteristic is necessary for a refrigerants working fluid. This same characteristic, however, has devastating consequences if the gas, having leaked from an appliance into the atmosphere, spends many years slowly diffusing upward into the stratosphere. There it is broken down, releasing chlorine, which destroys the protective ozone layer of the stratosphere. It is therefore of overwhelming importance to us all to eliminate completely the widely used but life-threatening CFCs, particularly R-11 and R-b 12, and to develop suitable and acceptable replacements. The CFCs containing hydrogen (often termed **HCFCs**), such as R-22, have shorter atmospheric lifetimes, and therefore are not as likely to reach the stratosphere before being broken up and rendered harmless. The most desirable fluids, called **HFCs**, contain no chlorine atoms at all.

There are two important considerations when selecting refrigerant working fluids: the temperature at which refrigeration is needed and the type of equipment to be used.

As the refrigerant undergoes a change of phase during the heat transfer process, the pressure of the refrigerant will be the saturation pressure during the heat supply and heat rejection processes. Low pressures mean large specific volumes and correspondingly large equipment. High pressures mean smaller equipment, but it must be designed to withstand higher pressure. In particular, the pressures should be well below the critical pressure. For extremely low temperature applications a binary fluid system may be used by cascading two separate systems.

The type of compressor used has a particular bearing on the refrigerant. Reciprocating compressors are best adapted to low specific volumes, which means higher pressures, whereas centrifugal compressors are most suitable for low pressures and high specific volumes.

It is also important that the refrigerants used in domestic appliances should be non-toxic. Other beneficial characteristics, in addition to being environmentally acceptable, are miscibility with compressor oil, dielectric strength, stability, and low cost. Refrigerants, however, have an unfortunate tendency to cause corrosion. For given temperatures during evaporation and condensation, not all refrigerants have the same coefficient of performance for the ideal cycle. It is, of course, desirable to use the refrigerant with the highest coefficient of performance, other factors permitting. [54]

4.6 Vapour Absorption Refrigeration Cycle

4.6.1 Overview

All the power plants are using coal, natural gas, oil or biomass, as burning fuel in order to produce power. In all cases, large amounts heat is produced. On the other hand, most of industrial process uses a lot of thermal energy by burning fossil fuel to produce steam or heat. After the processes, heat is rejected to the surrounding as waste. This waste heat can be converted to useful refrigeration by using a heat operated refrigeration system, such as an **absorption refrigeration cycle**.

Electricity purchased from utility companies for conventional vapour compression refrigerators can be reduced. The use of heat-operated refrigeration systems help to reduce problems related to global environmental, such as the so-called greenhouse effect from CO_2 emission from the combustion of fossil fuel in utility power plants.

Another difference between absorption systems and conventional vapour compression systems is the working fluid used. Most vapour compression systems commonly use chlorofluorocarbon refrigerants (CFCs), because of their thermophysical properties. It is though, the restricted use of CFCs, due to depletion of the ozone layer that will make absorption systems more prominent. However, although absorption systems seem to provide many advantages, vapour compression systems still dominates all market sectors. In order to promote the use of absorption systems, further development is required to improve their performance and reduce their cost.

The early development of an absorption cycle dates back to the 1700's. It was known that ice could be produced by an evaporation of pure water from a vessel contained within an evacuated container in the presence of sulfuric acid. In 1810, ice could be made from water in a vessel, which was connected to another vessel containing sulfuric acid. As the acid absorbed water vapour, causing a reduction of temperature, layers of ice were formed on the water surface. The major problems of this system were corrosion and leakage of air into the vacuum vessel. In 1859, Ferdinand Caue introduced a novel machine using water/ammonia as the working fluid. This machine took out a US patent in 1860. Machines based on this patent were used to make ice and store food. It was used as a basic design in the early age of refrigeration development.

In the 1950's, a system using lithium bromide/water as the working fluid was introduced for industrial applications. A few years later, a double-effect absorption system was introduced and has been used as an industrial standard for a high performance heat-operated refrigeration cycle. [64]

Nowadays, this technology is used in Europe, but is more common in the USA and Japan, where much has been done to improve its performance. Absorption chillers use heat as primary energy to produce cold, instead of mechanical rotation work for compression chillers. They can use the heat of steam, hot water or direct gas combustion, depending on technologies. There are various possibilities of use. They can be integrated in a steam, hot water or gas district network. This is the main district cooling policy in Germany and in Japan. They can also be used in industrial processes. Their application is optimal when low-grade heat is available, under conditions where a steam turbine cannot be driven. For example, two steam absorption chillers with lithium bromide are used in Paris-Orly Airport DCS, driven by steam from the thermoelectric plant. [55]

Commercially proven absorption cooling systems, ranging in size from 3 to 2,500 RTs, are readily available today. These systems come as stand-alone chillers or as chillers with integral heating systems. In addition, absorption heating and cooling systems suitable for residential or commercial use are under development and should be on the market within the next few years.

Gas absorption systems feature several **advantages** over conventional vapour compression electric systems:

1. Lower operating costs (operating with waste heat).
2. No ozone-damaging refrigerants (no use of CFCs or HCFCs).
3. No need for extra electric power (no overcharge of the existing electric power network, especially during the peak hours).
4. Lower-pressure systems with no large rotating components.
5. Low maintenance.
6. Safer operation.
7. High reliability.
8. Smaller total space requirements compared to an electric chiller with separate boiler.
9. Long lifetime, (25-30 years, compare to the 10-15 years of the vapour compression systems).
10. Silent operation. Except for two hermetically sealed pumps, absorption chillers do not have any moving parts. They run more quietly (there are few vibrations) than compression chillers. This difference could be significant in office buildings, hotels or hospitals.
11. Potential financial support from National Government, EU, etc.

The basic operating principle of an absorption chiller (see paragraph 4.6.2) is the same with that of a conventional vapour compression chiller, namely, cooling is provided by evaporating a refrigerant. However, large absorption systems are different in that they:

- use water rather than standard refrigerants
- operate at low pressure/vacuum conditions, rather than at moderate to high pressure
- use heat rather than a compression energy as their driving force

All water-cooled absorption systems on the market today, use **water as the refrigerant** and **a lithium bromide solution as the absorbent material** and they used for medium and large scale applications (3-2,500RTs or 10-9,000kW), while the COP_R is between 0.6 and 1,3. A typical air-cooled absorption chiller uses **ammonia as the refrigerant** and **water as the absorbent material** and they used for rather small applications (3-30RTs or 10-100kW), while the COP_R is between 0.6 and 0.7. [67],[72],[73],[74],[75]

Steam fired absorption is used today where there is a low cost of steam such as a cogen or waste energy plant. In the case of direct-fired units, electricity must be at a high cost or there must be a CFC refrigerant or other environmental issue. Larger tonnages (above 500RTs) have a more favourable first cost when compared to electric technologies, so unit must be big. They maybe also are used in places like campuses with a central steam loop and not enough electrical power distribution to run decentralize electric chillers. This may be the case where buildings either did not have chillers or used older single-effect absorption units and have upgraded to double-effect or direct fired technology.

However, gas absorption systems have three important **disadvantages** [59], [66], [72], [73], [74], [75]:

1. Low COP_R, the usual range for a absorption chillers is 0.6-1.3 depending to the technology used, instead of the 3.5-5.5 of the vapour compression systems.
2. In cases where there is no waste heat available, absorption chillers cost more to operate than electric chillers. They also cost about 50% more to purchase.
3. Water consumption in cooling tower.

4.6.2 Principle of operation

The working fluid in an absorption refrigeration system is a binary solution consisting of refrigerant and absorbent. In *Fig.4.5(a)*, two evacuated vessels are connected to each other. The left vessel contains liquid refrigerant while the right vessel contains a binary solution of absorbent/refrigerant. The solution in the right vessel will absorb refrigerant vapour from the left vessel causing pressure to reduce. While the refrigerant vapour is being absorbed, the temperature of the remaining refrigerant will reduce as a result of its vaporization. This causes a refrigeration effect to occur inside the left vessel. At the same time, solution inside the right vessel becomes more dilute because of the higher content of refrigerant absorbed. This is called the "absorption process". Normally, the absorption process is an exothermic process; therefore, it must reject heat out to the surrounding in order to maintain its absorption capability.

Whenever the solution cannot continue with the absorption process because of saturation of the refrigerant, the refrigerant must be separated out from the diluted solution. Heat is normally the key for this separation process. It is applied to the right vessel in order to dry the refrigerant from the solution as shown in *Fig.4.5(b).* Transferring heat to the surroundings will condense the refrigerant vapour. With these processes, using heat energy can produce the refrigeration effect.

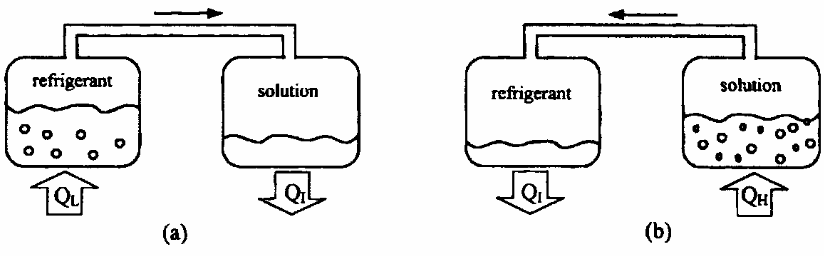


Fig.4.5: (a) Absorption process occurs in right vessel causing cooling effect in the other; (b) Refrigerant separation process occurs in the right vessel as a result of additional heat from outside heat source.

However, the cooling effect cannot be produced continuously as the process cannot be done simultaneously. Therefore, an absorption refrigeration cycle is a combination of these two processes as shown in *Fig.4.6.* As the separation process occurs at a higher pressure than the absorption process, a circulation pump is required to circulate the solution.

Coefficient of Performance of an absorption refrigeration system is obtained from:

$$COP_{RA} = \frac{\text{cooling capacilty obtained at evaporator}}{\text{heat input for the generator + work input from the pump}} \qquad (4\text{-}5)$$

The work input for the pump is almost negligible relative to the heat input at the generator; therefore, the pump work is often neglected for the purposes of analysis. (see paragraph 4.6.4) [61]

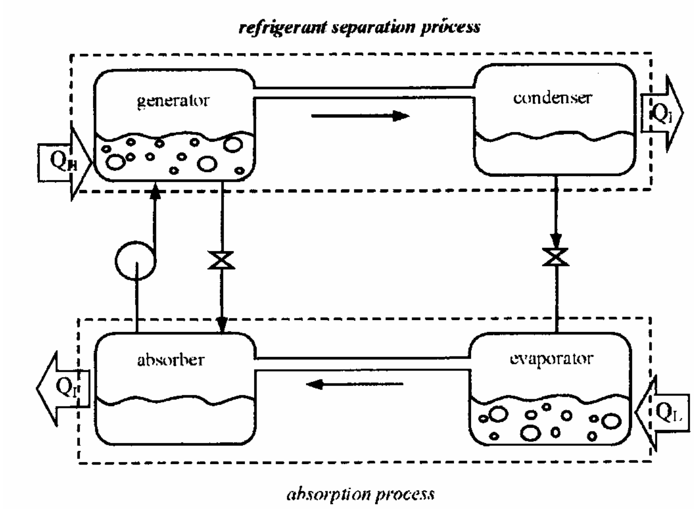


Fig.4.6: A continuous absorption refrigeration cycle composes of two processes

4.6.3 Working fluid for absorption refrigeration systems

Performance of an absorption refrigeration system is critically dependent on the chemical and thermodynamic properties of the working fluid. Few solutions work as suitable absorbent-refrigerant pairs. The materials that make up the refrigerant-absorbent pair should meet the following **requirements** to be suitable for absorption refrigeration:

- A fundamental requirement of absorbent-refrigerant combination is that, in liquid phase, they must have a margin of miscibility within the operating temperature range of the cycle.
- Absence of solid phase. The refrigerant-absorbent pair should not form a solid phase over the range of composition and temperature to which it might be subjected. If a solid forms, it presumably would stop flow and cause equipment to shut down.
- The elevation of boiling (the difference in boiling point between the pure refrigerant and the mixture at the same pressure) should be as large as possible.
- Refrigerant should have high heat of vaporization and high concentration within the absorbent in order to maintain low circulation rate between the generator and the absorber per unit of cooling capacity.
- Affinity. The absorbent should have a strong affinity for the refrigerant under conditions in which absorption takes place. This affinity causes a negative deviation from Raoult's law and results in an activity coefficient of less than unity for the refrigerant; allows less absorbent to be circulated for the same refrigerating effect so

sensible heat losses are less; and (3) requires a smaller liquid heat exchanger to transfer heat from the absorbent to the pressurized refrigerant-absorbent solution. However, calculations by Jacob et al. (1969) indicate that strong affinity has some disadvantages. This affinity is associated with a high heat of dilution; consequently, extra heat is required in the generator to separate the refrigerant from the absorbent.

- Pressure. Operating pressures, largely established by physical properties of the refrigerant, should be moderate. High pressures require the use of heavy-walled equipment, and significant electrical power may be required to pump the fluids from the low-pressure side to the high-pressure side. Low pressure (vacuum) requires the use of large volume equipment and special means of reducing pres- sure drop in refrigerant vapour flow.
- Chemical stability. High chemical stability is required because fluids are subjected to severe conditions over many years of service. Instability could cause the undesirable formation of gases, solids, of corrosive substances.
- Volatility Ratio. The refrigerant should be much more volatile than the absorbent so the two can be separated easily. Otherwise, cost and heat requirements can prohibit separation.
- Latent Heat. The refrigerant's latent heat should be high so the circulation rate of the refrigerant and absorbent can be kept at a minimum.
- Corrosion. Because absorption fluids can corrode materials used in constructing equipment corrosion inhibitors are used.
- Safety. Fluids must be, non-toxic and non-flammable if they are in an occupied dwelling. Industrial process refrigeration is less critical in this respect.
- Transport Properties. Viscosity, surface tension, thermal diffusivity, and mass diffusivity are important characteristics of the refrigerant and absorbent pair. For example, a low fluid viscosity promotes heat and mass transfer and reduces pumping power.
- Environmental friendly. The working pairs must devoid of lasting environmental effects.
- Low cost.

Many working fluids are suggested in literature. A survey of absorption fluids provided by Marcriss [61] suggests that, there are some 40 refrigerant compounds and 200 absorbent compounds available. No known refrigerant-absorbent pair meets all requirements listed. However, LiBr/Water and Water/NH_3 offer excellent thermodynamic performance and they have little long-term environmental effect.

Since the invention of an absorption refrigeration system, **Water/NH_3** has been widely used for both cooling and heating purposes. The Water/NH_3 pair meets most requirements. Both NH_3 (refrigerant) and water (absorbent) are highly stable for a wide range of operating temperature and pressure. NH_3 has a high latent heat of vaporization, which is necessary for efficient performance of the system. It can be used for low temperature applications, as the freezing point of NH_3 is -77°C. Water/NH_3 is environmental friendly and low-cost.
Since both NH_3 and water are volatility, requires high operating pressures and the cycle requires a rectifier (higher installation cost) to strip away water that normally evaporates with NH_3. Without a rectifier, the water would accumulate in the evaporator and offset the system performance. There are other disadvantages such as its high pressure, toxicity, and corrosive action to copper and copper alloy. Furthermore,

ammonia is a Safety Code Group 2 fluid (ASHRAE Stranded 15), which restricts its use indoors. [67]

The use of **LiBr/Water** for absorption refrigeration systems began around 1930 [59]. There are some outstanding features of LiBr/Water: non-volatility absorbent of LiBr (the need of a rectifier is eliminated) and extremely high heat of vaporization of water (refrigerant), high affinity, high stability, high latent heat and high safety

However, this pair tends to form solids. Because the refrigerant turns to ice at 0^oC, the pair cannot be used for low temperature refrigeration. Lithium bromide crystallizes at moderate concentrations, especially when it is air cooled, which typically limits the pair to applications where the absorber is water-cooled. However, using a combination of salts as the absorbent can reduce this crystallizing tendency enough to permit air-cooling. It is also corrosive to some metal and expensive. Some additive may be added to LiBr/water as a corrosion inhibitor [56] or to improve heat-mass transfer performance. Other disadvantages of the LiBr/water pair include the low operating pressures it requires and LiBr/water solution's high viscosity. Proper equipment design can overcome these disadvantages.

Other intriguing refrigerant-absorbent pairs include the following: [59]
- Ammonia-salt
- Methylamine-salt
- Alcohol-salt
- Ammonia-organic solvent
- Sulfur dioxide-organic solvent
- Halogenated hydrocarbons-organic solvent.
- Water-alkali nitrate
- Water-hydroxide
- Ammonia-water-salt

Several refrigerant-absorbent pairs appear suitable for certain cycles and may solve some of problems associated with the traditional pairs. However, stability, corrosion, and property information on several of them is limited. Also some of the fluids are somewhat hazardous. [58] [59].

4.6.4 The single-effect LiBr/Water absorption cycle flow description

LiBr/water is used as an absorption working fluid because it is one of the best choices found among hundreds of working fluids that have been considered. The fundamentals of operation of an absorption cycle using aqueous lithium bromide as the working fluid are discussed in this section. To keep the discussion simple, only the most basic cycle is considered.

A block diagram of a single-effect machine is provided as *Fig.4.7*. The diagram is formatted as if it were superimposed on a **Duhring plot** [66] of the working fluid. Thus, the positions of the components indicate the relative temperature, pressure and mass fraction. The cycle has five main components as shown in *Fig.4.7*: the **generator (sometimes called desorber)**, **the condenser**, **the evaporator, the absorber**, and the **solution heat exchanger**.

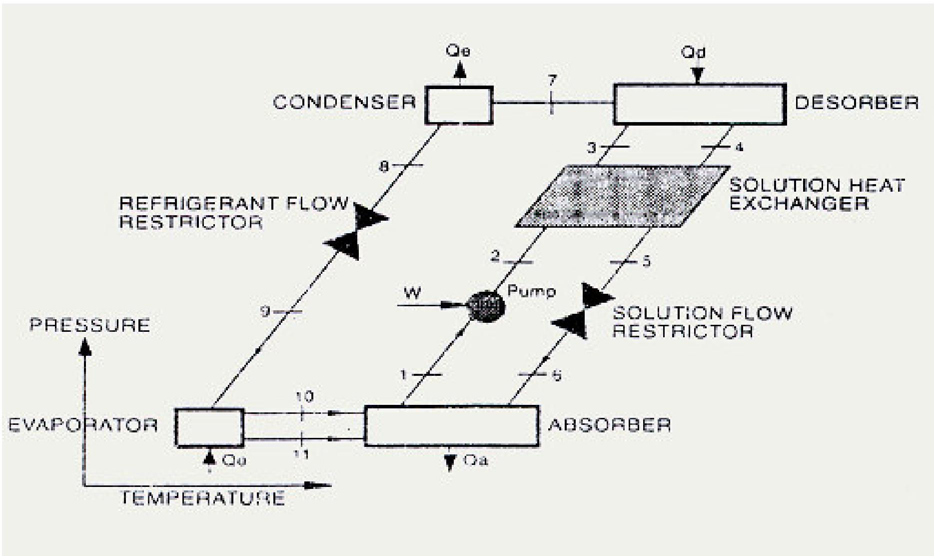


Fig.4.7: Single-effect LiBr/Water absorption cycle [66]

Starting with state point **4** at the generator exit, the stream consists of absorbent-refrigerant solution, which flows to the absorber via the heat exchanger. From points **6** to **1**, the solution absorbs refrigerant vapour (**10**) from the evaporator and rejects heat, to the environment. The solution rich in refrigerant (**1**) flows via the heat exchanger to the generator (**3**). In the generator thermal energy is added and refrigerant (**7**) boils off the solution. The refrigerant vapour (**7**) flows to the condenser, where heat is rejected as the refrigerant condenses. The condensed liquid (**8**) flows through a flow restrictor to the evaporator. In the evaporator the heat from the load evaporates the refrigerant, which then flows (**10**) to the absorber. A portion of the refrigerant leaving the evaporator leaves as liquid spillover (**11**).

The state points of absorption cycles are usually represented in Dϋhring chart *(Fig.4.8).* In this chart, refrigerant saturation temperature and its corresponding pressure are plotted versus the solution temperature. The lines of constant solution concentration are straight lines of decreasing slopes for increasing concentrations. In this schematic the lines represent constant LiBr/Water concentration, with water as the refrigerant. The solution at the exit of the generator (point **4**) is cooled to point **5** in the heat exchanger. In the absorber, the solution concentration decreases to that of **1**. The solution is then pumped to the generator via a heat exchanger, where its temperature is raised to that of **3**. In the generator the solution is reconcentrated to yield **4** again. The refrigerant from the generator condenses at **8** and evaporates at **10** to return to the absorber.

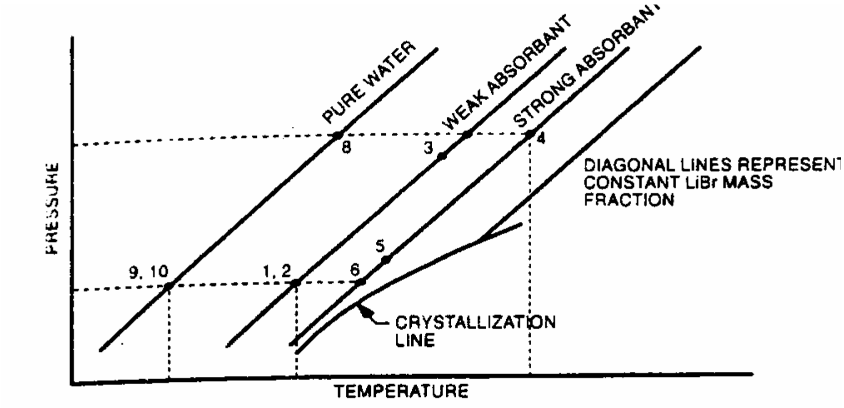


Fig.4.8: Single-Effect LiBr/Water Absorption Superimposed on Dϋhríng Plot [66]

The absorption cycle can be split into two separate circuits:

<u>Solution circuit</u>
The solution circuit circulates between the desorber and absorber. This liquid loop is pumped from the low pressure in the absorber to the high pressure in the desorber. As a first approximation, the entire machine can be considered to operate between two pressure levels.

The liquid solution is pumped into the desorber where heat is supplied by external means such as a combustion source or any other source with a sufficiently high temperature. The required temperature level is governed by the properties of his working fluid and the operation of the other components in the machine. For a typical single-effect LiBr/Water machine **the desorber heat must be supplied above a temperature of approximately 100°C** (this value is a rule of thumb with actual requirements depending on the details of the application). When heat is applied to the solution, the volatile component (i.e. the refrigerant, water) is boiled off.

The power required by the pump is very small, compare to the work required by the compressor in the vapour compression refrigeration system. This is because the specific volume flows of the solution are very small compare to an equal mass of vapour

In dealing with mixtures, the relative volatility of the components is a property of major interest. In the case of LiBr/Water, the salt (LiBr) is essentially non-volatile and the relative volatility is effectively infinite. From a molecular viewpoint, we expect that some salt molecules may escape from the liquid surface and be present in the vapour. However, the escaping tendency is so small under the conditions encountered in an absorption machine that the vapour above the liquid solution is essentially pure water vapour (steam). This fact can be appreciated more fully by realizing that the normal boiling point of solid LiBr/Water salt is 1,282°C. Thus, the vapour pressure of the salt at typical absorption machine conditions is exceedingly low. From a thermodynamic standpoint, we will assume that there is no salt content in the vapour and that the properties of the vapour are those of pure water (i.e., steam). At high vapour velocities, liquid entrainment can also carry salt throughout the machine. Trace salt quantities are important from a corrosion perspective. The presence of trace amounts of salt contributes to accelerated corrosion throughout the vapour space.

When heat is applied to the solution in the desorber, vapour is "generated" and the vapour flows to the condenser. The remaining liquid solution exits the desorber and flows back to the absorber. The process in the desorber is a partial evaporation. Since the vapour leaving the desorber is essentially free of salt, the liquid solution becomes concentrated during the partial evaporation process. Thus, the solution flowing back to the absorber is a relatively concentrated salt solution (compared to that exiting the absorber). A number of terms are in common use to describe the concentrations in absorption systems. In general, the mass fraction is used as a concentration measure in this text. However, it may be of use to the reader to define the commonly used terms. The terms "rich" and "poor" are sometimes used but care must be taken to know to which component these terms refer. When using these terms, one must say, for example, that the solution is "rich in refrigerant". A similar set of terms is "strong" and "weak".

The concentrated salt solution leaving the desorber passes through a solution heat exchanger and exchanges energy with the solution leaving the absorber. This heat exchange process occurs between two liquid streams and involves only sensible heat (no phase change occurs in this device under normal conditions). The purpose of this internal heat exchange device is to reduce the external heat input requirement by utilizing energy available within the machine that would otherwise be wasted. By including the solution heat exchanger, the quantity of rejected heat is also reduced. Thus, the solution heat exchanger is a key component; the performance of this component has a major impact on the design of an absorption machine.

The solution stream leaving the desorber returns to the absorber. The stream gives up energy in the solution heat exchanger and typically arrives at the flow restrictor sub cooled. As the liquid is throttled through the restrictor, some vapour usually evolves from the liquid. The two-phase stream then enters the absorber. In the absorber, the concentrated salt solution is brought into contact with vapour supplied by the evaporator. When the absorber is being cooled by an external sink (for example, a flow from a cooling tower) the absorption process occurs. As the vapour is absorbed, the liquid mass fraction is reduced to the level of the desorber input. Since vapour is absorbed into the solution, the mass flow rate of liquid leaving the absorber is greater than that of the liquid entering the absorber. The reverse is true for the desorber.

Refrigerant circuit

The refrigerant circuit of an absorption machine is identical in function to the corresponding components in a vapour compression machine. The refrigerant loop takes the refrigerant vapour from the desorber and directs it to the condenser where it is liquified by rejecting heat to a sink. In a typical installation, the absorber and the condenser would reject heat to the same sink (i.e., approximately the same temperature level). The subcooled liquid leaving the condenser is throttled through the restrictor to the low pressure. This throttling process is typically accompanied by some vapour flashing. However, due to the high latent heat of water, the vapour quality leaving the restrictor is relatively low as compared to common refrigerants used in vapour compression systems. The two-phase refrigerant then enters the evaporator. Evaporation takes place, accompanied by heat transfer from the evaporator environment, due to the low pressure created by the absorber. Complete evaporation then implies that all of the refrigerant flow arrives at the absorber as vapour.

4.6.5 Crystallization and vacuum requirements

The nature of salt solutions, such as LiBr/Water, is that the salt component precipitates when the mass fraction of salt exceeds the solubility limit. The solubility limit is a strong function of mass fraction and temperature and a weak function of pressure. Furthermore, crystal nucleation is a process sensitive to the presence of nucleation sites. If no suitable nucleation sites are present, supersaturation can occur where the salt content of the liquid is greater than the solubility limit. Once crystals begin to form, the crystals themselves provide favorable nucleation sites and the crystals grow on themselves. The phenomenon of precipitation of salt from an aqueous solution can be readily observed by preparing a solution of **0.70 mass fraction LiBr**. It is highly desirable to avoid crystallization events. Thus, manufacturers generally include controls that sense the possibility of crystallization and take appropriate action to avoid the condition, by reducing heat input to the desorber or by diverting liquid water from the evaporator to the absorber and thus diluting the solution.

Typical pressures in a LiBr absorption machine are sub atmospheric. The pressures are determined by the vapour pressure characteristics of the working fluids. Since essentially pure water exists in the condenser and evaporator, the temperature of operation of these components defines the pressure. For an evaporator temperature of 5°C approximately, the corresponding vapour pressure of water is 0.872kPa or approximately 0.009atm. [59] [66]

4.6.6 Thermodynamic simulation of the single-effect LiBr/Water absorption cycle

A single-effect, absorption cycle using LiBr/Water as the working fluid is perhaps the simplest manifestation of absorption technology. A schematic of such a cycle is provided in *Fig.4.9*. The major components are labelled and the state points in the connecting lines are assigned state point numbers. The schematic shows the energy transfers external to the cycle as arrows in the direction of transfer with variable names representing the four heat transfers and one work term. The schematic is drawn as if it were superimposed on a Duhring chart of the working fluid properties as indicated by the coordinates shown in the lower left-hand corner. The relative position of the components with phase change in the schematic indicates the relative temperature and pressure of the working fluid inside those components.

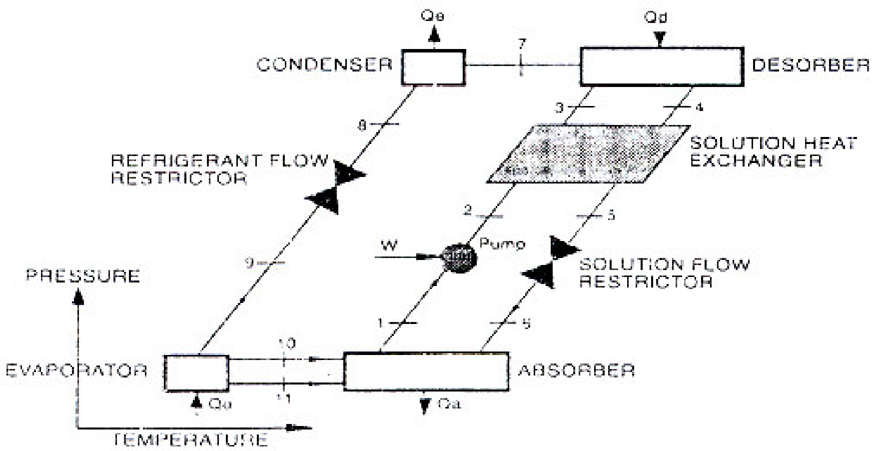


Fig.4.9: Single-effect LiBr/Water absorption chiller [66]

The exception to this is the subcooled and superheated states, which cannot be accurately represented on a Duhring chart which displays only saturated states. The limits of operating conditions for the cycle in *Fig.4.9* are listed in *Table 4.1*. These sets of conditions actually represent the input file for the simulation program (APPENDIX D.5), which will be analyzed below.

The thermodynamic state of each of the points within the cycle must be understood to properly understand the cycle. A summary of the state point descriptions is listed in *Table 4.2*. As listed in the table, four of the points are saturated liquid (3, 4, 8 and 11), one is saturated vapour (10), three are subcooled liquid (2, 3, and 5), one is superheated vapour (7), and two are two- phase vapour-liquid states for a total of 10 state points.

The vapour quality is assumed for four state points. Those are the three saturated liquid states and the saturated vapour state. These assumptions are made for convenience in modelling. In a real machine, the conditions at these points would not be exactly saturated. In general, the transfer processes within the components require a finite driving potential between the vapour and liquid phases. A saturated outlet condition would imply a zero potential difference at the outlet. This does not occur in practice. However, it has been found that this assumption does not introduce a large error and it is typical of a first order model of a cycle. In a real machine, the liquid streams would be expected to be subcooled and the vapour stream would be superheated. These states can also be modelled, but since additional data are then needed, they introduce more complication.

Table 4.1: Input envelop file (limits) of the Single-effect LiBr/Water simulation program

Capacity	$\dot{Q}_e$ (kW)	300 ÷ 2,000
Strong solution mass fraction	x_4	0.6 ÷ 0.65
Solution mass fraction difference	Dx	0.04 ÷ 0.055
Weak solution heat exchanger entry temperature	t_2 (°C)	280 ÷ 45
Temperature difference between strong solution outlet and weak solution inlet of the solution heat exchanger	Dt (°C)	10 ÷ 20
Desorber solution exit temperature	t_4 (°C)	80 ÷ 120
Desorber vapour exit temperature	t_7 (°C)	t_4-2 if t_4= 120 ÷ 110 t_4-4 if t_4= 110 ÷ 100 t_4-6 if t_4= 100 ÷ 90 t_4-8 if t_4= 90 ÷ 80
Evaporator temperature	t_{10} (°C)	5 ÷ 7
Mass percentage of the liquid carryover from evaporator	Dm	0.02 ÷ 0.03
Low pressure	p_1 (kPa)	0.86 ÷ 1.05
High pressure	p_2 (kPa)	8.6 ÷ 10.5
Pump efficiency	n_{pump}	0.9 ÷ 0.99

The state at the vapour outlet from the desorber (point 7) is specified as superheated water vapour (steam) based on the perspective that the stream is pure water. However, it is also possible to view the steam as the vapour component of a two-phase system where the solution in the desorber is the liquid phase. From this binary mixture perspective, the vapour is saturated. These two perspectives are both correct and they are both useful depending on the type of analysis being performed. This point is emphasized here since it is possible to generalize the assumption about the outlet state of the working fluid from each of the components. For this introductory mode!, saturated solution conditions are assumed at the outlet of each of the four major components (desorber, absorber, condenser and evaporator).

The outlet states from the expansion valves are determined by applying an energy balance to the valve assuming an adiabatic expansion. It should be noted that the state point data for these states (points 6 and 9) listed in Table 4.2 represent the overall two-phase state. Thus, the enthalpy and mass flow rate values listed are for the overall two-phase flow at that point.

At point 9, approximately 6.5% of the mass flow flashes into steam. Due to the substantial changes in volume that occur at such a low pressure, the flash gas significantly impacts the design of a refrigerant expansion device for this application. The amount of vapour that flashes at point 6 is much smaller for this example (only 0.1 %) as a result of the significant subcooling that occurs in the solution heat exchanger. A change in the performance of the solution heat exchanger can cause more or less flash gas at point 6. As at point 9, the flash gas has a very high specific volume and causes the velocity of the two-phase stream at 6 to be significantly greater than the velocity at point 5.

The temperature drop that occurs across each of the expansion valves occurs because the vapour has a higher internal energy than the liquid. Thus, some energy must be extracted from the liquid to drive the phase change. The process attains its own equilibrium at a temperature below the starting temperature. The magnitude of the temperature drop correlates with the amount of vapour, which flashes. The flow restrictors are assumed adiabatic.

Finally is assumed that no pressure changes expect through the flow restrictors and the pump.

Table 4.2: Thermodynamic state point summary

Point	State	Notes
1	Saturated liquid solution	Vapour quality set to 0 as assumption
2	Subcooled liquid solution	State calculated from pump model
3	Subcooled liquid solution	State calculated from solution heat exchanger model
4	Saturated liquid solution	Vapour quality set to 0 as assumption
5	Subcooled liquid solution	State calculated from solution heat exchanger model
6	Vapour-liquid solution state	Vapour flashes as liquid passes through expansion valve
7	Superheated water vapour	Assumed to have zero salt content
8	Saturated liquid water	Vapour quality set to 0 as assumption
9	Vapour-liquid water state	Vapour flashes as liquid passes through expansion valve
10	Saturated water vapour	Vapour quality set to 1.0 as assumption
11	Saturated liquid water	Vapour quality set to 0 as assumption

Based on the assumptions and inputs listed in *Table 4.2* the absorption cycle calculations are as follows.

Pressure situation:
The two levels of pressure are due to the existence of the solution pump. The indexes represent points on *Fig.4.9*,

$$p_1 = p_6 = p_9 = p_{10} \tag{4-6}$$

$$p_1 = p_6 = p_9 = p_{10} \tag{4-7}$$

Mass flow situation:
In the cycle there is no mass loss either for the pure water path, or for the solutions paths. Thus

$$\dot{m}_1 = \dot{m}_2 = \dot{m}_3 \tag{4-8}$$

$$\dot{m}_4 = \dot{m}_5 = \dot{m}_6 \tag{4-9}$$

$$\dot{m}_7 = \dot{m}_8 = \dot{m}_9 \tag{4-10}$$

Lithium bromide mass fraction situation:
In paragraph 4.3.4 the solution circuit has been analysed. In that circuit there are two paths one with weak solution (points1,2,3) and one with strong solution (points 4,5,6), so the solution mass fraction is as follow:

$$x_1 = x_2 = x_3 \tag{4-11}$$

$$x_4 = x_5 = x_6 \tag{4-12}$$

While

$$x_1 = x_4\text{-}Dx \tag{4-13}$$

Finally it is obvious (pure water or pure steam) that

$$x_7 = x_8 = x_9 = x_{10} = x_{11} = 0 \tag{4-14}$$

Throttling devices (expansion valves):
There are two expansion vales, one for the water flow and one for the strong solution. Both of them are consider being adiabatic. So

$$h_5 = h_6 \tag{4-15}$$

$$h_8 = h_9 \tag{4-16}$$

Desorber:
Overall mass balance on absorber

$$\dot{m}_3 = \dot{m}_4 + \dot{m}_7 \tag{4-17}$$

Lithium bromide mass balance on absorber

$$\dot{m}_3 \cdot x_3 = \dot{m}_4 \cdot x_4 \tag{4-18}$$

From equations (4-17), (4-18) the **mass factor c**, and the **solution circulation factor f**, can be calculated:

$$c = \frac{\dot{m}_4}{\dot{m}_7} = \frac{x_3}{x_4 - x_3} \tag{4-19}$$

$$f = \frac{\dot{m}_3}{\dot{m}_7} = \frac{x_4}{x_4 - x_3} \tag{4-20}$$

These two factors are characteristic of every absorption machine.
The energy balance on the desorber determines the heat input required to drive the machine as

$$\dot{Q}_d = \dot{m}_4 \cdot h_4 + \dot{m}_7 \cdot h_7 - \dot{m}_3 \cdot h_3 \tag{4-21}$$

The enthalpy at point 7 (h_7) can be determinate from *Table D.1* (APPENDIX D.1), knowing the temperature t_7 and assuming saturated gas at point 7.

Solution Heat Exchanger:
The solution heat exchanger transfers heat from the high temperature solution stream to the low temperature solution stream. The energy balance is written assuming an adiabatic shell.
Energy balance on solution heat exchanger

$$\dot{m}_2 \cdot h_2 + \dot{m}_4 \cdot h_4 = \dot{m}_3 \cdot h_3 + \dot{m}_5 \cdot h_5 \Rightarrow$$

$$\Rightarrow h_3 = h_2 + \frac{\dot{m}_4}{\dot{m}_3} \cdot \left(h_4 - h_5\right) \tag{4-22}$$

The mass fraction of the equation (4-22), with the help of (4.19) and (4.20) can be written

$$d = \frac{\dot{m}_4}{\dot{m}_3} = \frac{x_3}{x_4} \tag{4-23}$$

The enthalpy at point 2 (h_2) can be determinate from *Fig.4.10* or from *equation D-1* (APPENDIX D.2), knowing the temperature t_2 and mass fraction x_2.
The enthalpy at point 4 (h_4) can be determinate from *Fig.4.10* or from *equation D-1* (APPENDIX D.2), knowing the temperature t_4 and mass fraction x_4.

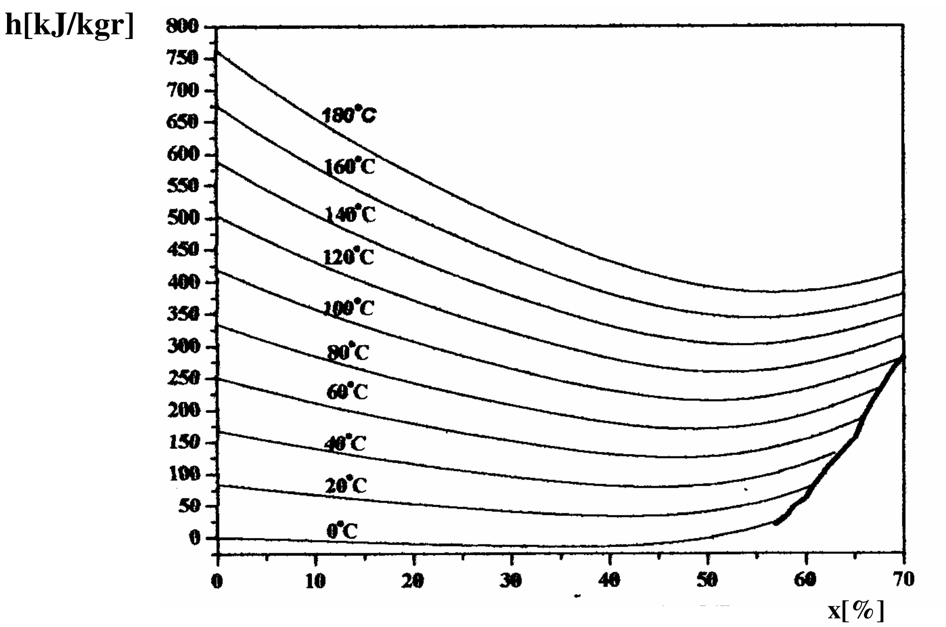


Fig.4.10: Enthalpy-concentration diagram for Lithium Bromide-Water combination [60]

It is also known that

$$t_5 = t_2 + Dt \tag{4-24}$$

The enthalpy at point 5 (h_5) can be determinate from *Fig.4.10* or from *equation D-1* (APPENDIX D.2), knowing the temperature t_5 and mass fraction x_5.

After calculating the enthalpy h_3 from (4-22), the temperature t_3 at point 3, can be determinate from *Fig.4.10* or from *equation D-7.* (APPENDIX D.4), also knowing the mass fraction x_3.

Evaporator:

Energy balance on evaporator

$$\dot{Q}_e = \dot{m}_{10} \cdot h_{10} + \dot{m}_{11} \cdot h_{11} - \dot{m}_9 \cdot h_9 \Rightarrow$$

$$\dot{m}_{10} = \frac{\dot{Q}_e}{h_{10} + Dm \cdot (h_{11} - h_9) - h_9} \tag{4-25}$$

It is assumed that

$$t_9 = t_{10} = t_{11} \tag{4-26}$$

The enthalpy at point 9 (h_9) is equal with the enthalpy h_8 (4-16).

The enthalpy h_8 and the temperature t_8 at point 8 can be determinate from *Table D.1* (APPENDIX D.1), knowing the pressure p_8 and assuming saturated water at point 8.

The enthalpy at point 10 (h_{10}) can be determinate from *Table D.1* (APPENDIX D.1), knowing the temperature t_{10} and assuming saturated gas at point 10.

The enthalpy at point 11 (h_{11}) can be determinate from *Table D.1* (APPENDIX D.1), knowing the temperature t_{11} and assuming saturated water at point 11.

It is given that

$$\dot{m}_{11} = Dm \cdot \dot{m}_{10} \tag{4-27}$$

Mass balance on evaporator

$$\dot{m}_9 = \dot{m}_{10} + \dot{m}_{11} \tag{4-28}$$

From (4-19) (4-20) and (4-10), the masses flows $\dot{m}_3$ and $\dot{m}_4$ can be calculated

$$\dot{m}_3 = f \cdot \dot{m}_7 \tag{4-29}$$

$$\dot{m}_4 = c \cdot \dot{m}_7 \tag{4-30}$$

Condenser:
The temperature at point 8 (t_8) can be determinate from *Table D.1* (APPENDIX D.1), knowing the enthalpy h_8 and assuming saturated gas at point 8.

To determine condenser heat (rejected heat to the environment), an energy balance on the condenser is

$$\dot{Q}_c = \dot{m}_{10} \cdot (h_7 - h_8) \tag{4-31}$$

Solution Pump:
The minimum solution pump work is obtained from an pump model is

$$\dot{W} = \dot{m}_1 \cdot v \cdot (p_2 - p_1) \cdot \frac{1}{n_{pump}} \tag{4-32}$$

where v is the specific volume of the solution and is given by 2.

$$v = \frac{1}{\rho} \tag{4-33}$$

where ρ, is the density of the solution and is given by the *Fig.4.11*, or by *equation D*-5 (APPENDIX D.3) and is assumed that does not change appreciably from point 1 to point 2.

The value of $\dot{W}$ is small enough and can be neglected for simplification reasons, without introducing any serious mistake. The value of $\dot{W}$ is very small, because the solution pump is compressing liquid (weak solution LiBr/water), which has very small values of specific volume compare to the refrigerant vapour in the compression cycles.

Actually, that low value of $\dot{W}$ is one of the biggest advantages of the absorption cycle in comparison to the compression cycle.

Absorber
The temperature t_6 at point 6, can be determinate from *Fig.4.10* or from *equation D-7* (APPENDIX D.4), knowing the enthalpy h_6 (4-15) and mass fraction x_6.

The enthalpy at point 1 is given by

$$h_1 = h_2 - \dot{W} \tag{4-34}$$

The temperature t_1 at point 1 can be determinate from *Fig.4.10* or from *equation D-7.* (APPENDIX D.4), knowing the enthalpy h_1 and mass fraction x_1 from (4-34),

The energy balance on the absorber can be written as

$$\dot{Q}_a = \dot{m}_{10} \cdot h_{10} + \dot{m}_{11} \cdot h_{11} + \dot{m}_6 \cdot h_6 - \dot{m}_1 \cdot h_1 \tag{4-35}$$

Note that the assumption that the solution leaving the absorber is saturated is not accurate. Finite driving potentials in the absorber require that the solution leave subcooled. By assuming saturated liquid, the model would be expected to over predict performance but this effect is beyond the scope of the present treatment.

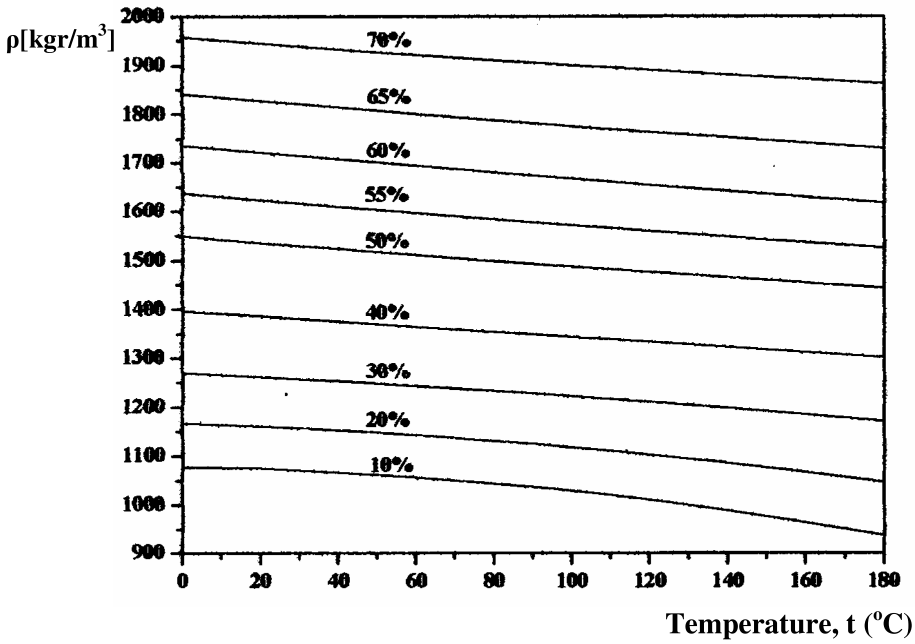


Fig.4.11: Lithium Bromide Water density-temperature diagram for different concentrations [60]

Coefficient of Performance:

As it already said in paragraph 4.1, the typical measure of cycle performance is the coefficient of performance defined as

$$COP_{RA} = \frac{\dot{Q}_e}{\dot{Q}_d + \dot{W}} \tag{4-36}$$

For a given absorption chiller, the above simulation procedure provides results depending only to the heat entering the desorber ($\dot{Q}_d$). The latter depends to EGT and GT's exhaust mass flow. When one of them or both of them is changing then the cooling production capability of the chiller $\dot{Q}_e$ changes proportionally (linearly), see *Fig 4.14*. This means that in contrast the GT performance simulation, there is no special procedure simulates the off design performance of the absorption cooling system.

4.6.7 Heat transfer from the gas turbine exhaust to the absorption chiller

As it has been mentioned in paragraph 4.6.1, the absorption cooling system is a very promising technology when waste heat is used as heating source of the absorption refrigeration cycle. From that point of view, the industrial gas turbine engine is the ideal prime mover, which produces -during its operation- except of the rotating power, hot gases, which are thronged out to the environment. The mass flow of the exhaust gasses is depending on the compressor pressure ratio and on the size on the engine. The mass flow can vary, for example, from 20kgr/sec for micro-turbines, up to 250kgr/sec for heavy-duty gas turbines. The temperature of the exhaust gasses Texh, is depending on TET and obviously on the type of the cycle used. Thus, in the Brayton cycle with reheat the exhaust temperature is higher relatively to the simple cycle due to the existence of the reheater at the back of the engine. In addition, in the case Brayton cycle with heat exchanger, the exhaust gasses are coming out with low temperature, relatively to the simple cycle due to the existence of the heat exchanger (CHAPTER 3). Typical Texh, range is 450K-850K.

In order to transfer heat from the gas turbine exhaust to the desorber of absorption chiller, the use of a heat exchanger is needed. That heat exchanger is called exhaust gas **heat exchanger**. (*Fig.4.12*)

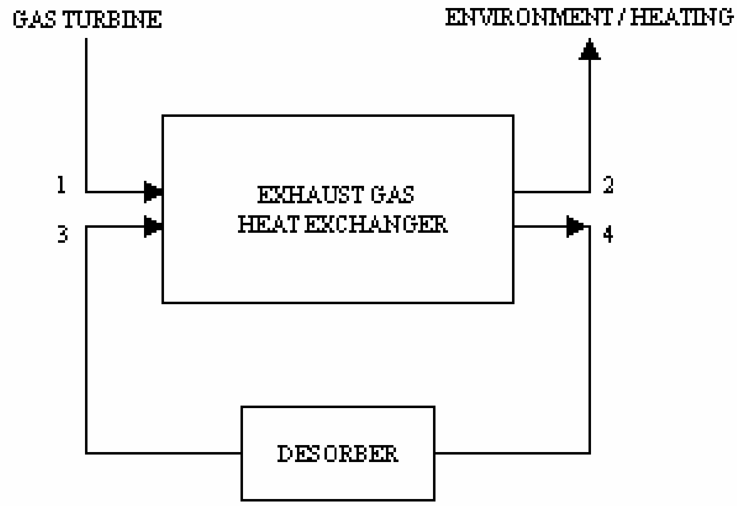


Fig.4.12: Heat transfer system from the gas turbine exhaust to the absorption chiller desorber

The exhaust gas heat exchanger is working with two streams. The “gas turbine stream” consist of exhaust gasses coming from the gas turbine (point 1), getting through the exhaust gas heat exchanger, and end up (point 2), either to the environment as waste heat gasses or to another heat exchanger where their **remaining heat can be used for others purposes (such as dry up), increasing further more the overall efficiency** of the installation. The “desorber stream” is closed cycle, circulating of water coming from the desorber (point 3), getting through the exhaust gas heat exchanger, and coming out as superheated water (point 4) with temperatures between 105-130^{o}C. It is obvious that the second stream is under pressure to reassure that the water remains always in liquid face, and so keeping the exhaust gas heat exchanger efficiency at relatively high level. Typical values of exhaust **gas heat exchanger efficiency (η_{EHE}) are 0.7-0.85**, while it is dependable on the manufacture technology and on the cleanness degree of the “gas turbine stream” internal paths. The mass flow of the circulating water depends on the cooling power capacity of the absorption chiller (8-80kgr/sec). [73], [74], [75]

4.6.8 Single-effect LiBr/Water absorption cycle, simulation program results

The simulation program of the absorption cooling system is carried out using FORTRAN as programming language. It is based on data and equations presented in paragraph 4.6.6 and 4.6.7. The program has constructed to be friendly to the user and also in a sense of independent module easily replaceable.

An example of the absorption cooling simulation program **input file** is presented in APPENDIX D.5. The program has the ability to calculate the enthalpy, the mass flow, the pressure, the temperature and the concentration fraction of the LiBr in every point of the absorption cooling system (*Fig.4.9*). In addition, performance characteristics of the cycle are also calculated. The program can make calculations for variety of cooling power that the user is asking for. In APPENDIX D.6, an example of the absorption cooling simulation program **output file** is presented for Qe: 300-2100kW, with step 300kW.

In the figures below are presented three most important quantities, which characterise the absorption cooling system. It is important to highlight that the results coming from the program was checked with case studies presented to references [59], [66], and the deviations were very small (2-3%), due to accuracy level.

In diagram *Fig.4.13*, COP is plotted versus Q_e. It can be seen that the COP is actually constant when Q_e, varies, having a value of about 0.68 independently of the cooling power.

In diagram *Fig.4.14*, the heat power needed by the desorber Q_d is plotted against the cooling power Q_e. It can be seen that their relationship is, in a sense, linear.

In *Fig.4.15*, the heat power provided by the gas turbine Q_{HGT}, is plotted against the cooling power Q_e, assuming η_{HEH}=0.85. That diagram is actually the connection between the program presented in CHAPTER 3 and the absorption cooling simulation program presented here.

Finally, it should be mentioned that the program can easily work in the opposite way i.e. having as input file Q_d values and calculating the Q_e values.

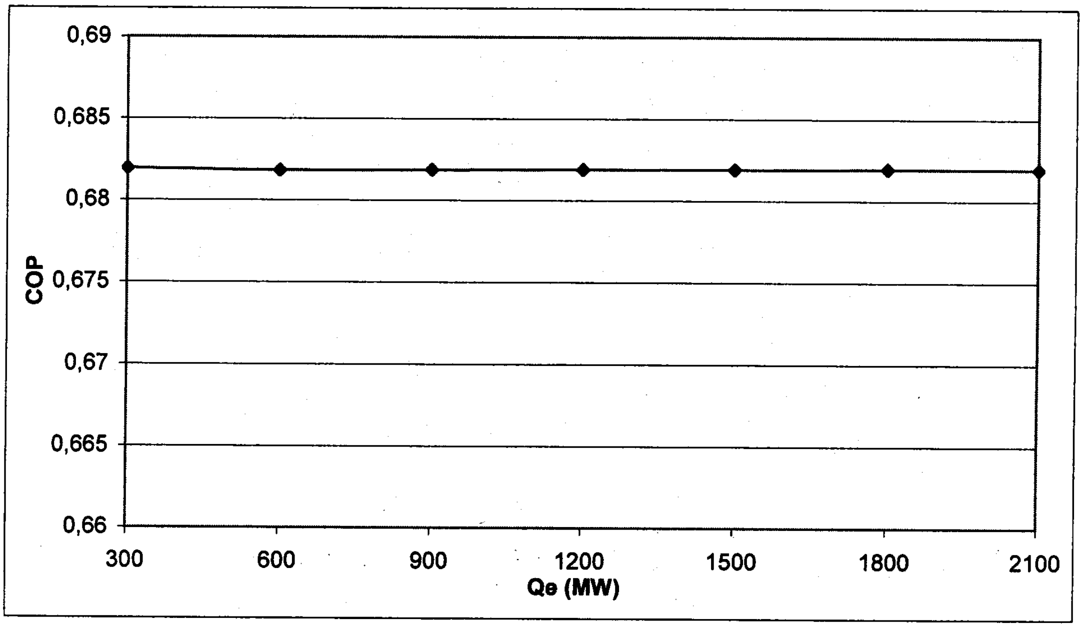


Fig. 4.13: COP versus Qe

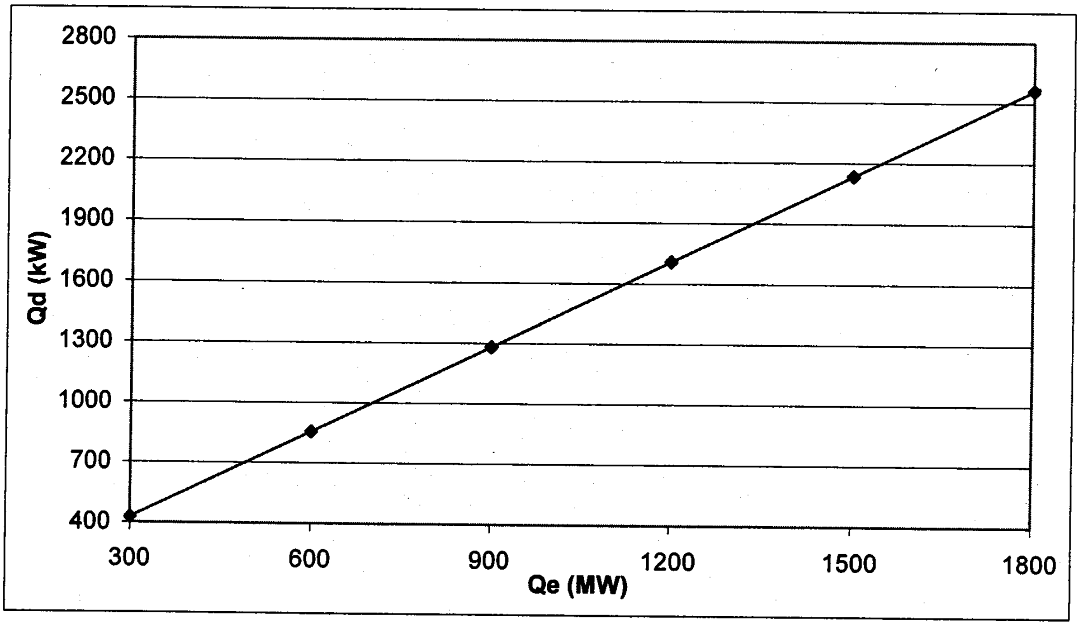


Fig. 4.14: Qd versus Qe

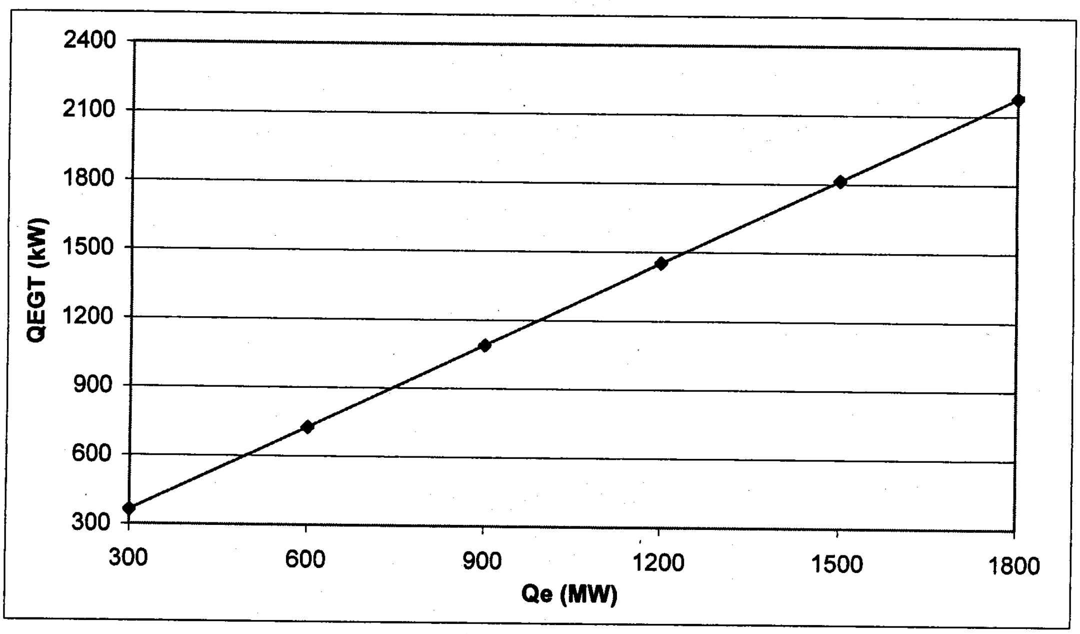


Fig. 4.15: Q_{EGT} versus Qe (assuming $\eta_{EHE=0.85}$)

4.7 Discussion-conclusions

Losses

Because absorption machines are thermally activated, large amounts of power input are not required. Hence, where power is expensive or unavailable, and gas, waste, geothermal or solar heat is, available, absorption machines provide reliable and quiet cooling. Because it takes about the same amount of heat to boil the refrigerant in both the generator and evaporator, it might be assumed that single effect cycles are capable of a COP_R of 1. Yet, the best single effect machines reach COP_R of only 0.5 to 0.7, as already said. The losses responsible for the COP degradation are traced to the following four phenomena:

- **Circulation loss**. When the cold solution from the absorber (1), is heated in the solution heat exchanger (3), the temperature at 3 is always less than the saturation temperature corresponding to the generator pressure and solution concentration, even for cycles with high heat exchanger effectiveness. Hence, heat must be added to boil the solution, which increases the generator heat input.
- **Heat of mixing.** Separating the refrigerant from the solution requires about 15% more thermal energy than merely boiling the refrigerant. This additional energy must be supplied to break the intermolecular bonds formed between the refrigerant and absorbent in solution. The heat of mixing also increases the generator o! Heat input.
- **Expansion loss.** As the refrigerant expands from the condenser to the evaporator, a mixture of liquid and vapour enters the evaporator. Not the entire refrigerant is available as liquid because some vapour was already produced by the expansion process. Thus, the evaporator heat transfer is reduced when vapour forms in the expansion process. This loss can be reduced by subcooling the liquid from the condenser.
- **Reflux condenser loss.** In the ammonia-water cycle another loss is introduced due to the volatility of water. In this cycle the refrigerant is ammonia and the absorbent is water. In the generator water vapour evaporates along with the ammonia. However, the proper operation, the water vapour must be removed from the ammonia vapour. The water vapour is separated in a distillation column, which has a reflux coil that condenses some ammonia-water. The heat removed in the reflux coil must be added to the generator, thus decreasing the COP_R.

In addition, other losses occur during transient operating conditions. For instance, if more refrigerant is produced than can be handled by the evaporator, the refrigerant is directly returned to the absorber via a spill over (point 11 in *Figs.4.4* and *4.5*). Liquid refrigerant returned directly to the absorber is a loss, and machines of recent design are tightly controlled to avoid this loss during transients.

Single-effect or double-effect

Thus, one of the limitations of single-effect absorption cycles is that they cannot take advantage of the higher availability of high temperature heat sources to achieve higher COP_R. Although the COP_R of a reversible cycle is quite sensitive to heat input temperature, the COP_R of a real absorption machine is essentially constant due to the irreversible effects associated with heat transfer. Thus, the cooling COP_R of a single-effect water/lithium bromide machine is around 0.7, essentially independent of the heat input temperature. To achieve higher cycle performance, it is necessary to design a cycle that can take advantage of the higher availability (or exergy) associated with a higher temperature heat input. Double-effect technology represents one such cycle variation.

In this cycle an additional generator and condenser are added (higher installation cost) to a single effect cycle. The heat input to the high temperature generator is used to drive off refrigerant, which on condensing drives a lower temperature generator to produce yet more refrigerant. In this way the heat input 10 the higher temperature generator is used twice. So the efficiency is increased, less heat is needed and thus less heat must be rejected. The double effect absorption chiller requires about 45% less energy input than a single effect absorption chiller; i.e. double effect chillers have a maximum COP of 1.2. Thus the double effect chillers are proposed for applications where heat is "valuable" regardless the increase of the complexity and the capital cost of the machine-installation.

Simple-effect chillers can be used from 65°C to 140°C in the generator, and double-effect chillers with a temperature up to 170°C. This temperature difference will determine their conditions of use.

Triple-effect absorption chillers are under development with a COP close to 1.5. The main technical problems are the high temperatures and pressures inside the machine. [59]

Capacity

Absorption cooling machines are available in sizes ranging from 10 to 6,000kW of refrigeration. Usually the single-effect machines vary from 300kW up to 2,500kW while the double effect from 350kW up to 6,000kW. These machines are configured for direct-fired operation as well as for waste heat or heat integration applications. The heat in indirect machines is transferred either by superheated water closed circuit (waste heat temperatures below 120^{o}C) or by steam closed circuit.

The capacity of an absorption chiller will drop concurrently with the decrease in temperature of the driving energy. More heat transfer surface is required for a given amount of cooling, resulting in higher investment cost per unit of cooling capacity.

Goteborg Energi examined the impacts on investment cost of installing chiller capacity to use the normal 75°C summertime hot water of its DHS, and concluded that it was more economical to increase the summer operating temperature to 100°C [72].

The other technical shortcoming is the re-cooling of the absorber and condenser of the absorption chiller. If re-cooling is done with a circuit using cooling towers, they must have large surfaces, and could have important plumes of water vapour. Price and environmental effects are increased.

A manner to avoid this shortcoming is the use of the rejected heat to produce hot water. Double-effect chillers, and especially direct fired chillers, can produce hot water up to 79.4°C while producing chilled water. [62]

Operation and maintenance

In the USA opinions vary regarding the comparison between maintenance costs for absorption and compression chillers. When taking in consideration their related equipment, costs seem almost the same for absorption as for electric centrifugal compression chillers. Maintenance is reduced because there are few moving parts, and their operating life is typically 30 years.

Start up and shut down take long time, which reduces the flexibility of operation in comparison with centrifugal chillers. Easy regulation of absorption chillers has to be noticed: the cooling performance can be easily regulated in the area between 10 and 100% of nominal load. The cooling performance can thus be well adjusted to the big differences in the required cooling caused by the ambient temperature and by solar insulation, with relatively low related COP variation. [59]

Investment and operating cost

The main problem for the development of this technical solution is the capital cost of the absorption chillers compared to compression chillers. The situation in the USA market is shown in Table 5.3.

Site-specific factors, such as additional costs to upgrade electrical service to power electric drive chillers; can change the comparative capital costs. Besides electricity “prices” are generally high in summer, which encourages the absorption chiller solutions.

A simple calculation shows that the gas price for a direct fired absorber should be roughly 3 to 4 times lower than the electricity price, to compensate the investment over cost.

One should notice that sales of absorption chillers are the far most important part of the Japanese air-conditioning market as well in individuals’ chillers as in District Cooling Systems (DCS). In Germany, progression of sales of absorption chillers has changed in 1990, because they offer a simple solution to replace CFCs, and they can be associated with existing DHSs. The part of their turnover was 5 to 10% before, but since it is in the range of 40 to 50%.

Depending on the nature of primary energy, the energetic consumption has to be evaluated for each heat source and situation. It is also the case for the calculation of CO_2 emissions.

The experience shows that 1MW of cooling power, corresponds to 120,000€ approximately (purchase, installation).

5. ECONOMIC EVALUATION

5.1 Introduction
Although it is not possible to predict the future some prediction and scenario studies can be useful to reduce the risk of the investment.

Investment in any new equipment is driven by economic return and so investment in trigeneration always competes with other projects that could prove more successful financially. A sophisticated discounting technique -based in **Net Present Value** method- is presented to allow the reader to make an initial assessment of the likelihood of trigeneration being attractive in a specific situation and hence whether further investigation into trigeneration would be worthwhile.

In order to carry out a realistic evaluation of a trigeneration plant, three actual **cases studies** were taken into account considering an airport, an island and finally a hotel. The energy demand data of the previous cases were analyzed in CHAPTER 2. Of course the overall investigation based in **hypothetical scenarios** that are likely to be perform in the future. This had been done with a **simulation program using FORTRAN** as programming language.

5.2 Economic data
In order to perform feasibility study or an economic analysis, there is need to know the cost for constructing and operating a system. Related information is given in this chapter. It must be emphasized that cost information given here is indicative and it can be used for first estimates only. Furthermore, cost changes with time and with the place where the plant will be installed. Therefore, the final decisions should be based on cost data -provided by companies, which will supply, install and, perhaps, maintain the equipment- adapted to the exact time and location where the investment will take place. An effort has been taken so the prices given at this thesis **are referring in year 2004 (1€=1.23$),** while the delivery country is **Greece**. The **average inflation of Greece for the years 2004, 2005 and 2006 is 3.5%**. The economics of trigeneration are made up of the investment costs, the unforeseen cost and the ongoing costs.

<u>**1. Investment cost** is also called **capital cost** or **initial cost** or **first cost**</u>.
This is the expenditure required for the establishment of an operational cogeneration on the site. It consists of equipment cost, installation cost, and "soft" (called also "project" or "engineering and management") costs:

- ***Equipment costs***

Equipment costs consist of the cost for purchase of the equipment, including any taxes, and transportation on the site. They depend on the components comprising the system and their particular specifications. The most important of those are the following.
Prime mover and generator set. Power output, alternative fuel capability, generator voltage, emission control techniques in prime mover, noise reduction.
Heat recovery and rejection system. Required media (steam, hot or chilled water), quality of thermal energy (pressure and temperature), number of pressure and temperature levels required, emission control equipment, water treatment unit.
Supplementary firing. Additional thermal capacity, alternative fuel capability.

Absorption cooling system. Including the purchase of the absorption chiller, the water-cooling tower, and the necessary additional infrastructure such as pippins etc.
Exhaust gas system and stack. Exhaust gas temperature, single or multiple stacks for multiple engines, emission control equipment, need for bypass valve.
Fuel supply. Interconnection with fuel supply system, storage capability, fuel metering; in particular for natural gas, need for compressor, if the line pressure has to be increased.
Control board. Extent of automation, requirements for unattended operation, inter-connection with the user's facility.
Interconnection with the electric utility. Connection line, one or two-way connection, safety and metering equipment.
Piping. Connection with the water, steam, compressed air (if needed) circuits.
Ventilation and combustion air systems. Ducts, filters, sound attenuation equipment.
Shipping charges.
Taxes, if applicable.

➢ ***Installation costs***
They consist of:
Installation permits,
Land acquisition and preparation,
Building construction,
Installation of equipment,
Documentation and as built drawings.
Grid connections, including reinforcement of local/national electricity networks
First set of spare parts and any special tools needed for servicing and repair
Some of these costs may not be applicable, e.g. if the space is already available for the trigeneration system.

➢ ***"Soft" costs***
Design and professional service fees for the analysis, planning and development of a cogeneration system are frequently referred to as soft costs. They may be in the range of **15-30% of the equipment cost**. The most significant professional fees and other costs are the following.

- *Architectural / engineering design fees.*
- *Construction management fees.*
- *Environmental studies and permitting costs.*
- *Special consultants and inspectors.*
- *Legal fees.*
- *Letters of credit.*
- *Training.*
- *Additional costs* may incur under certain financial arrangements (e.g. interest paid during construction, bank fees, and debt insurance).

2. Unforeseen cost that is obviously extra cost which cannot be predicted. It is desirable that cost to be as mush as low cans it be. In budget estimates, a contingency or allowance for unforeseen costs is taken into consideration. Early in the design process, the contingency may be in the range of 15-20% of the above three costs. At the completion of the design, when uncertainty is reduced, the contingency may be reduced to 5%.

3. Ongoing costs consists of fuel, staffing and maintenance

Examples of investment costs breakdown for cogeneration plants are given in the following tables.

Table 5.1: Breakdown of investment costs for small-scale cogeneration [78].

Type of cost	% of total
Cogeneration unit including heat recovery system (prime mover, ect)	55
Instrumentation, regulation and control	15
Auxiliary systems	5
Connection to grid	5
Civil work and/or acoustic enclosure	10
Installation and commissioning	5
Project costs	5
Total	100

Table 5.2: Examples of breakdown of investment costs for a gas turbine and a steam turbine cogeneration system [78].

Type of cost	% of total	
	Gas-turbine(1)	Steam-turbine(2)
Turbine-Generator	34	50 (3)
Heat recovery steam generator	20	-
Instrumentation, regulation, control	4	3
Auxiliary systems	7	4
Connection to grid	3	6
Civil work (land, buildings, roads)	6	11
Engineering and construction management	11	11
Contingency	15	15
Total	100	100

(1) Nominal power 10 MW.
(2) Non-condensing turbine. Nominal power 30 MW.
(3) Boiler cost is included.

A range of typical installed costs ($\$/kW_e$ or $€/kW_e$) for gas turbine cogeneration systems referred to year 1995 or 2001 respectively, can be seen from *Figs 5.1, 5.2.* Generally, systems less than 500kWe in size, cost between 800 and 1,300$/kWe with the specific cost rising sharply for the smallest systems. Systems greater than 500kWe in size have a lower specific cost. The graph does not show the neat curve of *Fig. 5.1.* The reasons for this is that, these systems have been installed at various times during the last five years, are operating in highly different situations and are designed for a variety of fuel types. Where fuels other than natural gas are used there is greater variation: projects that have converted diesel standby units cost less than $500/kWe while systems fuelled digester gas cost as much as $1,700/kWe.

Capital costs typically vary from 600€/kWe (for larger schemes) to more than 2,000€/kWe for the very small and depending on the choice of cogeneration plant and auxiliaries required (2004).

Costs for **steam turbine** systems which were also provided by companies are between: 200-500$/kWe. Conventional **thermal power burning lignite** has a total installed cost between 800-900E/kW, while a future similar using new low emissions technology fulfilling the emissions restrictions of the year 2008 will cost 1,500E/kW.

In Greece the **new thermal power plants (burning lignite) are estimated (2004) to have a production cost 35-45€/MWh**, (taking into consideration the any emission penalty) and they produce approximately 75% of the total national electric power. On the other **hand the production cost of the non-interconnected national grid** (see CHAPTER 2), **is 80-500€/MWh**, because the use of burning fuel is oil (diesel). For comparison reasons it is referred that the production cost of **a power plant using renewable sources will be 60-80€/MWh**.

The total specific cost is between 20 and 40% higher for a combined cycle than for a simple cycle. Manufacturing cost of a new Thermal Power Station 500-600MW is approximately 360-430 million Euros and depends to the technology that is going to use. (2005). As a rough estimate, the **total cost of a trigeneration plant** can be calculated as 2.8 times the budget price for the gas turbine quoted by the manufacturers. [78], [79], [83]

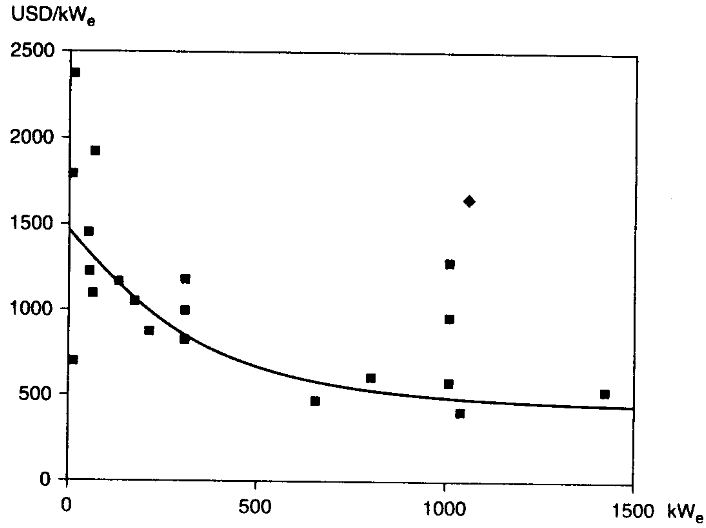


Fig.5.1: Installed $/kWe of gas engine cogeneration systems based on survey of equipment suppliers (1995). [77]

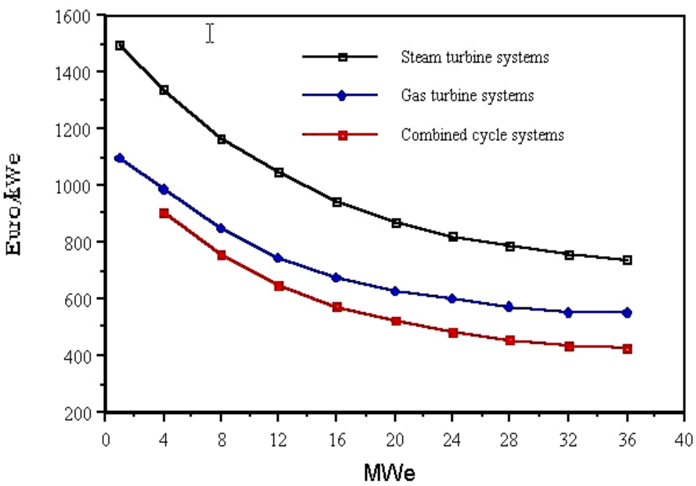


Fig.5.2: Specific investment cost of medium- to large-scale cogeneration systems (2001). [73]

It is evident that the investment cost of a cogeneration project depends on a lot of factors, which characterise the particular project. Comparison of equipment costs can give only indicative figures as each manufacturer offers different levels of quality, reliability, associated equipment etc. Furthermore, the standards set for pollution and noise emission differ from country to country. Any generalised costs cannot be useful but only for an initial and very rough estimate. **Costs may have an uncertainty of ±(20-25)%**
Obviously, the above information needs a slight mortification, in order to include the absorption cooling system, which is also included to a trigeneration power plant. (*Fig. 5.3*) [77][78] [81]

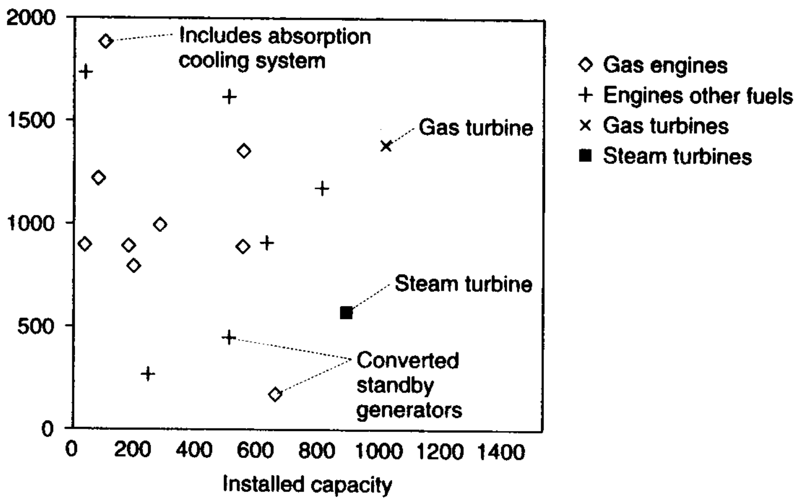


Fig.5.3: $/kWe information from demonstration projects (1995). [77]

5.2.1 Gas turbine system cost

While several major factors determine industry price levels, the major one has always been the balance between supply and demand for the product. As a result of huge over-ordering from 1998 through 2002 an excess of 60Hz units was built to be shipped by original equipment manufacturers for the U.S. market. There may have been as many as 150 units (or more) in storage back in 2003, when the glut was at its peak, waiting for final delivery and installation. In addition, several plants that were installed had less demand for their kWh production than was needed to be profitable, and were essentially put into mothballs at site until they could be used or sold.

Settling on a consensus can be challenging due to especially low prices in the large **60Hz** market as owners unloaded new equipment that they no longer needed (or wanted) and suppliers peddled surplus capacity. For 60Hz machines, particularly those 150MW+ units in popular demand for independent and merchant power plants and as components of popular 2-on-l combined cycle packaged plants, prices eroded by as much as 25 to 30 percent.

In contrast, **50Hz** large machines prices have either stabilized or seen a very smal1 decline, influenced by a general slowdown in global economies and wildly fluctuating fuel prices rather than disruption in market demand and supply. (*Fig.5.4*) The 50Hz market was never 'over bought' and continued at a reasonable pace for power generation equipment. And, as most manufacturing plants around the world are specialized for either 50Hz or 60Hz production, there was very little bleed- through of the 60Hz price deterioration into the 50Hz market. Some of the smaller 60Hz units in U.S. storage have been modified (geared or rebladed) to be sold into the Mid-East and a few other 50Hz market. All indications are that their prices did not represent a distress sale.

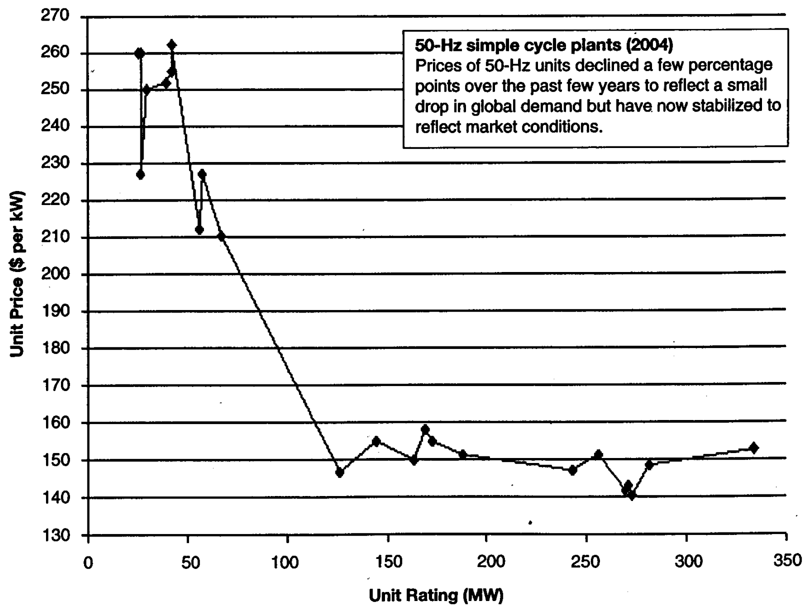


Fig.5.4: 50Hz simple cycle plants (2004). [79]

In general, **smaller machines in the 30MW** -and- below category are relatively easy to evaluate because that market has been fairly flat over the last few years and prices remain fairly stable. Taking a global look at the 1-30MW segment, sources indicate that the inquiry level is generally up. Europe and Japan are each accounting for about one-third of the total worldwide sales, with North America plus South America plus China and the rest of Asia Pacific accounting for the remaining one third. In the 1-30MW range, specifically in the Europe-Middle East-Africa theatre, a market slip occurred in 2000-2002, but recovered about halfway through 2003. It has remained strong through 2004, with general expectations that the current level is likely to hold on well into 2005. This is almost entirely CHP indusial power with- a waste boiler and sometimes an extraction steam turbine gensets.

Industry practice is to reference plant prices to base load design output rating on natural gas fuel at 59°F (15°C) ambient sea level site conditions and 60% relative humidity. Units are normally rated without water or steam injection for NOx reduction or power augmentation (unless otherwise specified) and without duct losses. The quoted nominal ISO output rating is measured across the electric generator terminals. As such, it includes electric generator efficiency and any reduction gearing losses. When evaluating the pricing of competitive machines, it is important to make sure that kW ratings are quoted for electric generator and not shaft output. Equipment and installation costs for the electrical substation, switchyard, pipe1ine connections, fuel gas compressor skids and black-start generator sets are not included in the prices quoted. Nor are storage tanks, fuel forwarding, and treatment systems for liquid fuel installations. Administrative offices, separate modular control room, workshops, storage buildings, spares, and consumables are not included. They also do not include water or steam injection systems for NOx control; complex multi-level inlet filtration; inlet chillers and anti-ice systems; tall exhaust stacks and chimneys. Electrical distribution, main step-up transformers, switchgear and motor control centres, poured concrete foundations and foundation bolting are not included.

Prices of single shaft gas turbines for various power output are shown in APPENDIX E.1. Two shaft gas turbines with similar power output have almost the same price. On the other hand gas turbines with heat exchanger or intercooler cost 30% and 20% respectively more than the simple cycle gas turbines with similar power output.

Capital cost plus installation cost of a gas turbine using fuel cells is estimated at 650-1500€/kW (2004), having efficiency at about 27-32%.

Installation prices, as is already said, are not included. However, they vary considerably depending on site location and local labour rates, and also the need for access roads, fuel gas pipeline extensions, training centres, repair facilities, and the like. The **installation cost of the gas turbine estimated around 10% of price shown in APPENDIX E.1**. [92]

Budgetary $/kW prices listed here are intended for preliminary project assessment and eva1uation of simple cycle electric power generating equipment. In genera1, installed and complete turnkey plant costs can conservatively add between 60-100% to the equipment-only prices shown here. Actua1 prices will depend on the changing situations in which competitive suppliers find themselves, geographic area of business interest, marketing strategies, and manufacturing capacity. All of these factors enter into the bid and evaluation process when shopping for new gas turbine generation.

5.2.2 Generator system cost

Generators convert the mechanical energy in the rotating engine shaft into electricity. They can be either synchronous or asynchronous. A **synchronous generator** can operate in isolation from other generating plant and the grid. This type of generator can continue to supply power during grid failure and so can act as a standby generator. An **asynchronous generator** can only operate in parallel with other generators, usually the grid. The unit will cease to operate if it is disconnected from the mains or if the mains fail, so they cannot be operated as standby units. However, connection and interface to the grid is simple. Synchronous generators with outputs below 200kWe are usually more expensive than asynchronous units. This is because of the additional control, starting and interfacing equipment that is required. In general, above 200kWe output the cost advantages of asynchronous over synchronous types disappear. There is a trend however, to use synchronous generators even on cogeneration units with low power output. Primarily air-cooled designs below 150MW output and hydrogen- cooled above 150MW. Even for the larger units, however, air-cooling is being chosen as a lower priced alternative.

Experience indicates that the **electrical efficiencies** realized are **97.5-98.5**% of the guaranteed values, although these levels can only be achieved by good maintenance.

Usually the **cost of the generator is included in the gas turbine package, paragraph 5.2.1. (APPENDIX E.1)**

The **gearbox**, which sometimes is necessary to use, for the reduction of speed, has **efficiency** approximately **98.5%**. The cost of the gearbox (if needed) **is included in the gas turbine package, paragraph 5.2.1. (APPENDIX E.1)** [79],[80], [81]

5.2.3 Heat exchanger and boiler system cost

The trigeneration plant needs the use of a heat exchanger which transfer heat from the gas turbine exhaust gases either to the heating system or to the absorption cooling system via water closed loop.

The average capital cost of the heat exchanger 48€/kWt (2004) with the installation included (10% of the capital cost). The **thermal efficiency of the heat exchanger** varies $\eta_{th,\,HE}$ **= 0.7-0.85**

On the other hand, **the average capital cost of the boilers, which are going to be replaced by a CHP system, is 45€/kWt (2004) plus 10% of that for the installation cost**. The **thermal efficiency of the boilers** varies $\eta_{th,\,b}$ **= 0.75-0.85** [80] [90] [91] [98]

5.2.4 Absorption chillers and electric centrifugal cost

Absorption cooling systems are considerably more expensive than conventional electric compressor chillers (*Table 5.3).* In addition, absorption chillers will often require larger cooling towers and larger condenser water pumps, which further increase system costs. [73][74] [75]

Table 5.3: Capital plus installation cost for the electric and absorption chillers of various capacities (2004)

Capacity (kW_c)	500	1,000	1,800	3,500	5,000	10,000
	Installed cost (€/kW_c)					
Electric centrifugal (10% installation)	128	80	79	65	57	40
Single effect absorption chiller (20% installation)	185	120	100	85	80	55
Double effect absorption chiller (20% installation)	210	145	130	120	110	70

5.2.5 District heating

An approximate estimation of the cost of a city district heating system could be derived of an existing case in Greece. The city of Ptolemaida is served by a CHP system with a capacity of 120MW_{th}, with an **overall efficiency of 0.85-0.90**. The thermal power station is located 4km away from the town and is piping superheated water (120^{o}C) to the town using pre-isolated pipelines. The **total cost of the installation system is estimated about 3.5€/kW**, (2004). [83],[91],[103]

5.2.6 Connection to the grid cost and cost of back up generator

It is obvious that there are some expenses having to do with the connection of the trigeneration power plant to the local national gird. These are dependable from the power and the location of the plant. A first estimation might be a **1€/kW$_e$ connection to the grid cost.**

The **capital and the installation** cost of back up generator for large power installations is approximately **80€/kW**. The maintenance cost is usually offered for free. [102]

5.2.7 Operation and Maintenance Costs

Operation and maintenance (O&M) costs depend to a certain extent on decisions taken at the design and construction phase of the system. The O&M, as it will be seen, depends a lot to the fuel prices (**fuel contributes about 70% of the total O&M costs**). It is possible that actions reducing the initial cost may lead to increased operation and maintenance costs, with a negative impact on the total economic performance of the project.

Typical O&M costs for a cogeneration plant referring to the year 2004 are [72],[77], [80]: for gas turbine cycles 0.005-0.0115E/kWh, for reciprocating engine 0.008-0.016E/kWh, and for steam cycles 0.0035E/kWh

The major operation and maintenance costs are the following:

Fuel is usually the most significant operation cost, which may reach 70% of the total operation cost, over a typical service life of 20 to 30 years. An exception can be when fuel is a by-product of a process or produced by wastes (**biomass products**). The particular fuel tariff or the agreement between the cogenerator and the fuel supplier has to be taken into consideration in calculating fuel cost. Much lower costs are due to other consumables, such as lubricating oil, made up water and chemicals. For a base load combined cycle plant in the 400-500MW range, burning \$4 to \$5MMBTU (10^6BTU) natura1 gas fuel, even a single percentage point in efficiency can reduce operating costs by more than \$20 million over the life of the plant. [79]

Gas turbine performance is calculated on the basis of the lower heating value (LCV) of the fuel to be burned. Purchase contracts for the amount of fuel required, however, are determined by the higher calorific value (HCV) of that fuel. The difference between the lower and higher heating value is Btu content that you pay for, but never see as gas turbine output. Technically, it is difficult to explain. But it all has to do with fuel-bound hydrogen that forms water as a by-product of combustion and is wasted.

HCV is measured on the basis of the chemical energy in the fuel, which accounts for the total heat given up when the fuel is burned -including formation of water vapor while LCV measures the useable energy. The bottom line is that some 6% by weight of liquid fuels ends up being "wasted" in the gas turbine combustion process versus 11% for natural gas fuel. Or, put another way, **the LCV fuel consumption must be increased by a factor of 1.06 for liquid fuels and by a factor of 1.11 for natural gas**. Cycle studies for gas turbine projects are carried out on an LCV basis and fuel requirement on a HCV basis. In short, figure on having to buy more fuel than you might expect by using the heat rate in the performance specifications to calculate your fuel requirements.

Consequently, the oil prices one of the most important factors, which determine the profitable investment of a trigeneration plant. These prices are following the laws of the offer-demand in a universal scale. On the other hand they are very dependable to the global politic scene and to the "stock market deals". Fig. 5.5 shows the trend of the oil market in Europe the last years.

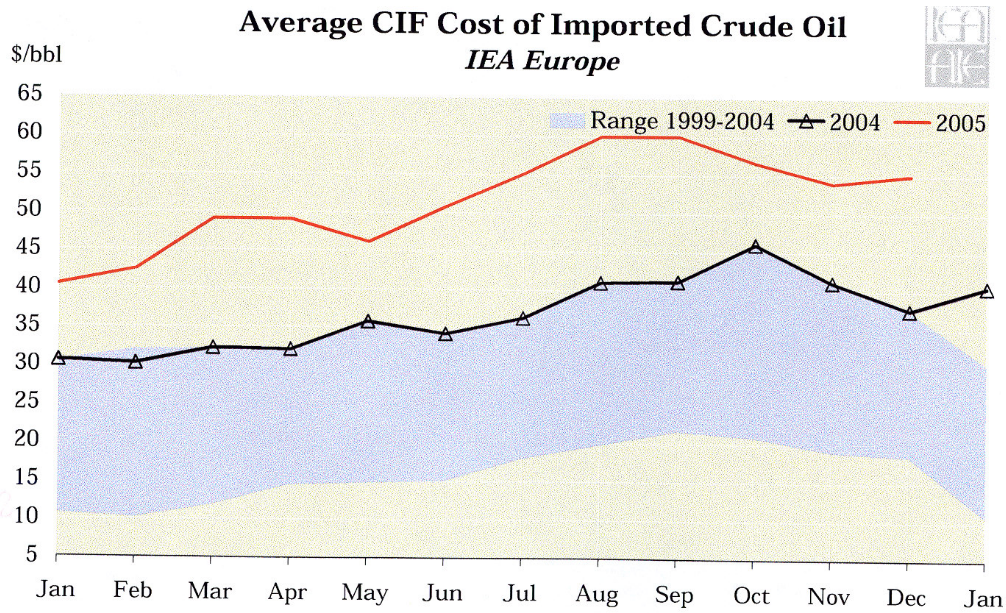


Fig.5.5: Crude oil prices [72]

The calculations of crude oil prices are based on the equation:

$$COpr=2.9t+42.1 \quad (5\text{-}1)$$

where COpr in $/ bbl is the crude oil price in the year t=1 which corresponding to the year 2004. This equation based on the assumption that the crude oil prices is 45$/bbl for the first year of the operation (2004), and the 20th year (2024) of the operation the crude oil prices will be 100$/bbl. It is obvious that there will be many fluctuations throughout theses 20 years, for example in 2006, the price of the crude oil went well over 65$/bbl, reaching a peak of 78$/bbl. (1barel=42gallons=159lt, Fig. 5.6)

It is evident that the price per liter of the crude oil given by the international oil markets should be multiply by a factor **(2.2 for motor oil, 1.8 for light heating oil, 1.7 for medium heating oil and 1.6 foe heavy heating oil)**, which takes into account the transportation, distillation fees, taxes, quantities, etc.

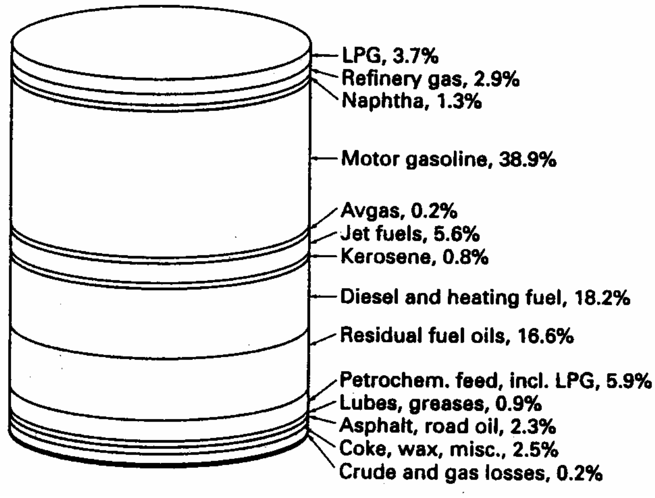


Fig.5.6: Typical end products from crude oil. A single refinery produces some, but not all, of the products shown. The percentages refer to overall production from total refinery output. [105]

As it has been said in CHAPTER 2, approximately 80% of the national electric power is coming from plants using **natural gas (NG)** as burning fuel. 80% of that comes through pipelines from Russia and the rest from Algeria in liquid form **(LNG)**, by ships. The national energy policy promotes the use of NG in power production. The pipeline is entering Greece from The North and is ending to the capital, Athens. It is passing through the major cities of Greece including Thessaloniki. The contracts so far, are until 2016 and 2021 for the NG and LNG respectively. The calorific value of the two types is about the same and the price is in general the same. The price policy of the Natural Gas depends to the consumer namely if he is a big and constant consumer. For example:

- Power less or equal 300kW up to 180days/year, the price will be **0.020€/kWh (0.27€/kgr)**
- Power less or equal 300kW up to 365days/year, the price will be **0.019€/kWh (0.256€/kgr)**
- Power less or equal 1MW up to 365 days/year, the price will be **0.018€/kWh (0.243€/kgr)**
- Power over 1MW for 365days/year, the price will be **0.015€/kWh (0.20€/kgr)**

All the above are approximate prices in 2004, when crude oil costs 45$/bbl.

Again, the equations calculating the NG price the next 20 years should be as follow and they are based in the assumption that at any time the NG price has a constant relation with the crude oil price.

$$NG1pr=2.39t+17.61 \qquad (5\text{-}2)$$
$$NG2pr=2.39t+16.61 \qquad (5\text{-}3)$$
$$NG3pr=2.39t+15.61 \qquad (5\text{-}4)$$
$$NG4pr=2.39t+14.61 \qquad (5\text{-}5)$$

where NGpr are the NG prices in €/MWh corresponding to the four previous mentioned demand categories, t is the number of the years after 2004 e.g. year t=1 (2004), t=2 (2005), ect. It is also assumed that the gradient of the NGpr is equal to the Copr gradient in €/MWh.

Table 5.4 shows the basic properties of the most frequently used fuels.

Table 5.4: Typical properties of common gaseous, liquid and solid fuels [99]

FUEL	Mass Composition	FCV (lower)		Density (ISA)	
		(MJ/kgr)	(MJ/m^3)	(kgr/m^3)	(kgr/lt)
Russian NG	CH_4:98% C_2H_6:0.6%	48.6	36.2	0.74	
Algerian NG	CH_4:91.2% C_2H_6:6.5% C_3H_8:1.1%	48.9	38.2	0.78	
Motor gasoline (diesel)	C:85.5% H:14.45% S:0.05%	43.5			0.762
Kerosene	C:86.5% H:13.2% O:0.01% S:0.6%	43.2			0.81
Motor diesel (oil)	C:86.0% H:13.2% O:0.2% S:0.6%	42.7			0.84
Light heating oil	C:85.5% H:12.5% O:0.8% S:1.2%	42.5			0.86
Medium heating oil	C:85.3% H:11.6% O:0.6% S:2.5%	41.0			0.92
Heavy heating oil (residual, mazut)	C:84.0% H:11.0% O:1.1% S:3.5% N:0.39%	40.3			0.97
Coke	C:97.5% H: 0.3% O:0.3% S:0.9% N:1.0%	29.0			
Lignite	C:65.0% H: 5% O:27% S:0.5% N:1.5%	5.0			
Turf	C:57.0% H: 5.5% O:34.0% S:1.0% N:3.0%	7.5			
Coalgas (Syngas)	CH_4:4.5% H_2:16.0% N_2:55.0% CO:32.0% CO_2:10.0%		6.1		
Biodiesel		8.0		0.6	
Biogas			22.5		

Personnel costs depend on the size of the system and the degree of automation. Smaller cogeneration systems (up to about 10MW) can operate unattended. Medium-size systems (10-30MW) will typically require attended operation (one person may be sufficient). Larger systems will require attended operation with two or more persons. If the system includes an exhaust gas boiler, then safety regulations may require attended operation even for smaller systems. If solid fuel is used, then increased personnel may be required. It is important to clarify whether additional personnel are needed, or if the personnel already available (e.g. in an industry) can operate the system. In the latter case, the incremental personnel cost will be zero.

Assumption

- **No extra expenses for personnel are needed.**

Maintenance costs depend on factors such as type of prime mover, type of fuel, operation cycle and operating environment. Heavy-duty engines usually require less maintenance than light-weight engines. Gas turbines operating with high TETs usually have increased maintenance cost, due to creep phenomena appearing in the turbine section. The use of heavier or dirty fuels and operation in a dirty environment will increase maintenance costs. Frequent cycling (starting up and close down) will increase thermal stresses, which results in increased maintenance costs.

If skilled personnel are available on site, then the incremental maintenance cost will be lower. A variety of maintenance contracts may also be available; if such a contract is signed, it will directly affect the cost. If a performance monitoring system is installed with the capability to identify and predict potential failures, then maintenance as-needed instead of as-scheduled can be followed. In such a case, it is expected that maintenance costs will be reduced.

Another important consideration, apart from routine maintenance requirements, is lifetime, or hours run, before a major overhaul is required. Oil assays and routine inspections will determine precisely when an overhaul is necessary. As a guide, a typical cogeneration unit might operate for 4-5000 hours/year and need an overhaul after 6-7 years. Any economic evaluation should be extended to encompass this period as the rebuild cost may be around a third of the engine's replacement cost. Over time, wear leads to deterioration in performance and efficiency and these will be returned to nearer design levels following an overhaul. [77]

One ordinary gas turbine can operate up to 30,000h, before stopped for a major overhaul. In the mean time slight inspections and corrections in the settings were taking place. The maintenance costs of gas turbine, which is stopping every hour, might be triple compare to another, which stops every 1,000h. On the other hand, the need of maintenance depends on the operating conditions in which the gas turbine is operating. This for example means that if the GT is operating at part load for long period, the time between the overhauls will increase (maintenance cost will decrease), or if the GT is operating at over load for long period, the time between the overhauls will decrease (maintenance cost will increase). Finally the maintenance costs of gas turbine, which is using oil as fuel, are triple compare to another using natural gas.

Typical maintenance costs of gas turbine using natural gas is 2.8-3.4€/MWh for larger plants (above 1MW) 3.4-4.6€/MWh for small plants (under 1MW) (2004)[79] **Expressed as an annual cost the value is between 4 and 6% of the total installed price for small plants (0.5-5MW) and between 0.5 and 1.5% for large multiple turbine plants**. [79]

Generators are considered to have negligible maintenance cost.

Heat exchanger has rather low maintenance cost in order of 2% heat exchanger capital cost, while boilers have maintenance cost in order of 2% boiler capital cost.

Although it costs more to maintain absorption chillers than electric chillers (expect to pay an additional 0.8 per RTh, less as capacity increases), maintenance can be a minimal expense for facilities with on-site maintenance personnel. **Maintenance costs for absorption chillers range from about the same as for electric chillers to as much as one-third more (consider: electric chillers maintenance cost 3% of the chiller capital cost).**

Most manufacturers offer long-term maintenance contracts to minimise the risk to end users and give visibility to the costs incurred.[80]

Insurance adds also to the operation costs. It may be only for equipment failure, or it may be extended also to loss of income, loss of savings, or business interruption. The cost of insurance varies depending on the type of prime mover, the equipment performance history, and the system design and operating mode. It can be in the range of 0.25-2% of the capital cost. In some cases, particularly for smaller units, the insurance may be covered under the owner's overall insurance program at no additional cost.
Other operation costs include **administrative** and **management fees**, **taxes**, **interest on loan** (if any).

It can be considered that operation and maintenance costs consist of fixed and variable costs. Fixed costs are those which occur no matter whether the system operates or not. Variable costs depend on the operation load and schedule of the system. As with the investment cost, operation and maintenance costs are system-specific. For a first estimate only, cost information published in the literature can be used, which often does not separate between fixed and variable costs, but provides average costs. This is the case with the values given in *Table5.5*

Table 5.5: Operation Maintenance costs for cogeneration systems (2004). [76]

CHP System based on	Maintenance cost* (E/MWh$_e$)
Steam turbine	2.5 – 1.6
Gas turbine	5.8 – 4,9
Combined cycle	8.1 – 6.9
Reciprocating engine	9.9 – 6.2
* Lower values are applicable to larger systems.	

5.2.8 Emissions price penalty

The most important issue regarding the environmental impacts is whether cogeneration improves or degrades air quality. This issue is especially critical in urban areas, where air quality may be lower than the national average, and the tolerance for additional emissions may be small. Assessment of the effects of cogeneration on air quality is often complicated, because effects vary from one location to the other. For example, the effect may be positive (decreased emissions) in the vicinity of the central power plant serving the region, but it may be negative (increased emissions) at the site where the cogeneration system is located. This difference makes it necessary to perform the analysis at two levels: local level, global level.

Depending on fuel type, and allowable emission levels, the cost of gas turbine emissions controls and post combustion treatment systems can add substantially to the base price of a plant. In general, the tighter the air quality emission regulations, the more you will have to spend on gas turbine and plant equipment.

Exhaust Gas Emissions

The components of the exhaust gases, which are of concern because they are hazardous, are the following: carbon dioxide (CO_2), carbon monoxide (CO), nitrogen oxides (NOx), sulfur oxides (SOx, usually sulfur dioxide: SO_2), unburned hydrocarbons (CxHy, also symbolised with the letters HC or UHC), solid particles, called also "particulates". Laws and regulations specify maximum emission levels for power plants. They usually are applicable for cogeneration systems too. Some countries may have a special legislation for cogeneration systems. *Table 5.6 A* gives typical levels of uncontrolled emissions for various cogeneration technologies. It should be mentioned that the emission level depends on the cogeneration technology, the year of manufacture, the condition (age) of the unit, the rated power, the load of operation (percent of the rated power), the type and quality of the fuel used, the operation of pollution abatement equipment, etc. Consequently, it is evident that tables such as are appropriate for first estimates only. Accurate assessment of a system should be based on data pertinent to the particular case.

Table 5.6: Typical values of [78]:

***A.** uncontrolled emissions from cogeneration*

System	Fuel	Electrical efficiency (%)	Specific emissions (gr/kWh$_e$)					
			CO_2	CO	NO_x	HC	SO_x	Particulates
Diesel	Diesel 0.2% S	35	738.15	4.08	15.56$^{(2)}$	0.46	0.91	0.32
	Dual$^{(1)}$		593.35	3.81	11.30$^{(3)}$	3.95	0.09	0.04
Gas engine	Natural gas	35	577.26	2.80	1.90	1.00	≈0	≈0
Gas turbine	Natural gas	25	808.16	0.13	2.14	0.10	≈0	0.07
	Diesel 0.2% S		1033.41	0.05	4.35	0.10	0.91	0.18
Gas turbine-low NO_x	Natural gas	35	577.26	0.30	0.50	0.05	≈0	0.05
Steam turbine (new)	Coal	25	1406.40	0.26	4.53	0.07	7.75	0.65
	Fuel oil		1100.00	≈0	1.94	0.07	5.18	0.65
	Natural gas		808.16	≈0	1.29	0.26	0.46	0.07
Fuel cells (PAFC)	Natural gas	40	505.10	0.03	0.03	0.05	≈0	≈0

(1) 90% of energy supplied by natural gas and 10% by Diesel oil.
(2) Engines of modern designs emit 11-12 gr NO_x/kWh$_e$.
(3) Engines of modern design emit 7-8 gr NO_x/kWh$_e$.

***B.** emissions from central power plant systems*

System	Fuel	Efficiency (%)	Specific emissions (gr/kWh$_e$)					
			CO_2	CO	NO_x	HC	SO_x	Particulates
Steam turbine (old)	Coal 3% S	34	1034.12	0.18	3.13	0.05	19.87	1.41
	Fuel oil 1% S	31	887.06	0.18	3.18	0.05	4.76	0.23
	Natural gas	31	651.74	0.09	3.04	0.18	≈0	0.05
Steam turbine (new)	Coal	31*	1134.20	0.18	2.50	0.05	6.00	0.14
	Fuel oil low sulfur	31	887.06	0.18	1.36	0.05	3.63	0.14
Gas turbine	Diesel oil	34	759.86	0.55	2.40	0.18	0.14	0.18
	Natural gas	34	594.24	0.55	1.95	≈0	≈0	0.05
Gas turbine low NO_x	Natural gas	38	531.68	0.30	0.50	≈0	≈0	0.04

* Lower efficiencies of new steam turbine systems are due to NO_x and SO_2 abatement equipment

***C.** emissions from water and stem boilers.*

System	Fuel	Specific emissions (gr/kWh$_{th}$ of useful heat)					
		CO_2	CO	NO_x	HC	SO_x	Particulates
Boiler for hot water	Natural gas	252.55	0.03	0.19	0.02	≈0	0.02
	Diesel 0.2% S	322.94	0.06	0.25	0.02	0.37	0.03
Steam boiler	Coal	439.50	0.08	1.36	0.02	2.32	0.20
	Fuel oil	343.73	0.06	0.57	0.02	1.55	0.20
	Natural gas	252.55	0.03	0.39	≈0	≈0	0.02
Industrial steam boiler	Coal 2% S	439.50	0.16	1.12	0.08	5.65	0.98
	Fuel oil 1% S	343.73	0.06	0.78	0.02	2.03	0.30
	Natural gas	252.55	0.03	0.33	≈0	≈0	0.03

An 80% efficiency of the boiler has been considered.

CO_2 Emissions (effect: global warming)

Carbon dioxide emissions depend primarily on the type, quality and quantity of the fuel used. To a satisfactory approximation, complete combustion can be assumed, which is very close to reality, when combustion takes place with excess air and the combustion equipment is in good condition and adjusted correctly. Then, the quantity of the emitted CO_2 is calculated by the equation

$$m_{CO_2} = \mu_{CO_2} m_f \tag{5-6}$$

where $\mu_{CO_2} = \dfrac{44}{12} c$ (5-7)

$$m_f = \frac{E}{\eta \cdot FCV} \tag{5-8}$$

m_{CO_2} mass of emitted CO_2,

μ_{CO_2} emissions of CO_2 per unit mass of fuel (e.g. kg CO_2/kg fuel),

c mass content of carbon in fuel (e.g. kg C/kg fuel),

m_f mass fuel consumption,

E useful energy produced by the system,

η efficiency of the system, based on the lower heating value of fuel,

FCV fuel calorific value (lower)

Equations (5-6)-(5-8) are applicable not only to cogeneration systems, but to any system burning fuel. For example, when they are applied to a power plant or a cogeneration system, E is the electricity produced and η is the electrical efficiency; η_e. Typical values of c, μ_{CO_2} and FCV for various fuels are given in *Table 5.7*.

Table 5.7: Typical properties of fuels for calculation of CO_2 emissions.

Fuel	Carbon content (**c**·100) %	CO_2 emissions μ_{CO_2} (kgCO_2/kgfuel)	**FCV** lower (MJ/kg)
Natural gas (Russian)	$\frac{12}{16}\cdot 0.98 + \frac{24}{30}\cdot 0.006 + \frac{36}{44}\cdot 0.002 + \frac{48}{58}\cdot 0,002 = 0.743$	2.7243	48.6
Natural gas (Algerian)	$\frac{12}{16}\cdot 0.912 + \frac{24}{30}\cdot 0.065 + \frac{36}{44}\cdot 0.011 + \frac{48}{58}\cdot 0.002 = 0.746$	2.7363	48.9
Motor diesel (oil)	0.86	3.1533	42.7
Light heating oil	0.855	3.135	42.5
Medium heating oil	0.853	3.1276	41.0
Heavy heating oil (residual, mazut)	0.84	3.08	40.3
Lignite*	0.65	2.3833	5.0
* Data are valid for fuel with no moisture and ash.			

It has to be clarified that if the values of the parameters appearing in Eqs. (5-7) and (5-8) change for any reason (e.g. change in efficiency due to partial load, change in quality and consequently in c and H_u of fuel), then the total CO_2 emitted during a period of time results as an integral over time (or summation over various times intervals) of Eq. (5-6).

The only way to decrease the CO_2 produced for a certain quantity of useful energy production is to increase the efficiency of the fuel utilization (if the fuel remains the same). However, the quantity of CO_2 finally released to the environment would be lower than the one produced, if CO_2 could (at least partially) be used in a process. Large-scale applications, perhaps not easily combined with cogeneration, include the enhancement of crude oil and coal recovery from oil wells and coalmines, respectively.

Also, CO_2 can be used with hydrogen for production of synthetic hydrocarbons. More close to cogeneration applications is the use of CO_2 for enhancing the growth rate of plants cultivated in greenhouses.

In the case of Greece, in the period 1990-2003 the electric power generation has increased 67%, while the CO_2 emissions about 32%. The average CO_2 emission factor of the electric generation system was 1.3kgr/kWh in 1990 and decreased to the level of 1.03kgr/kWh (1.08kgr/kWh for the lignite fuelled thermal power plants, 470kgr/kWh for the natural gas fuelled combined power plants) in 2003 (32% decrease). The estimations are 1kgr/kWh and 0.851kgr/kWh in 2005 and 2010 respectively.

After the Kyoto agreement, every country that participates in that has to restrict the CO_2 emissions below a certain limit. This limit stands for the country, or for the Public Power Company or for any Company which producing CO_2 emissions. If that limit is exceeded then a price penalty should be paid. In the case of Greece, that limit corresponds to the quantities of CO_2 emissions, which the country had already from the year 2002. Therefore, the limit is well overtaken and the **PPC will pay 40E/additional tone of CO_2 emissions for the period 2004-2007 and 80€/additional tone of CO_2 emissions for the period 2008-2024**.

The other solution for a CO_2 over productive company is to **buy CO_2 emissions rights from another company (the market price is 8-10€/ton CO2, 2004). In 2006 this price went up to 22-25€/tonCO_2**

<u>Emissions of CO and HC (effect: toxic)</u>
In spite of the excess air, at certain points in the combustion region the conditions are such that molecules of carbon monoxide are not further oxidised to carbon dioxide, or molecules of hydrocarbons are not burned to produce carbon dioxide and water vapor. The quantities of these two constituents in the exhaust gases are kept at a minimum; significant amounts would indicate low efficiency of combustion due to improper mixing of fuel with air, or bad operating conditions. There is no simple way to calculate the concentration of CO and HC in the exhaust gases. Experimental measurements performed by the manufacturers are used to derive results such as those presented in *Table 5.6 A,C.*

Proper maintenance and adjustment of the combustion equipment is absolutely necessary to keep CO and HC emissions inside specified limits. If a system does not satisfy legal limits or if further reduction is required, then **a catalytic converter** can be installed to promote the oxidation of both CO and C_xH_y. Supplementary air may be required for this oxidation, in particular if low excess air is used in the combustion.

NO_x Emissions (effect: toxic, depletion of zone within stratosphere)

Nitrogen oxides are formed in the combustion process from nitrogen chemically bound in the fuel or present in the air. It is the pollutant that causes the greatest concern and legislative attention; the toxic effects of NO_x occur at concentrations which are at least 10 times lower than the levels at which CO becomes toxic.

Research and development in combustion equipment succeeded in reducing NO_x emissions from gas turbines by nearly an order of magnitude during the last years. Also, boilers and steam power plants have relatively low NO_x emissions (*Table 5.6*). However, Diesel and gas turbine have much higher levels, which are due to the high combustion temperature and pressure. The most important parameters that determine the level of NO_x formation in a Diesel or gas turbine are
the combustion temperature in the primary zone of combustion chamber,
the retention time in the primary combustion zone,
the combustion pressure,
the mixing rate of air and fuel.
The stoichiometric air ratio

$$\lambda = \frac{\text{real mass of combustion air}}{\text{stoichiometric mass of combustion air}} \tag{5-9}$$

often called “lambda ratio” for convenience, has a direct or indirect effect on the aforementioned parameters and, consequently, on the NO_x emissions. It also affects CO and HC emissions, efficiency and power output of the engine. *Fig.5.7* gives an example of this effect.

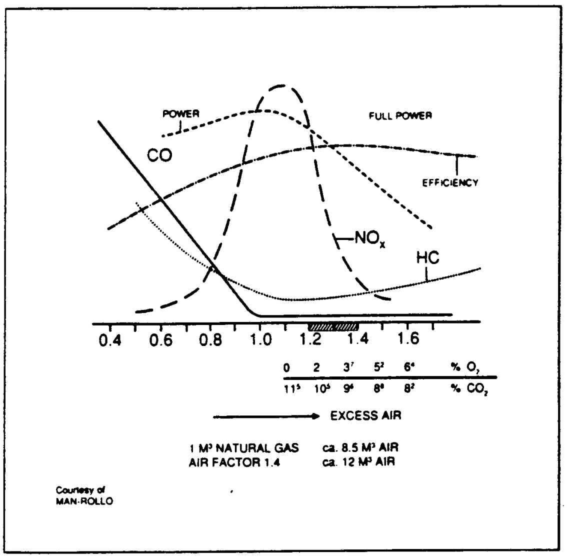


Fig.5.7: Effect of stoichiometric air ratio (λ) on NOx, CO and HC emission, power output and efficiency of a gas engine [78]

In gas turbines, low-NO_x burners, steam injection in the combustion chamber and catalytic reduction of the exhaust gases are the most usual techniques for NO_x abatement. The methods for reduction of NO_x emissions in Diesel and gas engines could be classified in two categories:

- active reduction of NO_x formation through modified engine design and operation,
- passive reduction of NO_x in the exhaust gases.

Active reduction of NO_x

Several methods are used by the manufacturers, which aim at reducing the combustion temperature and achieving complete and quick combustion:

Basic emission control systems include water or steam injection for combustion NOX reduction on both natural gas and distillate fuels. Nearly one half of the turbines are equipped with water or steam injection, and both types are equally represented.

Delaying the ignition timing. It decreases the temperature in the combustion chamber. However, it has adverse effects on the power output and efficiency of the engine, which limits the period by which the ignition timing may be delayed.

Changing the stoechiometric air ratio (λ). As shown in Fig. 5.7, NO_x emissions are maximum when $\lambda \cong 1.1$ (for the particular engine). They can be reduced either by rich combustion ($\lambda<1$), or by lean combustion ($\lambda>1.1$). Values of $\lambda<0.9$ are not acceptable, because they cause excessive formation of CO and HC (incomplete combustion). The value of λ finally selected is the result of a compromise between low emissions and high power and efficiency. Supercharging helps in meeting NO_x limits with no loss of power.

Air-fuel control. During partial load operation, the air and fuel flow rate must be controlled so that the performance of the engine is good and the emissions are low. The values of λ at partial load may be considerably different than the value at nominal power.

Exhaust gas recirculation (EGR). Part of the exhaust gases (up to 40%) is combined with the air and fuel mixture. Thus, the mixture entering the cylinders has a lower heating value. Consequently, the maximum combustion temperature is lower resulting in decreased NO_x formation. However, EGR may lead to increased corrosion rate, and decreased power output and efficiency.

Passive reduction of NO_x

While active techniques aim at decreasing the quantity of NOx produced during combustion, passive techniques aim at decreasing the NOx content in the exhaust gases by catalytic reduction of NOx to nitrogen and oxygen. Catalytic converters can be divided into two groups:

Non-selective catalytic reduction (NSCR). As the name implies (non-selective), it reduces not only NO_X, but also CO and C_xH_y. This is why the devices are called three-way catalytic converters. The process is based on the property of rhodium to temporarily bind oxygen present in NO_X, thus releasing the nitrogen. The oxygen subsequently reacts with CO and C_xH_y to form CO_2 and H_2O. Control of λ is of utmost importance in the proper functioning of the converter, because exhaust gases must have no oxygen. For this reason, such a converter can be used only with rich-burn engines (low λ) or engines with exhaust gas recirculation (EGR). The effect of λ on the conversion efficiency of the process is illustrated in Fig 5.8. As it is shown in the figure, the operating margin with respect to λ values is narrow. The conversion reactions are exothermic. If too much unburned fuel leaves the engine, it will result in too high temperatures in the converter, causing damage. EGR and non-selective catalytic converters reduce NO_X emissions by 80-90%, CO by about 80% and HC by about 50%.

Selective catalytic reduction (SCR). It is used to reduce only NO_X in the exhaust gases. It is used with engines, which operate with excess air, such as two-stroke, supercharged, and lean-burn engines. Ammonia (NH_3) has to be added in the exhaust gas for the NO_X reduction. The cheapest way is to inject liquid ammonia solution into the converter. Since the quantity of the solution depends on the load of the engine, a control system is required to adjust the flow of ammonia.

Adding exhaust flow NO_X and CO catalytic reduction to achieve single-digit emissions (in strict attainment areas) can increase equipment costs by 40-50%. Some installations also add post-firing treatment with NO_X and CO catalytic reduction, adding substantially to ba1ance-of-plant and operating costs.

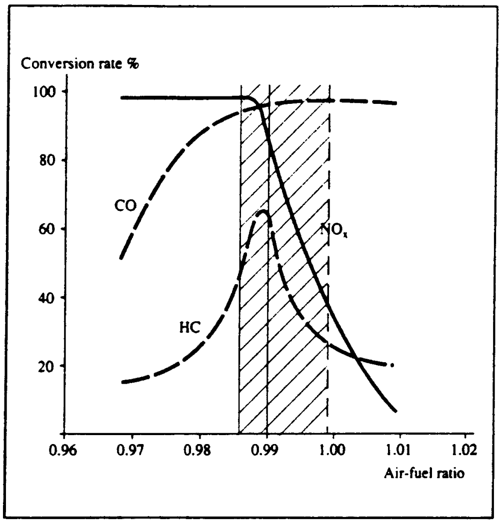


Fig.5.8: Effect of stoichiometric air ratio (λ) on conversion of non-selective catalytic reduction [78]

More and more gas turbines are being equipped at the factory with standard dry low-NO_X/CO combustors for operation on natural gas fuel. A few systems are coming out with dry low emissions on distillate as well, but this generally calls for fluid injection systems when burning liquid fuels. **Dry Low Emissions systems (DLE)** are often provided as standard equipment on large heavy frame engines without any appreciable increase in price level. This is generally true because the DLE system designs are relatively simple to engineer and install (more room in the combustion section). For aeroderivative machines, on the other hand, the complexity of dry low emissions systems can add 5-10% and more to the engine price.

For example, if no reduction method is used, NO_X, emissions for reported plants are between 0.5-1gr/kWh. When water or steam injection is applied, the emissions are between 0.16-0.27gr/kWh. The amount of water injected varies between 0.4-0.9kg water/kgfuel. The corresponding values for steam injection are between 0.8-1.2kg/kgfuel.

SO_x Emissions (effect: corrosive, toxic)

Sulphur present in the fuel appears in the exhaust gases as sulphur oxides, primarily as sulphur dioxide (SO_2). If it is considered that all sulphur is burned to SO_2, then the mass of emitted SO_2 is

$$m_{SO_2} = 2\left(1 - r_{SO_2}\right) s\, m_f \tag{5-10}$$

where: m_{SO_2} mass of emitted SO_2 with the exhaust gases,

r_{SO_2} SO_2 retention factor,

s mass content of sulphur in fuel (e.g. kg S/kg fuel),

m_f mass fuel consumption.

For liquid and gaseous fuels, it is $r_{SO_2} = 0$. For solid fuels burned on a grate or in a fluidised bed, retention of part of SO_2 may occur in the solid material. In such a case, it is $r_{SO_2} > 0$. The exact value of r_{SO_2} depends on the particular equipment. With natural gas, SO_x is usually of no concern, because the sulphur content in the fuel is very low. It is possible to remove up to 95% of SO_2 from the exhaust gases by flue gas desulphurisation techniques using, e.g., water and limestone (post-process abatement). These techniques are applied on rather large plants. For smaller systems, like those of small to medium size cogene5ration, the use of low-sulphur fuel is more economical; fuels with a high sulphur content (e.g. fuel or Diesel oil) are chemically treated in the refinery and low-sulphur fuels are produced (pre-process abatement). In case of solid fuels burned on a grate or in a fluidised bed, retention of SO_2 by mixing limestone with the combustible material is also possible (process abatement).

Emissions of particulates (effect: visible)

Particulates are of concern primarily for plants burning solid fuel, e.g. coal, and for Diesel engines burning fuel oil or Diesel oil (*Fig.5.7A*). For the former, filters or scrubbers are installed. For the latter, good quality fuel and proper control of combustion are the means to keep particulates emission at acceptable levels.

Emissions Balances

It is useful to compare a cogeneration system with the separate production of electricity and heat (systems replaced by cogeneration) from the point of view of pollutant emissions. This can be done with an emissions balance for each pollutant. However, the balance equation, and consequently the result, depends on the boundary of the region under study. In the separate production of electricity and heat, electricity usually comes from central power plants, which are far from the cogeneration site, while heat in produced locally by boiler(s). If all sources of pollutants are taken into consideration, no matter where they are, *a global balance* is obtained. If only on site sources are considered, *a local balance* is obtained.

Examples of global emissions balances for six different combinations of cogeneration systems and systems for separate production of electricity and heat are given in *Table 5.8*. Specific emissions from *Table 5.6* have been used. As the examples demonstrate, an impressive reduction of CO_2 emissions is achieved: 50-100 kg per 100 kWh of cogenerated electricity. Even with the lower value, i.e. 50kg/100 kWh_e, for every TWh (10^9 kWh) of cogenerated electricity, a reduction of 500,000 tons of CO_2 emissions is achieved. When natural gas replaces other fuels, such as fuel oil, emissions of SO_x and particulates nearly vanish (a reduction by 90-99.8% is achieved).

Another application example, global and local emissions balances of a gas engine cogeneration system as compared with three different combinations of systems for separate productions of electricity and heat have been performed. Data and results are given in *Table 5.9*. Values of specific emissions have been taken not from *Table 5.6* but from data available for particular systems. The importance of the boundary of the region, for which the analysis is performed, is revealed by the results of *Table 5.9*. [11][78][80][85]

Table 5.8: Examples of global emissions balance: comparison of cogeneration with separate production of electricity and heat (results per 100 kWh.)[78]

Pollutant	Systems under comparison											
	C1 – S1		C1 – S2		C2 – S1		C2 – S2		C3 – S1		C3 – S2	
	gr	%	gr	%	gr	%	gr	%	gr	%	gr	%
CO_2	-51024	-46.2	-88458	-59.9	-62454	-52.0	-99888	-64.3	-70791	-46.7	-108225	-57.2
CO	+320	+524.6	+357	+1487	-33	-52.4	+4	+15.4	-68	-100.0	-31	-100.0
NO_x	+812	+255.3	+802	+244.5	-290	-85.3	-300	-85.7	-283	-68.7	-293	-69.4
HC	+375	+1875	+388	+5543	-15	-75.0	-2	-28.6	+4	+18.2	+17	+188.9
SO_x	-208	-95.9	-794	-98.9	-273	-99.3	-859	-99.8	-415	-90.0	-1001	-95.6
Particulates	-44	-91.7	-40	-90.9	-51	-91.1	-47	-90.4	-77	-91.7	-73	-91.3

Cogeneration systems

C1. Dual-fuel Diesel engine (90% of energy from natural gas, 10% from Diesel oil); $\eta_e = \eta_{th} = 0.35$ (PHR = 1).

C2. New gas turbine fueled with natural gas; $\eta_e = 0.35$, η_{th}=0.45 (PHR = 0.778).

C3. New steam turbine fueled with natural gas; $\eta_e = 0.25$, $\eta_{th} = 0.55$ (PHR = 0.455).

Systems for separate production of electricity and heat

S1. Gas turbine fueled with Diesel oil and industrial steam boiler with fuel oil.

S2. New steam turbine plant fueled with coal and industrial steam boiler with fuel oil.

Negative sign indicates reduction of emissions with cogeneration.

Percent values are given with reference to the separate production of electricity and heat.

Table 5.9: Example of annual global and local emissions balances of a gas turbine cogeneration system [78].

System specifications: Natural gas, $\dot{W} = 1000\ kW_e$, $\dot{Q} = 1300\ kW_{th}$, $\eta_e = 0.38$, $\eta_{th} = 0.494$

Operation: 5000 h/a

Specific emissions of systems (gr/kWh of useful energy)

System	CO_2	CO	NO_x	SO_x	HC	Particulates
Gas engine	531.7	2.5	1.7	≈0	4.5	≈0
Power plant-Lignite	1250	0.18	1.2	1.5	0.05	1.5
Power plant-Fuel oil	900	0.18	1.6	14.5	0.05	1.4
Boiler-Diesel oil (0.2%S)	323	0.06	0.25	0.37	0.02	0.03
Boiler-Natural gas	253	0.03	0.19	≈0	0.02	≈0

Global emissions balances (kg/a)

Case	Power plant	Boiler	CO_2	CO	NO_x	SO_x	HC	Particulates
1	Lignite	Diesel oil	- 5,691,500	+ 11,210	+ 875	- 9,905	+ 22,120	- 7,695
2	Fuel oil	Diesel oil	- 3,941,500	+ 11,210	- 1,125	- 74,905	+ 22,120	- 7,195
3	Fuel oil	Natural gas	- 3,486,500	+ 11,405	- 735	- 72,500	+ 22,120	- 7,000

Local emissions balances (kg/a)

Case	Power plant	Boiler	CO_2	CO	NO_x	SO_x	HC	Particulates
1	Lignite	Diesel oil	+ 558,500	+ 12,110	+ 6,875	- 2,405	+ 22,370	- 195
2	Fuel oil	Diesel oil	+ 558,500	+ 12,110	+ 6,875	- 2,405	+ 22,370	- 195
3	Fuel oil	Natural gas	+ 1,013,500	+ 12,305	+ 7,265	0	+ 22,370	0

* Negative sign indicates reduction while positive sign incretion

5.2.9 Electricity price of the national grid

Figs. 5.9, 5.10 show the average electricity prices of two consumers sectors for the year 2004. The average price of electricity energy for **commercial** use is **5.5€cent/kWh**, (2004).

As it can be seen the prices in Greece are relatively low. This is due to the fact that 60%-65% of the total Electric Power of Greece is coming from Thermal Power Plants burning lignite. These power plants supply the interconnected system while the non-interconnected (islands, including Crete), are supplied by Power Plants burning diesel (see Paragraph 2.7.2). It is evident that the above prices might be up to 100% higher when referring to the non- interconnected grid consumers. On the other hand, the average electricity price in Greece is increasing almost equally to the average national inflation rate.

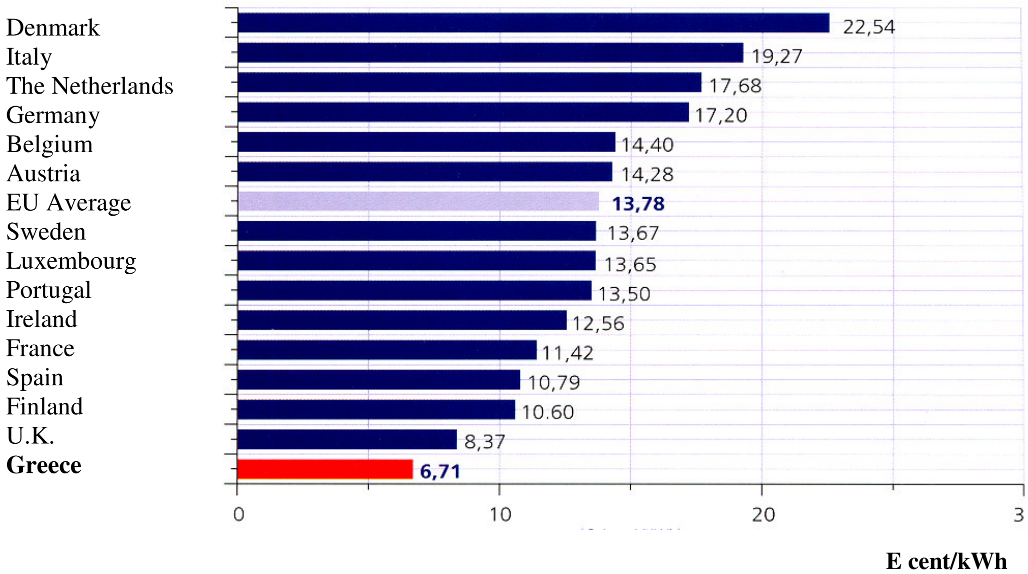


Fig.5.9: Final selling price (including taxes) for typical domestic consumer, with annual consumption3.5MWh and annual night consumption 1.3MWh.(2004) [101]

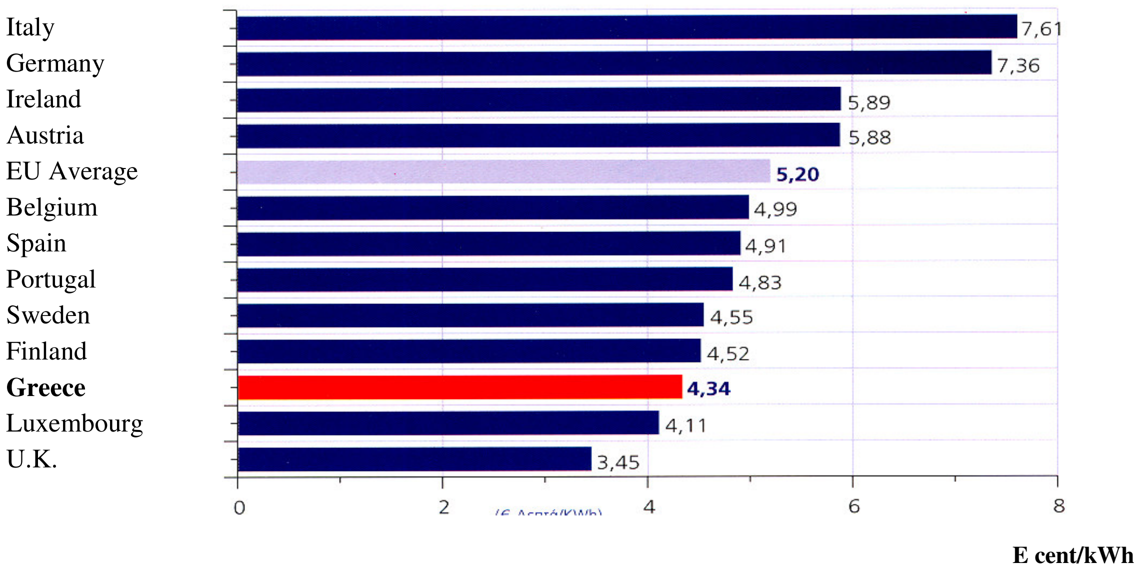


Fig.5.10: Selling price (before taxes) for typical industrial consumer, with annual consumption 70MWh and annual night consumption 1.3MWh.(2004) [101]

Every public or private producer is selling energy to the Hellenic Transmission System Operator, (HTSO). The HTSO is buying electricity coming from **cogeneration plants**, in the price of **6.8 or 7.88€cent/kWh (2004),** dependently to the type of the national system is the plant supplying power, namely interconnected or non- interconnected.

To give an idea of the Greek economic energy policy, it is refer that the corresponding prices (2004), for **renewable sources** are: [101]

- Wind: 6.8 or 7.88€cent/kWh
- Sun: 6.8-41.9 or 7.88-46.5€cent/kWh depending on the power output.
- Geothermic: 6.8 or 7.88€cent/kWh

5.2.10 Financing

Although trigeneration is a long-term investment, with equipment lifetimes of up to forty years, in most cases it has to compete with other potential business projects that are expected to yield rapid returns. In addition, since cogeneration is often not considered to be core business plant, it receives a lower priority. These factors may mean that schemes fall outside a company's investment criteria for utility plant so alternative methods of financing often need to be investigated if cogeneration is to be implemented. The source of finance, ownership and degree of risk are the main factors to be taken into account. If financed by direct capital injection using equity funds, debt or a combination of both, the purchaser takes on full ownership and risk. The risk will normally be offset by the terms of contract negotiated with all relevant parties.

There are two basic alternatives that may help to overcome the problems of justifying full self-financing of cogeneration. These schemes also have an effect on ownership and risk:

- to lease the plant, whilst undertaking all aspects of operation and maintenance; In this method, also known as equipment supplier leasing (ESL), the cogeneration unit is installed and owned by the equipment supplier. The host agrees to purchase heat and/or electricity at a discounted rate for a fixed contract period, often five or ten years. The price includes an element to cover maintenance, is subject to annual review and is often linked to the prevailing utility prices. Risk to the energy user is minimal but typically, only around 25% of the savings are passed on. Hosts may choose to purchase the unit after a few years of operation and thereafter benefit from all of the savings.
- to offer the opportunity to a Energy Service Company (ESCO) company which will manage the facility and may finance and own the installation as well. This third party company would offer the client guaranteed outputs, thermal and electrical, at a discount against present and projected costs. ESCOs can provide a complete range of services, from design, finance, installation, operation, maintenance and monitoring. Most importantly, the ESCO should undertake the technical risk, whilst sharing the economic risk and profit with the client.

Numerous variations of these basic concepts are available: contracts are negotiated between the ESCO and the client to take account of the particular circumstances and requirements of the site. These include Build Own and Operate (BOO), Build, Own, Operate and Transfer (BOOT), and Joint Venture Company.

The choice between these types of contract is dependent upon the nature of the cogeneration (large or small), the company's investment and accounting policy, the level of financial risk the purchaser is willing to bear and the financial return required. A number of financing options are available. Capital purchase is the traditional option where a company raises, or borrows, the investment itself. It has the highest associated risk but ail of the savings are returned directly to the host. Obtaining project finance from commercial bankers is often difficult to arrange for projects of this scale. A minimum investment of, typically, USD 15 million is required because of the necessary complexities of establishing contracts and satisfactorily spreading the risk. Many organizations, particularly in the pubic sector, find that raising capital internally can be very awkward and one of the following options can be more attractive. In the frame of new National Energy Policy the **Greek government is offering financial support (grant) up to 40% of the initial cost of the plant**. [72] [80] [81].

5.3 Economic model

5.3.1 Long-term decision-making

In economic analysis, certain parameters are used to evaluate measures of economic performance, which are used as criteria for any decisions regarding the investment. A measure or index of economic performance is used either as an indication of whether an investment (e.g. in a trigeneration system) is viable in itself, or as a basis for comparison among alternative investments (e.g. among various trigeneration systems or among trigeneration and completely different activities). The most common measure, which is appropriate also for investments in trigeneration, is defined below. In certain cases, there is need of a reference system, for comparison. If not otherwise specified, the conventional approach for covering electrical cooling and thermal needs will be considered as reference, i.e. purchase of electricity from the grid and production of heat by a boiler on site.

Net present value of the investment (NPV)

It is called also *net present worth.* It is the present worth of the total profit of an investment, which results as the difference between the present worth of all expenses and the present worth of all revenues, including savings, during the life cycle of the investment (system). A general expression for the net present value is

$$NPV = \sum_{t=0}^{N} \frac{F_t}{\left(1+d_t\right)^t} \tag{5-11}$$

or

$$NPV = \sum_{t=0}^{N} \frac{F_t}{\left(1+d\right)^t} \tag{5-12}$$

where

d_t is the *market interest rate* during the period t, and when it is considered constant $d_t=d$. N is the number of periods, for which the investment is assumed to operate. Any time period can be used: day, month, six-months, year, etc; a year is the most usual one. F_t is the *profit or net cash flow* (revenue + savings – expenses) in *year t.* The term "profit" here is used with a general meaning: F_t can be negative, when the net result of year t is a loss. F_0, in particular, usually represents the present worth of the investment (t = 0) and it is negative.

If the construction period has lasted for a few years, Eq. (5-13) or (5-14) can be used to calculate the present worth of each year's expenses. Their summation is F_0.

The present worth of a past cash flow can be determined by the equation

$$P = F \cdot \prod_{t=-1}^{-n} \left(1+d_t\right) \tag{5-13}$$

where n is the number of construction periods or, if d_t is considered constant:

$$P = F\left(1+d\right)^n \tag{5-14}$$

Since d is used to discount future amounts to their present worth, it is called also *market discount rate*

There are three characteristic situations:

NPV>0: The investment is economically viable under the specified conditions (N, d). The return on investment is higher than d.

NPV=0: The investment is economically viable and it has a return on the investment equal to d.

NPV<0: The investment is not viable economically, under the specified conditions (N, d). [78]

5.3.2 Procedure for economic analysis of cogeneration systems

In order to calculate the value of each measure using the equations of the previous paragraph, there is need to estimate (a) the initial cash flow, F0, and (b) the net cash flow Ft in year t≥1. A procedure for these estimates is presented in the following.

a. Initial Cash Flow (F_0, t=0)

Vendor quotations or information from other sources (see paragraph 5.2) is used to estimate the investment cost C of the system, where C is considered as the present worth of the cost (time t = 0). If there is need, instructions given paragraph 5.3.1, can be used to determine the present worth of costs occurring during the years of construction.

In certain countries (among them Greece), investment grants are provided for promotion of cogeneration. They are given with no obligation on cogenerator's side, other than observing certain standards, in particular regarding the real operating efficiency of the system. In addition, the investor may borrow a certain amount of money from a bank or another institution. In order to take these possibilities into consideration, it is written

$$F_0 = C_g + L - C = \left(c_g + \ell - 1\right)C \qquad (5\text{-}15)$$

where
C, investment cost of the system,
C_g, amount of grant,
L, amount of loan,
c_g, grant as a fraction of the investment cost: $c_g = C_g/C$,
l, loan as fraction of the investment cost: l = L/C.

Of course, zero values for C_g or L are acceptable and do not cause any problems with the rest of the calculations.

Assumptions

- **The construction duration is one year, so the investment cost C, is paid only in the first year, t=0 (corresponds to the year 2003)**
- **In all following scenarios it is assumed that c_g = 0.4 and l = 0.**

b. Net Cash Flow for the Years of Analysis (F_t, t ≥ 1)

<u>Annual operation profit</u>

The operation of a trigeneration system causes expenses, but it also results in savings (avoided cost of electricity that otherwise would be purchased from the grid and heat that would be produced by a boiler), and also in revenues, if excess electricity is sold. The *annual operation profit* of the cogeneration system is defined as

$$f_t = (C_e + R_e + C_h + C_r - C_f - C_{om}) \qquad (5\text{-}16)$$

where C_e avoided cost of electricity, i.e. cost of electricity that, if not cogenerated, it would be purchased from the grid. The avoided cost of electricity, C_e, is a function of the cogenerated electricity which is consumed on site, and on the tariff structure for electricity supplied by the grid, which may consider not only energy, but also power, power factor, time of the day, peak demand, etc. A cost component which is often overlooked, but it may be non-negligible, is an increase of the electricity bill due to taxes: utilities often include some tax imposed on behalf of a government body (state and/or local municipality). For example, such a tax of 8% on the total cost of electricity is imposed on apartments in Athens. Since all these parameters are site-specific, it is not possible to give a general expression for C_e here.

R_e revenue from selling excess electricity, if any. The revenue R_e from excess electricity sold to the grid or to a third party is a function of the electrical energy and of the tariff structure for electricity sold to the grid or of the agreement between the parties involved. For the same reasons as with C_e, no general expression for R_e will be attempted here.

C_h avoided cost of heat, i.e. cost of heat that, if not cogenerated, would be produced by boiler(s). The avoided cost of heat includes cost of fuel for the boiler that would produce the thermal energy, if not cogenerated, as well as other operation and maintenance expenses for the boiler and related auxiliary equipment. The fuel cost is a function of the fuel quality and the fuel tariff structure. Capital cost of boiler is usually not taken into consideration, because it is assumed that a boiler would be installed anyway for back up. However, if this is not the case, then the capital cost of boiler should be included.

Cogenerated heat can be used to drive an absorption air conditioning unit, in which case the compression air conditioning unit is not operated. Then C_r is the avoided costs related to the compression unit (should be included with a positive sign), plus operating costs related to the absorption unit (should be included with a negative sign). In such a case, proper modification of the investment cost might be required, depending on assumptions about the reference system and the alternative configuration.

C_f cost of fuel for the cogeneration system. The cost of fuel for the cogeneration system is a function of the fuel quantity and the fuel tariff structure

C_{om} operation and maintenance cost (except fuel) of the cogeneration system.

Subscript t indicates the year (t = 1, 2, …, N).

Assumptions

- **A trigeneration power plant proposed here is not similar with an ordinary company or commercial investment from the point of view of profit and expenses. The managerial authority of the airport or the hotel or power corporation, which operating a trigeneration power plant, is not really earning money operating that plant, but actually saving money compared with the operation of a conventional power, heat and cooling producing case. Having this in mind the author decided to calculate the NPV (negative) of the conventional case and compare it with the NPV's of the different scenarios (also negatives, because $C_e = C_h = C_r = 0$). In other words, it is consider no virtual profits.**
- **The operation of the plant starts from the beginning of the first year, t=1 (2004). Usually, these types of investments are investigated for a time period of 15, 20 or even 25years. In this thesis is a time period of 20 years (2004-2024) is chosen.**
- **The scenario with the lower absolute NPV will be the recommended.**

Annual net cash flow (F_t)

In order to determine the annual net cash flow due to the investment in cogeneration, there is need to know the taxation system, the terms of loan (if any), and the method of depreciation. Certain assumptions can be made in the following, which will allow completing the procedure. Proper modifications will be necessary for different conditions. The following equation can be used

$$F_t = f_t - A_{Lt} - r_T T_t + SV_N, \quad t = 1, 2, \ldots, N \qquad (5\text{-}17)$$

where

F_t :net cash flow in year t,

f_t :operation profit in year t, Eq. (5-11),

A_{Lt} :equal yearly payments of principal and interest for repayment of the loan,

r_T ;tax rate,

T_t :taxable income in year t, due to cogeneration,

SV_N :salvage value of the investment at the end of the economic life cycle, i.e. at the end of year N.

Assumptions

- **For simplicity reasons: $A_{Lt} = T_t = SV_N = 0$**

At this point the procedure is completed: everything needed for calculating the measures of economic performance of an investment in cogeneration is obtained by means of the previous equations and the accompanying instructions.

A comment should be made here: The procedure presented in this chapter is based on certain considerations and assumptions. In spite of the attempt to be generally applicable, it is impossible to incorporate all the different situations that are encountered in practice. It is left to the reader to make the modifications that may be needed for each particular application. [78]

5.4 Airport energy scenarios

5.4.1 Conventional case

In this paragraph, the different costs of energy will be analytically presented, for the conventional, namely the present energy situation of the New Macedonia Airport. Essential assistant to that will be given by the data presented in CHAPTER 2. *Tables 2.3* and *2.4* are presenting the power demand and the energy consumption respectively, for a typical day each month of the year. As it has been said in paragraph 2.1, these values do not include either the hypothetical future increase, or the estimation of the worst case for the energy demand point of view. After a relevant discussion with the supervisor, the author decided to multiple by a factor of 1.2 all the prices of the above mentioned *Tables*, in order to include the worst case situation. The results are shown in *Tables 5.10* and *5.11*

Table 5.10: Airport, power demand in MW

MONTHS (30 days per month)	**HEATING** (MW_{th})	**COOLING** (MW_c)	**LIGHTING & MOTION** (MW_e)	**TOTAL** (MW)
JAN	8.100	2.776	2.197	13.073
FEB	7.200	4.000	2.134	13.333
MAR	4.950	7.225	2.102	14.278
APR	2.700	8.790	2.038	13.528
MAY	0.676	10.426	2.040	13.141
JUN	0.046	11.950	1.974	13.969
JUL	0.046	11.926	1.968	13.939
AUG	0.046	11.890	1.981	13.916
SEP	0.450	10.950	2.015	13.415
OCT	2.700	8.820	2.096	13.616
NOV	3.150	6.830	2.206	12.186
DEC	6.750	3.826	2.231	12.806

Table 5.11: Airport energy consumption in MWh

MONTHS (30 days per month)	**HEATING** (MWh_{th})	**COOLING** (MWh_c)	**LIGHTING & MOTION** (MWh_e)	**TOTAL** (MWh)
JAN	194.4	66.6	52.741	313.741
FEB	172.8	95.988	51.216	320.015
MAR	118.8	173.388	50.455	342.654
APR	64.8	210.96	48.904	324.664
MAY	16.2	250.2	48.949	315.349
JUN	1.08	286.788	47.375	335.254
JUL	1.08	286.2	47.222	334.502
AUG	1.08	285.348	47.557	333.996
SEP	10.8	262.8	48.354	321.954
OCT	64.8	211.68	50.328	326.808
NOV	75.6	163.908	52.921	292.439
DEC	162.0	91.8	53.542	307.342

1. <u>Cost of electricity (lighting, motion, etc)</u>

The electricity cost of each month (€/month) is

$$[€/\text{month}] = [\text{MWh}] * [€/\text{MWh}] * 30 \qquad (5\text{-}18)$$

where MWh is the corresponding to each month value of the cells of column 4, *Table 5.11*, €/MWh is the price of electricity per MWh (see paragraph 5.2.9) and 30 is assumed the number of days of the month.

The electricity cost of the year (12 months) is

$$\text{Cost of Electricity} = \sum_{1}^{12} (€/\text{month}) = \mathbf{[€/year]} \qquad (5\text{-}19)$$

which varies accordingly to the inflation rate

2. <u>Cooling</u>

Electric compression refrigeration system.
The capital cost plus the installation cost -**which has fix value-** is

$$\text{Capital Cost + Installation Cost} = \text{MW} * \text{€/MW} = \textbf{€} \qquad (5\text{-}20)$$

where MW is the maximum value of the cells of column 3, *Table 5.10*, and €/MW is the price of electricity per MW (see paragraph 5.2.4)

The electric energy per month supplied from the local grid for cooling ($W_{e,\,c}$), can be calculated as follows:

$$\text{COP} = 4.5 \Rightarrow (\text{Eq. 4-3}) \Rightarrow W_{e,\,c} = \text{MWh} \,/\, 4.5 = \text{MWh} \qquad (5\text{-}21)$$

where MWh are the corresponding values of cells of column 3, *Table 5.11*.
Then the operation cost per month is given by the following equation:

$$\text{Operation Cost per month} = W_{e,\,c} * [\text{€/MWh}] * 30 = \text{€/month} \qquad (5\text{-}22)$$

where €/MWh is the price of electricity per MWh (see paragraph 5.2.9) and 30 is assumed the number of days of the month.
The operation cost of the year can easily calculated as

$$\text{Operation cost} = \sum_{1}^{12} (\text{€/month}) = \textbf{€/year} \qquad (5\text{-}23)$$

which varies accordingly to the inflation rate

Finally, the maintenance cost -**which has fix value, for every year-** is given accordingly to paragraph 5.2.7:

$$\text{Maintenance Cost} = (\text{Capital cost + Installation cost}) * 0.90 * 0.03 = \textbf{€/year} \qquad (5\text{-}24)$$

We assume that the maintenance of the cooling system is taking place in January (last week)

3. <u>Heating</u>

The boilers (power in MW) operate using medium heating oil (*Table 5.4*: ρ =0.92kgr/lt, FCV=41MJ/kgr). The capital cost -**which has fix value-** is given by

$$\text{Capital Cost} = \text{MW} * \text{€/MW} = \textbf{€} \qquad (5\text{-}25)$$

where MW is the maximum value among the cells of column 2, *Table 5.10* and €/MW is the price of boiler per MW (see paragraph 5.2.3)

The installation cost -**which has fix value-** is given by

$$\text{Installation cost} = \text{Capital Cost} * 0.10 = \textbf{€} \qquad (5\text{-}26)$$

where factor 0.10 is explained in paragraph 5.2.3

The energy provided from the medium heating oil per month is calculated as follow:

$\eta_{th,\,b} = 0.8 \Rightarrow$ <Eq. (4-3)> $\Rightarrow Q_{f,\,b} = (\text{MWh} * 30) \,/\, 0.8 = \text{MWh per month}$ (5-27)

where $\eta_{th,\,b}$ is given in paragraph 5.2.3, MWh are the corresponding values of cells of column 2, *Table 5.11* and 30 is assumed the number of days of the month.

The mass of medium heating oil per month and per year are calculated as follow:

$$m_{fm,b} = [(Q_{f,\,b} * 3{,}600) \,/\, \text{FCV}\,] = \text{kgr/month} \qquad (5\text{-}28)$$

$$m_{fy,b} = \sum_{1}^{12} m_{fm,b} = \text{kgr/year} \qquad (5\text{-}29)$$

where FCV is taken from *Table 5.4*.

And so the cost of medium heating oil per year -**which varies accordingly to the international oil prices,** can be estimated:

Cost of medium heating oil = (($\dot{m}_{fy,b}$ **/** 0.92) $*$ (2.9t+42.1) **/** 159) $*$ 1.7 = **€/year** (5-30)

Where 0.92 is the density ρ of the medium heating oil (paragraph 5.), (2.9t+42.1) is the crude oil price equation (5-1), while the factors 159 is the capacity of barrel (paragraph 5.2.7) and 1.7 is resulting from paragraph 5.2.7.

Finally, the maintenance cost -**which has fix value, for every year-** is given accordingly to paragraph 5.2.7:

Maintenance Cost = Capital cost $*$ 0.02 **= €/year** (5-31)

We assume that the maintenance of the heating system is taking place in June (last week)

4. Back up generator

The capital cost plus the installation cost -**which has fix value-** is

Capital Cost + Installation Cost **=** MW $*$ €/MW **= €** (5-32)

where MW is the maximum sum of cells of columns 3 and 4, *Table 5.10*, and €/MW is the price of electricity per MW (see paragraph 5.2.6)

5. CO_2 emissions estimation and penalty

The energy per year supplied from national grid (PPC) is calculated as follow:

$$\text{Energy per year} = \sum_{1}^{12} \text{values per typical day} * 30 = \text{MWh/year} \qquad (5\text{-}33)$$

where the values per typical day are the cells of columns 3 and 4 of *Table 5.11* and 30 is assumed the number of days of the month.

The airport is connected with the interconnected national grid (continental) of Greece. Taking into account the data from *Table A.5*, it can be estimated how many MWh/year are produced from the available kinds of power plants, assuming that analogical distribution of the total electric energy per year:

Lignite: MWh/year $*$ 0.674

Natural gas: MWh/year $*$ 0.168

Diesel (heavy heating oil): MWh/year $*$ 0.056

Using the CO_2 calculation method presented in paragraph 5.2.8, the m_{CO2} produced from every type of power plant respectively, can be estimated:

Eq.(5-6) ⇒ [Eqs (5-7), (5-8) (*Table 5.7*)] ⇒ $\mathbf{m_{CO2}}$ **= kgr$\mathbf{CO_2}$/year** (Assuming η_{LIG}=0.3)

Assuming that lignite power plants exceed the CO_2 emission limit at about 10% then the mass of CO_2, which must be accounted for penalty, will be:

$\mathbf{m_{CO2,Li,pen}}$ **= kgr$\mathbf{CO_2}$/year** $*$ 0.1 = **kgr$\mathbf{CO_2}$/year**

Similarly (5-6) ⇒ (5-7), (5-8) (*Table 5.7*) ⇒ $\mathbf{m_{CO2}}$ **= kgr$\mathbf{CO_2}$/year** (Assuming η_{NG}=0.55)

Assuming that NG power plants exceed the CO_2 emission limit at about 0% then the mass of CO_2, which must be accounted for penalty, will be:

$\mathbf{m_{CO2,NG,pen}}$ **= kgr$\mathbf{CO_2}$/year** $*$ 0.0 = **kgr$\mathbf{CO_2}$/year**

Finally, (5-6) ⇒ (5-7), (5-8) (*Table 5.7*) ⇒ $\mathbf{m_{CO2}}$ **= kgr$\mathbf{CO_2}$/y** (η_{DIE}=0.36, APPENDIX B.3)

Assuming that diesel (heavy) power plants exceed the CO_2 emission limit at about 7% then the mass of CO_2, which must be accounted for penalty, will be:

$\mathbf{m_{CO2,Di,pen}}$ **= kgrCO_2/year** $*$ 0.07 = **kgrCO_2/year**

Thus, **CO_2 emission cost -which varies with the CO_2 penalty price-** paid by PPC is

CO_2 emission cost paid by PPC **=** ($m_{CO2,Li,pen}$ + $m_{CO2,NG,pen}$ + $m_{CO2,DiH,pen}$) $*$ (€/1,000)=

= **€/year** (5-34)

where factor 1,000 is for units similarity (paragraph 5.2.8)

The airport is using boilers burning only medium heating oil. Thus the electric energy per year produced burning medium heating oil per year will be:

$$\text{Heating energy per year of boilers}=\sum_{1}^{12}\text{(values per typical day)}*30=\text{MWh/year} \quad (5\text{-}35)$$

where the values per typical day are the cells of column 2, *Table 5.11*and 30 is assumed the number of days of the month.

Using the CO_2 calculation method presented in paragraph 5.2.8, the m_{CO2} produced by boilers burning medium heating oil will be:

(5-6) $\Rightarrow$ (5-7), (5-8) (*Table 5.7*)$\Rightarrow$ $\mathbf{m_{CO2}}$ **= kgrCO_2/year** (η_{DIE}=0.8, APPENDIX B.3)

Assuming that boilers burning medium heating oil exceed the CO_2 emission limit at about 6% then the mass of CO_2, which must be accounted for penalty, will be:

$\mathbf{m_{CO2,Di,b,pen}}$ **= kgrCO_2/year** $*$ 0.06 = **kgrCO_2/year**

Thus, **CO_2 emission cost -which varies with the CO_2 penalty price-** paid by airport company

CO_2 emission cost paid by airport company **=** $m_{CO2,Di,b,pen}$ $*$ (€/1,000) = **€/year** (5-36)

where 1,000 is for units similarity (paragraph 5.2.8)

Total CO_2 emission cost = CO_2 emission cost paid by PPC + CO_2 emission cost paid by airport company = **€/year**

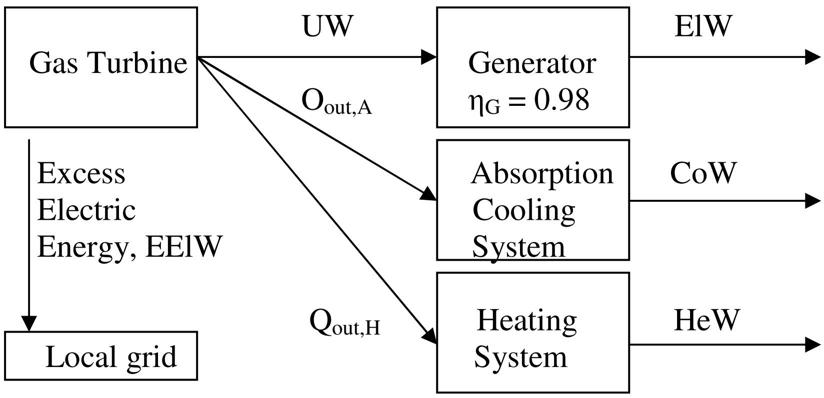
5.4.2 Hypothetical operation scenarios

A scenario-mode of operation is characterised by the criterion on which the adjustment of the electrical and useful thermal-cooling output of a trigeneration system is based. There are various modes of operation possible, the most distinct of those being the following:

Scenario 1: Maximum capacity GT, operation scenario

There is complete coverage of the electrical, thermal and cooling loads at any instant of time. The possible excess in electric power supplies the national local grid.

The block diagram of the technical configuration is showed in *Fig. 5.11*.



Notice: **Continue arrows** display definite transfer

Fig. 5.11: Technical block diagram of the scenario 1

In this mode the following points (key points) must be taken into consideration:

1. Selection of the GT power (choosing the TET, R_c from diagrams *Fig. 3.2, 3.4, 3.6*) and regulating the mass flow of the engine at the DP performance in such way to cover the electric, heating and cooling power and energy demand of the most energy-demanded month. The yearly operation is characterised by constant TET (the same with the DP), while the ambient conditions P_a T_a are vary according to the conditions referred in *Table 2.1* (OD performance).
2. The selected engine has the best η_{th} for the maximum TET without cooling system.
3. The proportional factor z, is representing the way of the power or the energy of the GT exhaust gasses is split between the heating and cooling demand every month.
4. The availability of the plant is assumed to be about 98%, in other words the plant is assumed to shut down for one week (the first of November, when total needs are minimum) for the annual service of the entire system. During that week electricity is supplied from the local grid while heating is supplied from a stand by boiler.
5. Salaries for extra personnel are assumed to be negligible.

Having the above in mind the economic simulation procedure for the 1-Shaft GT, is as following:

1-Shaft GT, 2-Shaft GT,

- One GT package cost **-which has fix value-** including the necessary generator, the contribution devices, while the price of the gear box is assumed relatively negligible:

$$\text{GT Package Cost} = [\text{MW}] * [\$/\text{kW}] * 1000 / 1.23 = \textbf{€} \qquad (5\text{-}37)$$

where MW is the useful work of the GT at the design point, $/kW is the corresponding price of column (see APPENDIX E.1), the factor 1000 is due to transformation from kW to MW and finally factor 1.23 is due to transformation from $ to €.

The GT package installation cost **-which has fix value-** is given by the following equation:

$$\text{GT Package Installation Cost} = \text{GT Package Cost} * 0.1 = \textbf{€} \qquad (5\text{-}38)$$

where factor 0.1 is explained in paragraph 5.2.1.

The GT package maintenance cost **-which has fix value, for every year-** is given by the following equation:

$$\text{GT Package Maintenance Cost} = \text{GT package cost} * 0.01 = \textbf{€/year} \qquad (5\text{-}39)$$

where factor 0.01 is explained in paragraph 5.2.7.

- The heat exchanger cost **-which has fix value-** including the installation cost, is given by the following equation:

$$\text{Heat Exchanger Cost} = [\text{MW}] * [€/\text{MW}] = \textbf{€} \qquad (5\text{-}40)$$

where MW is the maximum cell of columns 2 plus 3 (*Table 5.10*) and €/MW is explained in paragraph 5.2.3.
The heat exchanger maintenance cost **-which has fix value, for every year-** is given by the following equation:

$$\text{Heat Exchanger Maint. Cost} = 0.9 * [\text{Heat Exchanger Cost}] * 0.02 = \textbf{€/year} \qquad (5\text{-}41)$$

where factors 0.9 and 0.02 are explained in paragraphs 5.2.3 and 5.2.7 respectively

- The district heating installation cost **-which has fix value-** is given by the following equation:

$$\text{District Heating Installation Cost} = [\text{MW}] * [€/\text{MW}] * ([\text{MW}]/120) = \textbf{€} \qquad (5\text{-}42)$$

where MW is the maximum cell of columns 2 plus 3 (*Table 5.10*), €/MW is the corresponding price (paragraph 5.2.5), and the factor 120 is due to the relatively small system (paragraph 5.2.5).

- The absorption chiller cost **-which has fix value-** including the installation cost is given by the following equation:

$$\text{Absorption Chiller Cost} = [\text{MW}] * [€/\text{MW}_c] = \textbf{€} \qquad (5\text{-}43)$$

where MW is the maximum cell of column 3 (*Table 5.10*) and €/MW_c is the corresponding price (paragraph 5.2.4).
The absorption chiller maintenance cost **-which has fix value, for every year-** is given by the following equation:

$$\text{Absorption Chiller Maint. Cost} = 0.8 * \text{Absorption chiller cost} * 0.031 = \textbf{€/year} \qquad (5\text{-}44)$$

where factors 0.8 and 0.031 are explained in paragraphs 5.2.4 and 5.2.7 respectively.

- Cost of back up cooling.

  The electric compression refrigeration system capital cost including the installation cost **-which has fix value-** is given by the following equation:

  Electric Compression Refrigeration System Capital Cost + Installation Cost **=**

  $$= 0.5 * [MW] * [€/MW] = \textbf{€} \qquad (5\text{-}45)$$

  where the factor 0.5 is due to the assumption that the back up cooling power is the 50% of the maximum cooling demand power in MW, MW is the maximum value of cells of column 3, (*Table 5.10*) and €/MW is the corresponding price (paragraph 5.2.4)

  The operation cost of electric compression refrigeration system is calculated as follows:

  Operation back up cooling energy: COP=4.5⇒(Eq. 4-3)⇒$W_{e,\,c}$= [MWh]/4.5=MWh

  where $W_{e,\,c}$ is the electric energy supplied from the local grid for cooling and MWh corresponds to the November-cell of column 3 (*Table 5.11*)

  Operation cost in November = $W_{e,\,c} *$ [€/MWh] <paragraph 5.2.9> $*$ 7 = €/month

  where €/MW is the corresponding price (paragraph 5.2.9), and 7 is the number of November days when the back cooling system works.

  $$\text{Operation cost} = \sum_{1}^{12} (\text{€/month}) = \textbf{€/year} \qquad (5\text{-}46)$$

  which varies accordingly to the inflation rate.

  The maintenance cost **-which has fix value, for every year-** is given by the following equation:

  Maint. Cost=([Capital Cost]+[Installation Cost])$*$0.90$*$0.03$*$(7/360)**=€/year** (5-47)

  where factors 0.90 and 0.03 are discussed in paragraphs 5.2.4, 5.2.7, while the factor (7/360) is simulates the relative duration of the operation.

  We assume that the maintenance of the cooling system is taking place in January (last week)

- The NG mass flow per month is given by

  $$\text{NG mass flow per month} = \dot{m}_f * 60 * 60 * 24 * 30 = \dot{m}_{fm}\ \text{kgr/month} \qquad (5\text{-}48)$$

  where first factor 60 is for the conversion of seconds to minutes, second factor 60 is for conversion of minutes to hours, factor 24 is for the conversion of hours to days, while factor 30 corresponds to the number of days of the month, except from November which is assumed to operate 23 days, due to shut down for annual service

  Thus, the NG mass flow per year is

  $$\text{NG mass flow per year} = \dot{m}_{fy} = \sum_{1}^{12} \dot{m}_{fm}\ \text{kgr/year} \qquad (5\text{-}49)$$

  The cost of NG per year **-which varies accordingly to the international oil prices-** is given by the equation:

  Cost of NG per year = $\dot{m}_{fy} * 1.11 * 48.6 * 0.0002778 * (2.39t+14.61)$ = **€/year** (5-50)

  where factors 1.11 and 48.6MJ/kgr is accordingly to paragraph 5.2.7, while the factor 0.0002778 is due to transformation from MJ to MWh. Finally the factor (2.39t+14.61) is equation (5-5).

- Cost of back up boilers (assume 50% of the maximum heating demand power in MW) using medium heating oil. *Table 5.4*: ρ =0.92kgr/lt, FCV=41MJ/kgr
  Using the same methodology as in paragraph 5.4.1 and especially the part labeled heating. The capital cost (**fix value**) and the installation cost (**fix value**) can be calculated with the help of equations (5-25) and (5-26) respectively, while the maintenance cost (**fix value, for every year)** is given by the equation (5-31) with a slight modulation:

  $$\text{Maintenance Cost} = \text{capital cost} * 0.02 * (7/360) = \textbf{€/year} \qquad (5\text{-}51)$$

  where the factor (7/360) is due to the fact that operates regularly only 7 days per year
  The energy provided from the medium heating oil:
  $\eta_{th,\,b} = 0.8 \Rightarrow$<Eq. (4-3)>$\Rightarrow Q_{f,\,b}=(MWh*7)/0.8=MWh$ for 7 days in November (5-52)
  where $\eta_{th,\,b}$ is given in paragraph 5.2.3, MWh is the corresponding November-cell of column 2, *Table 5.11* and factor 7 is introduced because is operating only 7 days
  The mass of medium heating oil:

  $$m_{fm,b} = [(Q_{f,\,b}\ \mathbf{X}\ 3{,}600) / FCV = kgr \qquad (5\text{-}53)$$

  and finally the cost of medium heating oil **-which varies accordingly to the international oil prices-** is calculated accordingly to eq. (5-30).

- The connection to the grid cost -**which has fix value**- is given by the following equation:

  $$\text{Connection to the grid Cost} = MW * [€/MW_e] = \textbf{€} \qquad (5\text{-}54)$$

  where MW is the maximum difference between the monthly GT power production and the corresponding cell of column 4, *Table 5.10*, because that is the maximum power difference that might be sold to the local grid and €/MW$_e$ is the corresponding price (paragraph 5.2.6).

- The electricity cost **-which varies accordingly to the inflation rate -** is given by

  $$\text{Electricity cost} = [MWh] * [€/MWh]\ \mathbf{X}\ 7 = \textbf{€/year} \qquad (5\text{-}55)$$

  where MWh is the cell of column 3 plus 4, for November (*Table 5.11*), €/MWh is the corresponding price (paragraph 5.2.9) and factor 7 is due to the fact that no electric energy imported, except from the 1 week = 7 days in November, when service works are in process.

- The excess of electric energy per month in MWh is equal to energy produced by the GT when it is operating 24 hours per day for 30 days per month, -23 for November- minus cells for each month of column 4, (*Table 5.11*)

  The electricity profit per year **-which varies accordingly to the inflation rate -** is given by

  $$\text{Electricity Profit per year} = \sum_{1}^{12} \text{Excess of electric energy per month} * 30 * 68 = \\ = \textbf{€/month} \qquad (5\text{-}56)$$

  where factor 30 is the number of days per month and 68€/MWh is the corresponding price (paragraph 5.2.9).

- Greek government is offering financial support -**which has fix value**- is given by the following equation:

Greek Government Financial Support = 0.4 ∗ (GT package cost + GT package installation cost + Heat exchanger cost + District heating installation cost + Absorption chiller cost + Capital cost of back up boiler + Installation cost of boiler + Connection to the grid cost) = **€** (5-57)
where factor 0.4 is explained in paragraph 5.2.10

- CO_2 emissions estimation penalty.
  Total CO_2 emission cost = 0€ because it is assumed that trigeneration plants do not exceeds the official limits of CO_2 emissions. Emissions from the operation of the back up boiler or from the power plants of PPC to produce the electricity during the shut down period of the GT, are assumed negligible.

- CO_2 emissions estimation profit.
  The excess of energy per year is supplied to national grid is given by the equation
  Excess of energy per y= $\sum_{1}^{12}$ Excess of electric energy per month**=**MWh/year (5-58)
  The airport is connected with the interconnected national grid (continental) of Greece. Taking into account the data from *Table A.5*, it can be estimated how many MWh/year are produced from the available kinds of power plants, assuming that analogical distribution of the total electric energy per year:
  Lignite: MWh/year ∗ 0.674
  Natural gas: MWh/year ∗ 0.168
  Diesel (heavy heating oil): MWh/year ∗ 0.056
  Using the CO_2 calculation method presented in paragraph 5.2.8, the m_{CO2} produced from every type of power plant respectively, can be estimated:
  Eq.(5-6) ⇒ [Eqs (5-7), (5-8) (*Table 5.7*)] ⇒ $\mathbf{m_{CO2}}$ **= kgr$\mathbf{CO_2}$/year** (Assuming η_{LIG}=0.3)
  Assuming that lignite power plants exceed the CO_2 emission limit at about 10% then the mass of CO_2, which must be accounted for penalty, will be:
  $\mathbf{m_{CO2,Li,pen}}$ **= kgr$\mathbf{CO_2}$/year** ∗ 0.1 = **kgr$\mathbf{CO_2}$/year**
  Similarly (5-6) ⇒ (5-7), (5-8) (*Table 5.7*) ⇒ $\mathbf{m_{CO2}}$ **= kgr$\mathbf{CO_2}$/year** (Assuming η_{NG}=0.55)
  Assuming that NG power plants exceed the CO_2 emission limit at about 0% then the mass of CO_2, which must be accounted for penalty, will be:
  $\mathbf{m_{CO2,NG,pen}}$ **= kgr$\mathbf{CO_2}$/year** ∗ 0.0 = **kgr$\mathbf{CO_2}$/year**
  Finally, (5-6) ⇒ (5-7), (5-8) (*Table 5.7*) ⇒ $\mathbf{m_{CO2}}$ **= kgr$\mathbf{CO_2}$/y** (η_{DIE}=0.36, APPENDIX B.3)
  Assuming that diesel (heavy) power plants exceed the CO_2 emission limit at about 7% then the mass of CO_2, which must be accounted for penalty, will be:
  $\mathbf{m_{CO2,Di,pen}}$ **= kgr$\mathbf{CO_2}$/year** ∗ 0.07 = **kgr$\mathbf{CO_2}$/year**
  Thus, **$\mathbf{CO_2}$ emission cost -which varies with the $\mathbf{CO_2}$ penalty price-** paid by PPC is
  CO_2 emission cost paid by PPC = ($m_{CO2,Li,pen}$ + $m_{CO2,NG,pen}$ + $m_{CO2,DiH,pen}$) ∗ (€/1,000) = **€/year** (5-59)
  where factor 1,000 is for units similarity (paragraph 5.2.8)

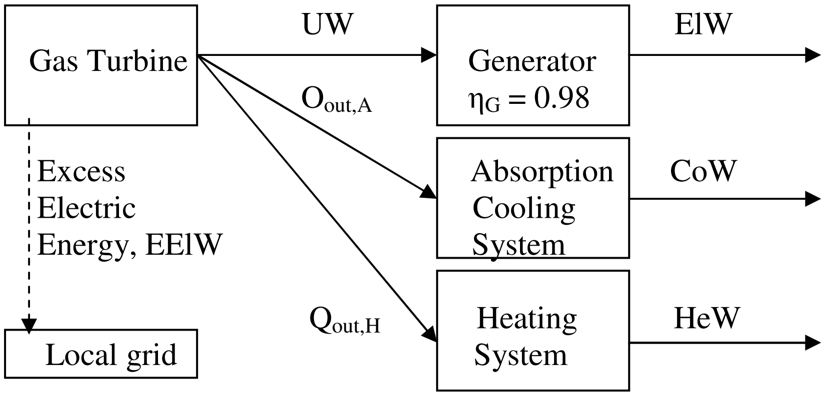
<u>1-Shaft GT, HE</u>

The only difference is concerning the GT package cost, (+30%).

<u>Scenario 2:</u> Maximum capacity GT, following the total demand load scenario

Similar to the previous scenario, but the system is always working to exactly cover all its needed power at any time. The distribution of the power demand is such (cooling power is relatively higher than heating and electric), that when the cooling power is covered, then there is an excess of electric energy to export to the local national grid

The block diagram of the technical configuration is showed in *Fig. 5.12*.



Notice:
Continue arrows display definite transfer, **Dot arrows** display possible transfer

Fig. 5.12: Technical block diagram of the scenario 2

In this mode the following points (key points) must be taken into consideration:

1. selection of the GT power (choosing the TET, Rc from diagrams *Fig. 3.2, 3.4, 3.6* and regulating the mass flow of the engine at the DP performance) in such way to cover the electric, heating and cooling power and energy demand at any month. The monthly operation is characterised by variant TET (less than that of the DP performance,), while the ambient conditions P_a T_a are vary according to the conditions referred in *Table 2.1* (This is actually part load performance of the GT)
   The selected engine has the best η_{th} for the maximum TET without cooling system.

2. 3. and 4. key points are same as scenario1

Having the above in mind the economic simulation procedure is as following:

<u>1-Shaft GT</u>, <u>2-Shaft GT</u>

- One GT package cost (generator, included) = **€ [same as in scenario 1,** (5-37)**]**
  GT package installation cost = **€ [scenario 1,** (5-38)]
  GT package maintenance cost = **€/year [scenario 1,** (5-39)**]**

- Heat exchanger cost (installation cost, included) = **€ [scenario 1,** (5-40)]
  Heat exchanger maintenance cost = **€/year [scenario 1,** (5-41)**]**

- District heating installation cost = **€ [scenario 1,** (5-42)**]**

- Absorption chiller cost (installation cost, included) = **€ [scenario 1,** (5-43)**]**
  Absorption chiller maintenance cost = **€/year [scenario 1,** (5-44)**]**

- Cost of back up cooling
  The electric compression refrigeration system capital cost -**which has fix value**- including the installation cost can be calculated with the help of equation (5-45) **[scenario 1,** (5-45)**]** where again the factor 0.5 is due to the assumption that the back up cooling power is the 50% of the maximum cooling demand power in MW, MW is the maximum value of cells of column 3, (*Table 5.10*) and €/MW is the corresponding price (paragraph 5.2.4)
  Operation cost = **€/year [scenario 1,** (5-46)**]**
  Maintenance cost = **€/year [scenario 1,** (5-47)**]**
  We assume that the maintenance of the cooling system is taking place in November (last week)

- NG mass flow per month = **[scenario 1,** (5-48)**]**
  NG mass flow per year = **[scenario 1,** (5-49)**]**
  Cost of NG per year = **[scenario 1,** (5-50)**]**

- Cost of back up boilers (assume 50% of the maximum heating demand power in MW) using medium heating oil. *Table 5.4*: ρ =0.92kgr/lt, FCV=41MJ/kgr>
  Capital cost = **€ [scenario 1,** (5-25)**]**
  Installation cost = **€ [scenario 1,** (5-26)**]**
  Maintenance cost = **€/year [scenario 1,** (5-51)**]**
  Cost of medium heating oil = **€/year [scenario 1,** (5-53)**]**

- Connection to the grid cost = **€ [scenario 1,** (5-54)**]**

- Electricity cost = **€/year [scenario 1,** (5-55)**]**

- Electricity profit = **€/year [[scenario 1,** (5-56)**]**

- Greek government is offering financial support = **€ [scenario 1,** (5-57)**]**

- CO_2 emissions estimation penalty = **€/year [scenario 1]**

- CO_2 emissions estimation profit = **€/year [scenario 1,** (5-58)**]**

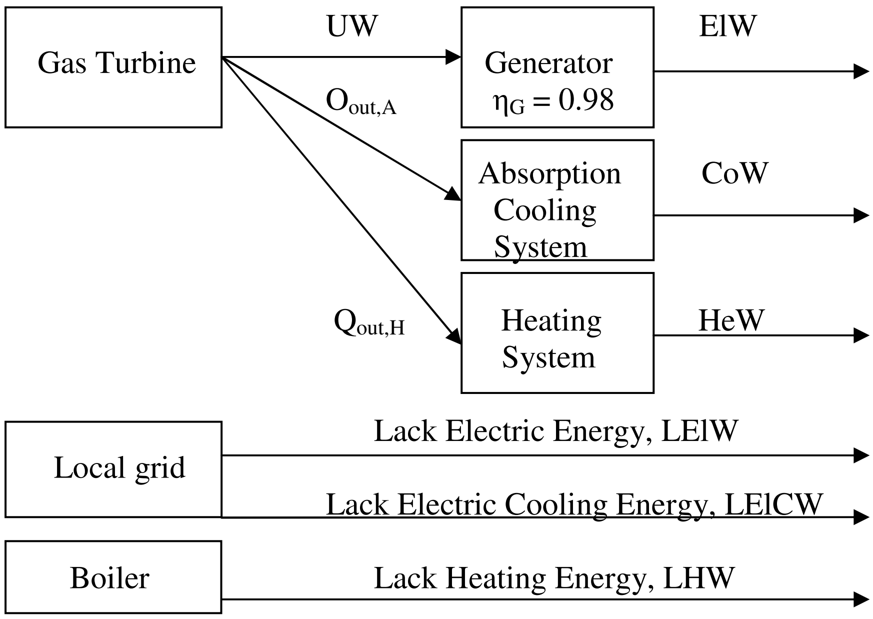
<u>1-Shaft GT HE</u>

The only difference is concerning the GT package cost, (+30%).

<u>Scenario 3:</u> Minimum electric capacity GT

In this scenario the GT at the design point has the power equal to that month which has the minimum electric power between the months of the year (minimum value of column 4, -July- *Table 5.10).* Thus, the needed surplus electric energy is supplied from the local national grid. This also means that if there is a lack of heating energy, which is necessary for the proper operation of an absorption chiller system, it will be covered by conventional air conditioners. Finally, the possible lack of heating power will be covered by the use of boilers.

The block diagram of the technical configuration is showed in *Fig. 5.13*.



<u>Notice:</u>
Continue arrows display definite transfer

Fig. 5.13: Technical block diagram of the scenario 3

In this mode the following points (key points) must be taken into consideration:

1. Selection of the GT power (choosing the TET, R_c from diagrams *Fig. 3.2, 3.4, 3.6* and regulating the mass flow of the engine at the DP performance) in such way to cover only the electric, power and energy demand of the lowest energy demand month. The monthly operation is characterised by constant TET (the same with the DP performance), while the ambient conditions P_a T_a are vary according to the conditions referred in *Table 2.1* (OD performance)
   The selected engine has the best η_{th} for the maximum TET without cooling system.

2. 3. and 4. key points are same as scenario 1

Having the above in mind the economic simulation procedure is as following:

<u>1-Shaft GT</u>, <u>2-Shaft GT</u>

- One GT package cost -**which has fix value**- including the necessary generator, the contribution devices, while the price of the gear box is assumed relatively negligible:

  (5-37) $\Rightarrow$ GT Package Cost = [MW] $*$ [\$/kW] $*$ 1000 / 1.23 = **€**

  where MW is minimum value of cells of column 4, *Table 5.10*, \$/kW is the corresponding price of column (see APPENDIX E.1), the factor 1000 is due to transformation from kW to MW and finally factor 1.23 is due to transformation from \$ to €.

  GT package installation cost = **€ [scenario 1,** (5-38)]

  GT package maintenance cost = **€/year [scenario 1,** (5-39)**]**

- The heat exchanger cost -**which has fix value**- including the installation cost, is given by the following equation:

  (5-40) $\Rightarrow$ Heat Exchanger Cost = [MW] $*$ [€/MW] = **€**

where MW is the maximum capability of GT heat power production and €/MW is explained in paragraph 5.2.3.

  Heat exchanger maintenance cost = **€/year [scenario 1,** (5-41)**]**

- The district heating installation cost -**which has fix value**- is given by the following equation:

  (5-42) $\Rightarrow$ District Heating Installation Cost = [MW] $*$ [€/MW] $*$ ([MW]/120) = **€**

  where MW is the capability of GT heat power production, €/MW is the corresponding price (paragraph 5.2.5) and the factor 120 is due to the relatively small system (paragraph 5.2.5).

- The absorption chiller cost -**which has fix value**- including the installation cost is given by the following equation:

  (5-43) $\Rightarrow$ Absorption Chiller Cost = [MW] $*$ [€/MW$_c$] = **€**

  where MW is the maximum capability of GT heat power production and €/MW$_c$ is the corresponding price (paragraph 5.2.4).

  The absorption chiller maintenance cost -**which has fix value, for every year-** is given by the following equation:

  (5-44)$\Rightarrow$Absorption Chiller Maint. Cost =0.8$*$Absorpt. chiller cost $*$ 0.031= **€/year**

  where factors 0.8 and 0.031 are explained in paragraphs 5.2.4 and 5.2.7 respectively.

- Electric compression refrigeration system.

  The capital cost plus the installation cost -**which has fix value-** is

  Capital Cost + Installation Cost **=** ([MW]$_a$ -([MW]$_b$ $*$ [1/(z+1)])) $*$ €/MW **= €** (5-60)

  where MW$_a$ is cells of column 3 (*Table 5.10)*, MW$_b$ is the capability of GT heat power production and €/MW is the price of electricity per MW (see paragraph 5.2.4)

  The electric energy per month supplied from the local grid for cooling ($W_{e,\ c}$), can be calculated as follows:

COP = 4.5 ⇒ (Eq. 4-3) ⇒ $W_{e,c}$=

$$\{([MWh_a]*30-([MW_a]*[1/(z+1)]*\eta_{HE}*24*30))-([MW_b]*24*30-[MWh_b]*30)\}/4.5=MWh \quad (5\text{-}61)$$

where MWh_a are the corresponding values of cells of column 3, *Table 5.11,* MW_a is the capability of GT's heat power production, [1/(z+1)] is the portion of GT exhaust heat, going for cooling, factor 24 is the number of hours of one day, factor 30 is the number of days of the month -except November which consider to operate 23 days-, MW_b is the UW_{OD} of each month and MWh_b are the cells of column 4 (*Table 5.11*).

Then the operation cost per month is given by

$$(5\text{-}22) \Rightarrow \text{Operation Cost per month} = W_{e,c} * €/MWh * 30 = €/month$$

where €/MWh is the price of electricity per MWh (see paragraph 5.2.9) and 30 is assumed the number of days of the month.
The operation cost of the year can easily calculated as

$$(5\text{-}23) \Rightarrow \text{Operation cost} = \sum_{1}^{12} (€/month) = \mathbf{€/year}$$

which varies accordingly to the inflation rate.

Finally, the maintenance cost -**which has fix value, for every year-** is given accordingly to paragraph 5.2.7:
(5-24)⇒Maintenance Cost = (Capital cost + Installation cost) $*$ 0.90 $*$ 0.03 **= €/year**
We assume that the maintenance of the cooling system is taking place in January (last week).

- NG mass flow per month = **[scenario 1,** (5-48)**]**
  NG mass flow per year = **[scenario 1,** (5-49)**]**
  Cost of NG per year = **[scenario 1,** (5-50)**]**

- The heating power of the GT is proved to produce not enough heat to cover the heating demand in every month. The capital cost of boilers -**which has fix value**- is given by the following equation:
  Capital Cost of boilers = Maximum$\{MW_a-(MW_b*[z/(z+1)]*\eta_{HE})\}$*€/MW=**€** (5-62)
  where MW_a are the cells of column 2, *Table 5.10*, MW_b is the capability of GT's heat power production, [z/(z+1)] is the portion of GT exhaust heat, going for heating and €/MW is the corresponding price (paragraph 5.2.3).
  Boilers using medium heating oil from *Table 5.4*: ρ =0.92kgr/lt, FCV=41MJ/kgr

  The energy provided from the medium heating oil:
  $\eta_{th,b}$=0.8⇒<Eq. (4-3)>⇒$Q_{f,b}$=(MWh $*$ 30 - MW $*$ 24 $*$ 30) **/** 0.8 = MWh per month
  where $\eta_{th,b}$ is given in paragraph 5.2.3, MWh are the corresponding values of cells of column 2, *Table 5.11* Mwis the GT's production capability of heat energy and factor 30 is the number of days per month.
  The mass of medium heating oil:

$$(5\text{-}53) \Rightarrow m_{fm,b} = [(Q_{f,b} \mathbf{X}\ 3{,}600) / FCV = kgr$$

  and finally the cost of medium heating oil **-which varies accordingly to the international oil prices-** is calculated accordingly to eq. (5-30).

The Maintenance cost -**which has fix, for every year-** is given by following equation

$$\text{Maintenance Cost} = \text{capital cost} * 0.02 * (7/360) = \text{€/year} \qquad (5\text{-}63)$$

where factor 0.02 is explained in paragraph 5.2.7 and factor (7/360) is due to the fact that operates only 7 days per year. We assume that the maintenance of the heating system is taking place in June (last week)

- Cost of electricity **= 0 €/year (varies accordingly to the inflation rate)**
  Due to the fact that the GT electricity production every month is turn out to be higher than the demand (column 4, *Table 5.10*). This little excess of electricity power is covering part of the electricity needed to operate the conventional Electric compression refrigeration system

- Greek government is offering financial support -**which has fix value**- is given by the following equation:
  Greek Government Financial Support = 0.4 * (GT package cost + GT package installation cost + Heat exchanger cost + District heating installation cost + Absorption chiller cost + Capital cost of boilers + Installation cost of boiler) = **€** (5-64)
  where factor 0.4 is explained in paragraph 5.2.10

- CO_2 emissions estimation and penalty
  The energy per year supplied from national grid (PPC) is calculated as follow:

$$\text{Energy per y}=\sum_{1}^{12} \{[MWh_a*30-MW_a*24*30]-[MW_b*24*30-MWh_b]\}=MWh/y \quad (5\text{-}65)$$

where MWh_a the cells of column 3 (*Table 5.11),* MW_a is the GT production capability of cooling energy per day, MW_b is the electric power produced every month from GT, MWh_b the cells of column 4 (*Table 5.11)* and 30 is assumed the number of days of the month November 23-.

The airport is connected with the interconnected national grid (continental) of Greece. Taking into account the data from *Table A.5*, it can be estimated how many MWh/year are produced from the available kinds of power plants, assuming that analogical distribution of the total electric energy per year:
Lignite: MWh/year * 0.674 = MWh/year
Natural gas: MWh/year * 0.168 = MWh/year
Diesel (heavy heating oil): MWh/year * 0.056 = MWh/year

Using the CO_2 calculation method presented in paragraph 5.2.8, the m_{CO2} produced from every type of power plant respectively, can be estimated:
(5-6) $\Rightarrow$ (5-7), (5-8) (*Table 5.7*) $\Rightarrow$ **m_{CO2} = kgrCO_2/year** (Assuming η_{LIG}=0.3)
Assuming that lignite power plants exceed the CO_2 emission limit at about 10% then the mass of CO_2, which must be accounted for penalty, will be:
$m_{CO2,Li,pen}$ = kgrCO_2/year
Similarly (5-6) $\Rightarrow$ (5-7), (5-8) (*Table 5.7*) $\Rightarrow$ **m_{CO2} = kgrCO_2/year** (Assuming η_{NG}=0.55)
Assuming that NG power plants exceed the CO_2 emission limit at about 0% then the mass of CO_2, which must be accounted for penalty, will be:
$m_{CO2,NG,pen}$ = kgrCO_2/year

Finally, (5-6) $\Rightarrow$ (5-7), (5-8) (*Table 5.7*) $\Rightarrow$ **m_{CO2} = kgrCO_2/y** (η_{DIE}=0.36, APPENDIX B.3)
Assuming that diesel (heavy) power plants exceed the CO_2 emission limit at about 7% then the mass of CO_2, which must be accounted for penalty, will be:
$m_{CO2,Di,pen}$ = kgrCO_2/year
Thus, **CO_2 emission cost** paid by PPC = ($m_{CO2,Li,pen}$ + $m_{CO2,NG,pen}$ + $m_{CO2,DiH,pen}$) **X** €/1,000kgr <paragraph 5.2.8> = **€/year (varies with the CO_2 penalty price)**

The airport is using boilers burning only medium heating oil. Thus the electric energy per year produced burning medium heating oil will be:
The heating energy per year provided by the boilers can be determined by the following equation

Heating energy per year <boilers>= $\sum_{1}^{12}\{MWh_a * 30-(MW_a*[z/(z+1)]*24*30)\}/0.8=$

=MWh/year
where MWh_a are cells of column 2, *Table 5*.11,MW_a is the GT production capability of heat energy, factor z/(z+1)] is the portion of GT exhaust heat, going for heating, and 30 is is assumed the number of days of the month November 23-.

Using the CO_2 calculation method presented in paragraph 5.2.8, the m_{CO2} produced by boilers burning medium heating oil will be:
(5-6) $\Rightarrow$ (5-7), (5-8) (*Table 5.7*)$\Rightarrow$ **m_{CO2} = kgrCO_2/year** (η_{DIE}=0.8, APPENDIX B.3) Assuming that diesel power plants exceed the CO_2 emission limit at about 7% then the mass of CO_2, which must be accounted for penalty, will be: **$m_{CO2,Di,b,pen}$ = kgrCO_2/year.** Thus, **CO_2 emission cost** paid by airport company = $m_{CO2,Di,b,pen}$ *€/1,000kgr<paragraph 5.2.8> = **€/year (varies with the CO_2 penalty price)**

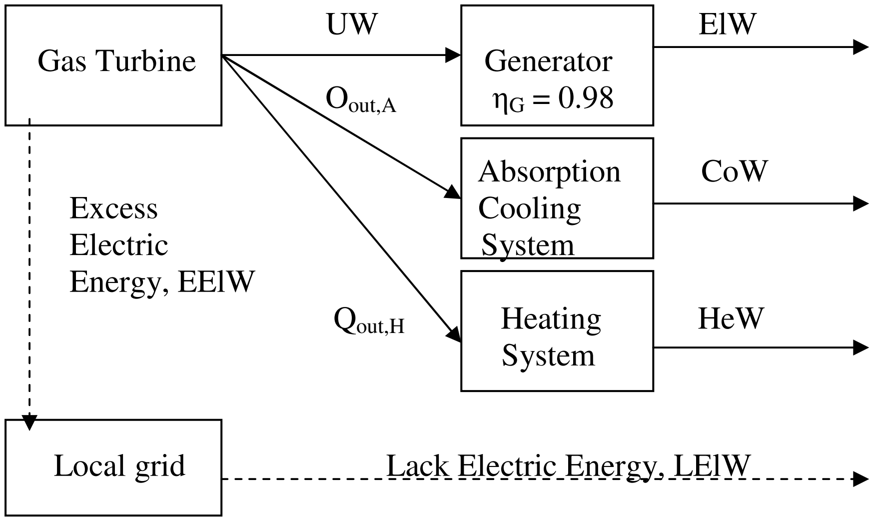
Total CO_2 emission cost = CO_2 emission cost paid by PPC + CO_2 emission cost paid by airport company

1-Shaft GT HE

The only difference is concerning the GT package cost, (+30%).

***Scenario 4:* Covering the thermal and cooling demand GT,**
The useful thermal and cooling output of the GT, is equal to the demand of thermal and cooling load, at any instant of time. If the generated electricity is higher than the load, surplus electricity is sold to the grid; if it is lower, supplementary electricity is purchased from the local national grid.

The block diagram of the technical configuration is showed in *Fig. 5.14*.



Notice:
Continue arrows display definite transfer
Dot arrows display possible transfer

Fig. 5.14: Technical block diagram of the scenario 4

In this mode the following points (key points) must be taken into consideration:

1. selection of the GT power (choosing the TET, R_c from diagrams *Fig. 3.2, 3.4, 3.6* and regulating the mass flow of the engine at the DP performance) in such way to cover the heating and cooling power and energy demand of the most demand month. The monthly operation is characterised by constant TET (the same with the DP), while the ambient conditions P_a T_a are vary according to the conditions referred in *Table 2.1* (OD performance)

2., 3. and key points are same as scenario1

Having the above in mind and observing the numbers of the *Tables 5.10* and *5.11*, it can be seen that in the case of the Airport the scenario 4 is actually the same with scenario 1. This is because that the light and motion electric power of the airport is very low relatively to the cooling or heating power, which fortunately set the power of the GT.

5.5 Island energy scenarios
5.5.1 Conventional case
In this paragraph, the different costs of energy will be analytically presented, for the conventional namely the present energy situation of the Rhodes Island. This can be done with the essential assistance of the data presented in CHAPTER 2. *Tables 2.10* and *2.11* are presenting the power demand and the energy consumption respectively, for a typical day each month of the year. As it has been said in paragraph 2.1, these values do not include either the hypothetical future increase, or the estimation of the worst case for the energy demand point of view. After a relevant discussion with the supervisor, the author decided to multiply by a factor of 1.2 all the prices or demand of the above mentioned Tables, in order to include the worst case situation. The results are shown in *Tables 5.12* and *5.13*

Table 5.12: Rhodes power demand in MW

MONTHS (30 days per month)	**COOLING** (MW_C)	**LIGHTING & OTHER** (MW_e)	**HEATING** MW_{th}		**TOTAL** MW
			ELECTRIC	BOILERS	
JAN	0	53.604	19.824	19.824	93.252
FEB	1.560	64.692	11.688	11.700	89.640
MAR	5.856	60.060	7.320	7.332	80.556
APR	19.548	46.068	4.188	4.188	73.992
MAY	26.976	52.272	5.052	5.064	89.364
JUN	37.824	65.148	2.100	2.100	107.172
JUL	52.128	75.588	2.604	2.604	132.960
AUG	60.084	75.456	4.140	4.248	143.880
SEP	40.092	63.924	4.332	4.332	112.680
OCT	25.164	56.400	5.208	5.208	91.980
NOV	1.752	50.388	6.456	6.432	65.040
DEC	0	54.600	21.240	21.228	97.068

Table 5.13: Rhodes energy demand MWh

MONTHS (30 days per month)	**COOLING** (MWh_C)	**LIGHTING & OTHER** (MWh_e)	**HEATING** MWh_{th}		**TOTAL** MWh
			ELECTRIC	BOILERS	
JAN	0.0	1,286.4	475.2	476.4	2,238.0
FEB	37.2	1,552.8	280.8	280.8	2,151.6
MAR	140.4	1,441.2	176.4	175.2	1,933.2
APR	469.2	1,105.2	100.8	100.8	1,776.0
MAY	648.0	1,254.0	121.2	121.2	2,144.4
JUN	908.4	1,563.6	50.4	50.4	2,572.8
JUL	1,251.6	1,814.4	61.2	63.6	3,189.6
AUG	1,442.4	1,810.8	99.6	102.0	3,453.6
SEP	962.4	1,534.8	103.2	104.4	2,704.8
OCT	603.6	1,353.6	124.8	124.8	2,208.0
NOV	42.0	1,209.6	154.8	154.8	1,561.2
DEC	0.0	1,310.4	510.0	508.8	2,329.2

1 Cost of electricity (lighting, motion, etc)

As it has been said in CHAPTER 2, Rhodes is an island and its electrification, which is based on autonomous petrol stations. The PPC has estimated that the selling price of the electricity is about the double comparing to the electricity price for the interconnected system of the country. For political and national reasons the PPC is forced to balance the prices and finally keep the same price for all the Greek consumers wherever they live inside the country. In this study the real price will be used namely the double price. Using the date of *Table B.13* in APPENDIX B.3 and the prices presented in paragraph 5.2.9, *Table 5.14* is made.

Table 5.14: Electricity prices and percentage of different types of electrical consumption

Type of Use	Final selling prices (€/MWh), 2004	Percentages % of electric energy
Domestic	2 X 67.1 = 134.2	38.9
Commercial	2 X 55.0 = 110	43.6
Industrial	2 X 43.4 = 86.8	4.4
Remaining	2 X 50.0 = 100	13.1

Taking into account the data from *Table 5.14* a formula can be created, calculating the electricity price for a day/month.
El. Cost per month= $((A * 0.389) * 134.2 + (A * 0.436) * 110 + (A * 0.044) * 86.8 + (A * 0.131) * 100) * 30 = 117.08 * A * 30$ = €/month (5-66)
where A is a value from cell of the column 3 of *Table 5.13* and 117.08€/MWh is the equivalent electricity price for the entire Island.
The electricity cost of the year (12 months) is

$$(5\text{-}19) \Rightarrow \text{Cost of Electricity} = \sum_{1}^{12} (\text{El. Cost per month}) = \textbf{€/year}$$

which varies accordingly to the inflation rate.

2 Cooling
Electric compression refrigeration system.
The capital cost plus the installation cost **-which has fix value-** is

$$(5\text{-}20) \Rightarrow \text{Capital Cost + Installation Cost} \mathbf{=} \text{MW} * \text{€/MW} \mathbf{= €}$$

where MW is the maximum value of the cells of column 2, *Table 5.12*, and €/MW is the price of electricity per MW (see paragraph 5.2.4)
Notice: The maximum cooling power is cumulated, because it includes installations of various types (domestic, commercial, industrial, etc.). Therefore, the capital cost plus the installation cost, is considered to be 70,000€/MW.

Operation cost
The electric energy per month supplied from the local grid for cooling ($W_{e,\ c}$), can be calculated as follows:

$$(5\text{-}21) \Rightarrow \text{COP} = 4 \Rightarrow (\text{Eq. 4-3}) \Rightarrow W_{e,\ c} = \text{MWh} \mathbf{/} 4 = \text{MWh}$$

where MWh are the corresponding values of cells of column 2, *Table 5.13*.
Then the operation cost per month is given by the following equation:

$$\text{Operation Cost per month} = W_{e,\ c} * 117{,}08 * 30 = \text{€/month} \qquad (5\text{-}67)$$

where 117.08€/MWh is the equivalent electricity price for the entire Island and 30 is assumed the number of days of the month.
The operation cost of the year can easily calculated as

$$\text{Operation cost} = \sum_{1}^{12} (\text{€/month}) \mathbf{=} \textbf{€/year} \qquad (5\text{-}68)$$

which varies accordingly to the inflation rate

Finally, the maintenance cost **-which has fix value, for every year-** is given accordingly to paragraph 5.2.7:

(5-24) $\Rightarrow$ Maintenance Cost = (Capital cost + Installation cost) $*$ 0.90 $*$ 0.03 **= €/year**

We assume that the maintenance of the cooling system is taking place in January (last week)

3 <u>Heating</u>

Heating is coming from two sources: a) from individual boilers using light diesel as fuel and b) from electric inverters, heaters etc, which consume electricity.

a) Individual boilers using light diesel as fuel (from *Table 5.4*: ρ = 0.86kgr/lt, FCV=42.5MJ/kgr).

The capital cost **-which has fix value-** is given by

(5-25) $\Rightarrow$ Capital Cost = MW $*$ **€/MW = €**

where MW is the maximum value among the cells of column 5, *Table 5.12* and €/MW is the price of boiler per MW (see paragraph 5.2.3)

The installation cost **-which has fix value-** is given by

(5-26) $\Rightarrow$ Installation cost = Capital Cost $*$ 0.10 **= €**

where factor 0.10 is explained in paragraph 5.2.3

The energy provided from the light diesel per month is calculated as follow:

(5-27) $\Rightarrow \eta_{th,\,b} = 0.8 \Rightarrow$ <Eq. (4-3)> $\Rightarrow Q_{f,\,b}$ = (MWh $*$ 30) / 0.8 = MWh per month

where $\eta_{th,\,b}$ is given in paragaph 5.2.3, MWh are the corresponding values of cells of column 5, *Table 5.13* and 30 is assumed the number of days of the month.

The mass of light diesel per month and per year are calculated as follow:

(5-28) $\Rightarrow$ $m_{fm,b} = [(Q_{f,\,b} * 3{,}600) / FCV]$ = kgr/month

(5-29) $\Rightarrow$ $m_{fy,b} = \sum_{1}^{12} m_{fm,b}$ = kgr/year

where FCV is taken from *Table 5.4*.

And so the cost of light diesel per year **-which varies accordingly to the international oil prices** can be estimated:

Cost of light diesel = (($\dot{m}_{fy,b}$ / 0.86) $*$ (2.9t+42.1) / 159) $*$ 1.9 = **€/year** (5-69)

Where 0.86 is the density ρ of the light diesel (paragraph 5.), (2.9t+42.1) is equation (5-1), while the factor 159 is the capacity of barrel (paragraph 5.2.7) and 1.9 is due to the isolated consumption area namely Island paragraph 5.2.7.

Finally, the maintenance cost **-which has fix value, for every year-** is given accordingly to paragraph 5.2.7:

(5-31) $\Rightarrow$ Maintenance Cost = Capital cost $*$ 0.02 **= €/year**

We assume that the maintenance of the heating system is taking place when there is no need for heating e.g. summer time.

b) Electric inverters, heat pumps, heaters etc

Capital cost and installation cost and maintenance cost is negligible due to the fact that the majority of them are used for cooling and heating purposes, so the capital cost of them is already calculated in the previous cooling section.

Operation cost

$$COP = 2.5 \Rightarrow (4\text{-}3) \Rightarrow W_{e,\,h} = [MWh] / 2.5 = MWh \qquad (5\text{-}70)$$

where 2.5 is the assumed average COP of heat pumps heaters, etc, (paragraph 4.6.1) and MWh are the cells of column 4 (*Table 5.13*).

$$\text{Operation cost per month} = W_{e,\,c} * 117.08 * 30 = €/\text{month} \qquad (5\text{-}71)$$

$$\text{Operation cost} = \sum_{1}^{12} (€/\text{month}) = \mathbf{€/year} \qquad (5\text{-}72)$$

(varies accordingly to the inflation rate)

4. <u>CO_2 emissions estimation and penalty</u>

The energy per year supplied from national grid (PPC) is calculated as follow:

$$(5\text{-}33) \Rightarrow \text{Energy per year} = \sum_{1}^{12} \text{ values per typical day} * 30 = \text{MWh/year}$$

where the values per typical day are the cells of columns 2 plus 3 plus 4 of *Table 5.13* and 30 is assumed the number of days of the month.

The island is part of the non-interconnected national grid of Greece. Taking into account the data from *Table A.5*, it can be estimated how many MWh/year are produced from the available kind of power plants, (obviously, renewable excluded) assuming that the total electric energy per year is supplied only diesel power plants:

Diesel (medium heating oil): MWh/year $*$ 0.985 = MWh/year

Using the CO_2 calculation method presented in paragraph 5.2.8, the m_{CO2} produced from every type of power plant respectively, can be estimated:

(5-6) $\Rightarrow$ (5-7), (5-8) (*Table 5.7*) $\Rightarrow$ $\mathbf{m_{CO2}}$ **= kgr$\mathbf{CO_2}$/y** (η_{DIE}=0.36, APPENDIX B.3)

Assuming that medium diesel power plants exceed the CO_2 emission limit at about 6% then the mass of CO_2, which must be accounted for penalty, will be:

$\mathbf{m_{CO2,DiM,pen}}$ **= kgr$\mathbf{CO_2}$/year** $*$ 0.06 = **kgr$\mathbf{CO_2}$/year**

Thus, **$\mathbf{CO_2}$ emission cost** paid by PPC = $m_{CO2,Di,pen}$ $*$ €/1,000kgr<paragraph 5.2.8> = **€/year (varies with the $\mathbf{CO_2}$ penalty price)**

The island is using boilers burning only light heating oil. Thus, the electric energy per year produced burning light heating oil will be:

$$\text{Heating energy per year of boilers} = \sum_{1}^{12} \text{ MWh} * 30 = \text{MWh/year} \qquad (5\text{-}73)$$

where the values per typical day are the cells of column 5, *Table 5.13* and 30 is assumed the number of days of the month.

Using the CO_2 calculation method presented in paragraph 5.2.8, the m_{CO2} produced by boilers burning light heating oil will be

(5-6) $\Rightarrow$ (5-7), (5-8) (*Table 5.7*)$\Rightarrow$ $\mathbf{m_{CO2}}$ **= kgr$\mathbf{CO_2}$/y** (η_{DIE}=0.8, APPENDIX B.3)

Assuming that diesel power plants exceed the CO_2 emission limit at about 3% then the mass of CO_2, which must be accounted for penalty, will be:

$\mathbf{m_{CO2,Di,L,pen}}$ **= kgr$\mathbf{CO_2}$/year** $*$ 0.03 = **kgr$\mathbf{CO_2}$/year**

Thus, **$\mathbf{CO_2}$ emission cost** paid by island consumers=$m_{CO2,Di,b,pen}$$*$(€/1,000)=**€/year (varies with the $\mathbf{CO_2}$ penalty price)** where 1,000 is for units similarity (paragraph 5.2.8)

Total $\mathbf{CO_2}$ emission cost = CO_2 emission cost paid by PPC **+** CO_2 emission cost paid by island consumers = **€/year**

5.5.2 Hypothetical operation scenarios

The scenarios-modes of operation are proportional to those of the airport. Obviously, they adjusted to the special needs of the Rhodes Island:

***Scenario 1**: Maximum capacity GT, operation scenario*

There is coverage of the electric (light & others plus 60% of electric heating), 60% of thermal and 60% of cooling loads at any instant of time. The possible excess in electric power cannot supply the national local grid due to the autonomous grid of the island.

In this thesis the author aims to investigate the general case where the island is not near continental land or another big island. On the other hand the possible submarine connection with other islands is practically prohibitive due to:

- Excessively increase of the "connection to the grid cost"
- Relatively low excess of electric power to secure the financial viable of the investment.
- Fluctuant excess of electric power, which means that there must be and another autonomous plant on the possible neighbour island to secure the electric power supply.

For the above reasons this scenario is actually not economically feasible.

***Scenario 2:** Maximum capacity GT, following the total demand load scenario*

It is the same scenario with the previous, but the system is always working only to cover all its needs at any time. There is no excess of electric energy to export to the national grid. The block diagram of the technical configuration is showed in *Fig. 5.12*.

In this mode the following points (key points) must be taken into consideration:

1. selection of the GTs power (choosing the TET, Rc from diagrams *Fig. 3.2, 3.4, 3.6* and regulating the mass flow of the engine at the DP performance) in such way to cover the electric (100% of light & others plus 40% of electric heating plus 40% cooling), 60% of thermal and 60% of cooling loads at any instant of time. The monthly operation is characterised by variant TET (less than that of the DP performance,), while the ambient conditions P_a T_a are vary according to the conditions referred in *Table 2.8* (This is actually part load performance of the GT).
2. The number of the engines is two for better reliability and availability, factors extremely crucial for the local grid of an isolated island with autonomous power station. The selected engines are same and they have the best η_{th} for the maximum TET without cooling system
3. The proportional factor z, is representing the way of the power or the energy of the GT exhaust gasses is split between the heating and cooling demand every month.
4. The availability of the engines is assumed to be about 96%, in other words each engine is assumed to shut down for two weeks (in November, when total needs are minimum) for the annual service. During those weeks electricity is supplied from the conventional diesel plant and the second GT, while heating is supplied from stand by boilers.
5. Salaries for extra personnel are assumed to be negligible.

Having the above in mind the economic simulation procedure for the 1-Shaft GT, is as following:

1-Shaft GT, 2-Shaft GT

- Two GT package cost -**which has fix value**- including the necessary generator, the contribution devices, while the price of the gear box is assumed relatively negligible:

$$\text{GT Package Cost} = 2 * [\text{MW}/2] * [\$/\text{kW}] * 1000 / 1.23 = \textbf{€} \quad (5\text{-}74)$$

where MW is the useful work of the GT at the design point, \$/kW is the corresponding price of column (see APPENDIX E.1), the factor 1000 is due to transformation from kW to MW and finally factor 1.23 is due to transformation from \$ to €.

GT package installation cost = **€ [scenario 1, paragraph 5.4.2,** (5-38)**]**
GT package maintenance cost = **€/year [scenario 1, paragraph 5.4.2,** (5-39)**]**

- The heat exchanger cost -**which has fix value**- including the installation cost, is given by the following equation:

$$\text{Heat Exchanger Cost} = 0.6 * [\text{MW}_a + \text{MW}_b + \text{MW}_c] * [€/\text{MW}] = \textbf{€} \quad (5\text{-}75)$$

where MW_a, MW_b and MW_c are cells of columns 2, 4 and 5 (*Table 5.12*) respectively and €/MW is explained in paragraph 5.2.3.
Heat exchanger maintenance cost = **€/year [scenario 1, paragraph 5.4.2,** (5-41)**]**

- The district heating installation cost -**which has fix value**- is given by the following equation:

$$\text{District heating installation cost} = 0.6 * \text{MW}_a * [€/\text{MW}] * (\text{MW}_b) / 120 = \textbf{€} \quad (5\text{-}76)$$

where factor 0.6 is scenario constrain, MW_a are cells of column 2 plus cells of column 4 plus cells of column 5, (*Table 5.12*), €/MW is the corresponding price (paragraph 5.2.5), MW_b is the maximum sum of cells of columns 2+4+5, (*Table 5.12)* and the factor 120 is due to the relatively small system (paragraph 5.2.5).

- The four absorption chillers cost -**which has fix value**- including the installation cost is given by the following equation:

$$\text{Absorption Chiller Cost} = 4 * ([\text{MW}]/4) * [€/\text{MW}_c] = \textbf{€} \quad (5\text{-}77)$$

where MW is the maximum cell of column 2 (*Table 5.12*) and €/MW_c is the corresponding price (paragraph 5.2.4).
Absorption Chiller Maintenance Cost = **[scenario 1, paragraph 5.4.2,** (5-44)**]**.

- Electric compression refrigeration system
The capital cost plus the installation cost -**which has fix value-** is

$$(5\text{-}20) \Rightarrow \text{Capital cost} + \text{Installation cost} = \text{MW}* €/\text{MW} = \textbf{€}$$

where MW is the maximum cell of column 2, *Table 5.12.* The maximum cooling power is cumulated, because it includes installation of various types (domestic, commercial, industrial, etc.). Therefore, the capital cost plus the installation cost, is considered to be 70,000€/MW.

Operation cost
The electric energy per month supplied from the conventional plant for cooling ($W_{e,c}$), can be calculated as follows:

$$(5\text{-}21) \Rightarrow \text{COP}=4.5 \Rightarrow (\text{Eq.}4\text{-}3) \Rightarrow W_{e,c}=[0.4*\text{MWh}_a*30+0.6*\text{MWh}_b*15]/4.5=\text{MWh}$$

where Mwh_a are the corresponding values of cells of column 2, (*Table 5.16),* MWh_b is the value of November-cell of column 2, (*Table 5.16),*30 is the number of days of the month and 15 is the number of days of November.

Then the operation cost per month is given by the following equation:

(5-67) $\Rightarrow$ Operation Cost per month = $W_{e,\,c} * 117{,}08 * 30$ = €/month

where 117.08€/MWh is the equivalent electricity price for the entire Island and 30 is assumed the number of days of the month.

The operation cost of the year can easily calculated as

$$(5\text{-}68) \Rightarrow \text{Operation cost} = \sum_{1}^{12} (\text{€/month}) = \textbf{€/year}$$

which varies accordingly to the inflation rate.

Finally, the maintenance cost -**which has fix value, for every year-** is given accordingly to paragraph 5.2.7:

(5-24) $\Rightarrow$ Maintenance Cost = (Capital cost + Installation cost) $*$ 0.90 $*$ 0.03**= €/year**

We assume that the maintenance of the cooling system is taking place in January (last week)

- The NG mass flow per month is given by

  (5-48) $\Rightarrow$ NG mass flow per month = $\dot{m}_f * 60 * 60 * 24 * 30 = \dot{m}_{fm}$ kgr/month

  where first factor 60 is for the conversion of seconds to minutes, second factor 60 is for conversion of minutes to hours, factor 24 is for the conversion of hours to days, while factor 30 corresponds to the number of days of the month, except from November which is assumed to operate 15 days, due to shut down for annual service

  Thus, the NG mass flow per year is

  $$(5\text{-}49) \Rightarrow \text{NG mass flow per year} = \dot{m}_{fy} = \sum_{1}^{12} \dot{m}_{fm} \text{ kgr/year}$$

  The cost of NG per year -**which varies accordingly to the international oil prices-** is given by the equation:

  Cost of NG per year = $\dot{m}_{fy} * 1.11 * 48.9 * 0.0002778 * (2.39t+14.61)$ = **€/year** (5-78)

  where factors 1.11 and 48.9MJ/kgr is accordingly to paragraph 5.2.7, while the factor 0.0002778 is due to transformation from MJ to MWh. Finally the factor (2.39t+14.61) is equation (5-5).

- The heating power of the GT is proved to produce enough heat to cover the 60% of the boilers heating demand. So, the power of boilers will be the needed 40% of the boilers heating power, which working regularly and the 60% of the boilers heating power, which are for back up reasons working regularly only for 15 days in November

  The capital cost -**which has fix value-** is given by

  (5-25) $\Rightarrow$ Capital Cost = MW $*$ **€/MW = €**

  where MW is the maximum value among the cells of column 5, *Table 5.12* and €/MW is the price of boiler per MW (see paragraph 5.2.3)

  (boilers using light heating oil from *Table 5.4*: ρ =0.92kgr/lt, FCV=41MJ/kgr)

  The installation cost -**which has fix value-** is given by

  (5-26) $\Rightarrow$ Installation cost = Capital Cost $*$ 0.10 **= €**

  where factor 0.10 is explained in paragraph 5.2.3

  Finally, the maintenance cost -**which has fix value, for every year-** is given accordingly to paragraph 5.2.7:

  (5-31) $\Rightarrow$ Maintenance Cost = Capital cost $*$ 0.02 **= €/year**

We assume that the maintenance of the heating system is taking place when there is no need for heating e.g. summer time (last week of June)

Energy provided from the light heating oil is calculated as follow:

$\eta_{th,\,b} = 0.8 \Rightarrow$ <Eq. (4-3)> $\Rightarrow Q_{f,\,b} = (0.4 * MWh_a * 30) * 30 + (0.6 * MWh_b * 15) / \eta_{HE}$ = MWh/m per month.

where $\eta_{th,\,b}$ is given in paragraph 5.2.3, MWh_a are the corresponding values of cells of column 5 -except November- (*Table 5.13)*, MWh_b is the corresponding values of November-cell of column 5 (*Table 5.13)* and 30 is assumed the number of days of the month (15 for November).

The mass of light diesel per month and per year are calculated as follow:

$$(5\text{-}28) \Rightarrow m_{fm,b} = [(Q_{f,\,b} * 3{,}600) / FCV\,] = \text{kgr/month}$$

$$(5\text{-}29) \Rightarrow m_{fy,b} = \sum_{1}^{12} m_{fm,b} = \text{kgr/year}$$

where FCV is taken from *Table 5.4*.

And so the cost of light diesel per year **-which varies accordingly to the international oil prices,** can be estimated:

$$\text{Cost of light diesel} = ((\dot{m}_{fy,b} / 0.86\,) * (2.9t+42.1) \;/\; 159) * 1.9 = \textbf{€/year} \qquad (5\text{-}79)$$

where 0.86 is the density ρ of the light diesel (paragraph 5.), (2.9t+42.1) is equation (5-1), while the factor 159 is the capacity of barrel (paragraph 5.2.7) and 1.9 is due to the isolated consumption area namely Island paragraph 5.2.7.

The power of the GTs is proved to produce not enough electricity to cover the lighting & others demand. So, the electricity power for the electric heating power will come from the conventional plant.

$$\text{Operation cost of electr. heating energy} = \sum_{1}^{12} [(0.4 * MWh_a) * 30 + (0.6 * MWh_b)] * \text{€/MWh}$$

where MWh_a are the cells of column 4 -except November-, (*Table 5.13*) MWh_b is the ell of column 4, November, (*Table 5.13)*, €/MWh is the corresponding price (paragraph 5.2.9) and 30 is assumed the number of days of the month (15 for November).

- Connection to the grid cost = 0 **€ (fix)**

- Electricity cost = 0 **€/year (varies accordingly to the inflation rate)** due to the scenario constrain>

- Greek government is offering financial support = **€ [scenario 1, paragraph 5.4.2** (5-57)]

- CO_2 emissions estimation penalty.
  CO_2 emission cost paid by PPC = 0€ because we assume that trigeneration plants do not exceeds the official limits of CO_2 emissions.

The island is using boilers burning only light heating oil. Thus, the electric energy per year produced burning light heating oil will be:

$$\text{Heating energy per year of boil.} = \sum_{1}^{12}([\text{MWh}] * 30 - (z/z+1) * [\text{MW}] * 24 * 30) = \text{MWh/y}$$

where MWh are cells of column 5, (*Table 5.13*), (z/z+1) is the portion of GTs exhaust heat, going for heating, MW is the heat power of the GTs per month 30 is the number of days of the month, except November, which is assumed 15, due to shut down period

Using the CO_2 calculation method presented in paragraph 5.2.8, the m_{CO2} produced by boilers burning light heating oil will be:
(5-6) ⇒ (5-7), (5-8) (*Table 5.7*)⇒ **m_{CO2} = kgrCO_2/y** (η_{DIE}=0.8, APPENDIX B.3)
Assuming that diesel power plants exceed the CO_2 emission limit at about 3% then the mass of CO_2, which must be accounted for penalty, will be:
$m_{CO2,Di,L,pen}$ = kgrCO_2/year * 0.03 = **kgrCO_2/year**
Thus, **CO_2 emission cost** paid by island consumers = $m_{CO2,Di,b,pen}$ * €/1,000kgr = **€/year (varies with the CO_2 penalty price)**

Total CO_2 emission cost = CO_2 emission cost paid by PPC **+** CO_2 emission cost paid by island consumers = **€/year**

<u>1-Shaft GT, HE</u>

The only difference is concerning the GT package cost, (+30%).

<u>Scenario 3:</u> Minimum electric capacity GT

In this scenario the GTs have the power at the design point, of the minimum lighting & others electric power between the months of the year (minimum value of column 3, - November - *Table 5.12).* Thus, the needed surplus electric energy is supplied from the conventional plant. This also means that if there is a lack of heating energy, which is necessary for the proper operation of absorption chiller system or the heating system, will be covered with conventional air conditions and boilers respectively. The block diagram of the technical configuration is showed in *Fig. 5.13*.

In this mode the following points (key points) must be taken into consideration:

1. Selection of the GT power (choosing the TET, R_c from diagrams *Fig. 3.2, 3.4, 3.6* and regulating the mass flow of the engine at the DP performance) in such way to cover only the electric, power and energy demand of the lowest energy demand month. The monthly operation is characterised by constant TET (the same with the DP performance), while the ambient conditions P_a T_a are vary according to the conditions referred in *Table 2.1* (OD performance).

The selected engine has the best η_{th} for the maximum TET without cooling system.
2. 3. 4. and 5. key points are same as scenario2

Having the above in mind the economic simulation procedure is as following:

<u>1-Shaft GT</u>, <u>2-Shaft GT</u>

- Two GT package cost -**which has fix value**- including the necessary generator, the contribution devices, while the price of the gear box is assumed relatively negligible:

$$\text{GT Package Cost} = 2 * [\text{MW}] * [\$/\text{kW}] * 1000 / 1.23 = \text{€} \qquad (5\text{-}80)$$

where MW is the minimum of cells of column 3, November (*Table 5.12*), $/kW is the corresponding price of column (see APPENDIX E.1), the factor 1000 is due to transformation from kW to MW and finally factor 1.23 is due to transformation from $ to €.
GT package installation cost = **€ [scenario 1, paragraph 5.4.2,** (5-38)]
GT package maintenance cost = **€/year [scenario 1, paragraph 5.4.2,** (5-39)**]**

- The heat exchanger cost -**which has fix value**- including the installation cost, is given by the following equation:

$$\text{Heat Exchanger Cost} = 2 * [\text{MW}] * [\text{€/MW}] = \text{€} \qquad (5\text{-}81)$$

where factor 2 is due to the number of GTs, MW is the Maximum capability of GT heat power production and €/MW is explained in paragraph 5.2.3.
Heat exchanger maintenance cost = **€/year [scenario 1, paragraph 5.4.2,** (5-41)**]**

- The district heating installation cost -**which has fix value**- is given by the following equation:

$$\text{District heating installation cost}=2 * 0.6 * MW_a * [\text{€/MW}] * (MW_b) / 120 = \text{€} \qquad (5\text{-}82)$$

where factor 2 is due to the of two GTs, 0.6 is due to the fact that the exhaust heat capability of the GTs is over covering the 60% of the demand in cooling, MW_a are cells of column 2 plus cells of column 4 plus cells of column 5, (*Table 5.12*) multiply 0.6, €/MW is the corresponding price (paragraph 5.2.5), MW_b is the maximum sum of cells of columns 2+4+5, (*Table 5.12)* and the factor 120 is due to the relatively small system (paragraph 5.2.5).

- The four absorption chillers cost -**which has fix value**- including the installation cost is given by the following equation:

$$\text{Absorption Chiller Cost} = 4 * 0.6 * ([\text{MW}]/4) * [\text{€}/MW_c] = \text{€} \qquad (5\text{-}83)$$

where MW is the maximum cell of column 2 (*Table 5.12*) and €/MW_c is the corresponding price (paragraph 5.2.4).
Absorption Chiller Maintenance Cost = **[scenario 1, paragraph 5.4.2,** (5-44)**]**.

- The cooling power of the GTs is proved to produce enough heat to cover the 60% of the cooling demand. So, the power of the conventional cooling systems will be the needed 40% of the cooling power, which working regularly and the 60% of the cooling power, which are for back up reasons working regularly only for 15 days in November
Electric compression refrigeration system (conventional cooling system)
The capital cost plus the installation cost -**which has fix value-** is

$$(5\text{-}20) \Rightarrow \text{Capital cost + Installation cost} = \text{MW} * \text{€/MW} = \text{€}$$

where MW is the maximum cell of column 2, *Table 5.12.*

Operation cost
The electric energy per month supplied from the conventional plant for cooling ($W_{e,c}$), can be calculated as follows:

$$(5\text{-}21) \Rightarrow COP=4.5 \Rightarrow (\text{Eq.}4\text{-}3) \Rightarrow W_{e,c}=[0.4*MWh_a*30+0.6*MWh_b*15]/4.5=MWh$$

where Mwh_a are the corresponding values of cells of column 2, (*Table 5.16),* MWh_b is the value of November-cell of column 2, (*Table 5.16),*30 is the number of days of the month and 15 is the number of days of November.
Then the operation cost per month is given by the following equation:

$$(5\text{-}67) \Rightarrow \text{Operation Cost per month} = W_{e,c} * 117{,}08 * 30 = €/\text{month}$$

where 117.08€/MWh is the equivalent electricity price for the entire Island and 30 is assumed the number of days of the month.
The operation cost of the year can easily calculated as

$$(5\text{-}68) \Rightarrow \text{Operation cost} = \sum_{1}^{12} (€/\text{month}) = \mathbf{€/year}$$

which varies accordingly to the inflation rate.
Finally, the maintenance cost **-which has fix value, for every year-** is given accordingly to paragraph 5.2.7:
(5-24) ⇒ Maintenance Cost = (Capital cost + Installation cost) ∗ 0.90 ∗ 0.03**= €/year**
We assume that the maintenance of the cooling system is taking place in January (last week)

- The NG mass flow per month is given by

  $$(5\text{-}48) \Rightarrow \text{NG mass flow per month} = \dot{m}_f * 60 * 60 * 24 * 30 = \dot{m}_{fm}\ \text{kgr/month}$$

  where first factor 60 is for the conversion of seconds to minutes, second factor 60 is for conversion of minutes to hours, factor 24 is for the conversion of hours to days, while factor 30 corresponds to the number of days of the month, except from November which is assumed to operate 15 days, due to shut down for annual service
  Thus, the NG mass flow per year is

  $$(5\text{-}49) \Rightarrow \text{NG mass flow per year} = \dot{m}_{fy} = \sum_{1}^{12} \dot{m}_{fm}\ \text{kgr/year}$$

  The cost of NG per year -**which varies accordingly to the international oil prices-** is given by the equation:
  (5-78)⇒Cost of NG per year= $\dot{m}_{fy}$ ∗ 1.11 ∗ 48.9 ∗ 0.0002778 ∗ (2.39t+14.61)= **€/year**
  where factors 1.11 and 48.9MJ/kgr is accordingly to paragraph 5.2.7, while the factor 0.0002778 is due to transformation from MJ to MWh. Finally the factor (2.39t+14.61) is equation (5-5).

- The heating power of the GT is proved to produce enough heat to cover the 60% of the boilers heating demand. So, the power of boilers will be the needed 40% of the boilers heating power, which working regularly and the 60% of the boilers heating power, which are for back up reasons working regularly only for 15 days in November
  The capital cost -**which has fix value-** is given by

  $$(5\text{-}25) \Rightarrow \text{Capital Cost} = MW * \mathbf{€/MW = €}$$

  where MW is the maximum value among the cells of column 5, *Table 5.12* and €/MW is the price of boiler per MW (see paragraph 5.2.3)
  (boilers using light heating oil from *Table 5.4*: ρ =0.86kgr/lt, FCV=42.5MJ/kgr)
  The installation cost -**which has fix value-** is given by

  $$(5\text{-}26) \Rightarrow \text{Installation cost} = \text{Capital Cost} * 0.10 = \mathbf{€}$$

  where factor 0.10 is explained in paragraph 5.2.3

Finally, the maintenance cost **-which has fix value, for every year-** is given accordingly to paragraph 5.2.7:

(5-31) $\Rightarrow$ Maintenance Cost = Capital cost $*$ 0.02 **= €/year**

We assume that the maintenance of the heating system is taking place when there is no need for heating e.g. summer time (last week of June)

Energy provided from the light heating oil is calculated as follow:

$\eta_{th,\,b}$ = 0.8 $\Rightarrow$ <Eq. (4-3)> $\Rightarrow$ $Q_{f,\,b}$ = (0.4 $*$ MWh_a $*$ 30) **+** (0.6 $*$ MWh_b $*$ 15)**/** η_{HE} = MWh/m per month.

where $\eta_{th,\,b}$ is given in paragraph 5.2.3, MWh_a are the corresponding values of cells of column 5 -except November- (*Table 5.13)*, MWh_b is the corresponding values of November-cell of column 5 (*Table 5.13)* and 30 is assumed the number of days of the month (15 for November).

The mass of light diesel per month and per year are calculated as follow:

$$(5\text{-}28) \Rightarrow m_{fm,b} = [(Q_{f,\,b} * 3{,}600) / FCV\ ] = \text{kgr/month}$$

$$(5\text{-}29) \Rightarrow m_{fy,b} = \sum_{1}^{12} m_{fm,b} = \text{kgr/year}$$

where FCV is taken from *Table 5.4*.

And so the cost of light diesel per year **-which varies accordingly to the international oil prices,** can be estimated:

$$(5\text{-}79) \Rightarrow \text{Cost of light diesel} = ((\dot{m}_{fy,b} \,/\, 0.86\,) * (2.9t+42.1) \;/\; 159) * 1.9 = \textbf{€/year}$$

where 0.86 is the density ρ of the light diesel (paragraph 5.), (2.9t+42.1) is equation (5-1), while the factor 159 is the capacity of barrel (paragraph 5.2.7) and 1.9 is due to the isolated consumption area namely Island paragraph 5.2.7.

The power of the GTs is proved to produce not enough electricity to cover the lighting & others demand. So, the electricity power for the electric heating power will come from the conventional plant.

$$\text{Operation cost of electr. heating energy} = \sum_{1}^{12} [(0.4*MWh_a)*30+(0.6*MWh_b)]* \text{€/MWh}$$

where MWh_a are the cells of column 4 -except November-, (*Table 5.13*) MWh_b is the ell of column 4, November, (*Table 5.13)*, €/MWh is the corresponding price (paragraph 5.2.9) and 30 is assumed the number of days of the month (15 for November).

- Connection to the grid cost = 0 **€ (fix)**

- The electricity cost **-which varies accordingly to the inflation rate -** is given by

$$\text{Electricity cost} = \sum_{1}^{12} \{[MWh]*30\textbf{-}[MW]*24*30*[\text{€/MWh}]\} = \textbf{€/year} \qquad (5\text{-}84)$$

where MWh are the cells of column 3 (*Table 5.13*), MW is the GT production capability of cooling energy per day, €/MWh is the corresponding price (paragraph 5.29) and 30 is assumed the number of days of the month (15 for November).

- Greek government is offering financial support -**which has fix value**- is given by the following equation:
  Greek Government Financial Support = 0.4 $*$ (GT package cost + GT package installation cost + Heat exchanger cost + District heating installation cost + Absorption chiller cost = **€** (5-85)
  where factor 0.4 is explained in paragraph 5.2.10

- CO_2 emissions estimation and penalty
  Energy per year <produced from the conventional plant, PPC> = $\sum_{1}^{12}\{(MWh_a*30-MW_a*24*30)+\{[0.4*MWh_b*30+0.6*MWh_c*15]/4.5\}+\{\sum_{1}^{12}(0.4*MWh_d*30+0.6*MWh_e*15)*30)\}$=MWh/year (5-86)
  where MWh_a are the cells of Column 3, (*Table 5.13*), 30 is the number of days of the month, MW_a is the GT production capability of cooling energy per day, 30 is the number of days of the month -November 15-, MWh_b are the cells of column 2, (*Table 5.16*), MWh_c is November-cell of column 2 (*Table 5.16*), MWh_d are the cells of column 4, (*Table 5.13*), 30 is the number of days of the month, MWh_e is the November-cell of column 4, , (*Table 5.13*) and 30 is the number of days of the month.

  The island is part of the non-interconnected national grid of Greece. Taking into account the data from *Table A.5*, it can be estimated how many MWh/year are produced from the available kind of power plants, (obviously, renewable excluded) assuming that the total electric energy per year is supplied only diesel power plants:
  Diesel (medium heating oil): MWh/year $*$ 0.985 = MWh/year
  Using the CO_2 calculation method presented in paragraph 5.2.8, the m_{CO2} produced from every type of power plant respectively, can be estimated:
  (5-6) $\Rightarrow$ (5-7), (5-8) (*Table 5.7*) $\Rightarrow$ $\mathbf{m_{CO2}}$ **= kgrCO$_2$/y** (η_{DIE}=0.36, APPENDIX B.3)
  Assuming that medium diesel power plants exceed the CO_2 emission limit at about 6% then the mass of CO_2, which must be accounted for penalty, will be:
  $\mathbf{m_{CO2,DiM,pen}}$ **= kgrCO$_2$/year** $*$ 0.06 = **kgrCO$_2$/year**
  Thus, **CO_2 emission cost** paid by PPC = $m_{CO2,Di,pen}$ $*$ €/1,000kgr = **€/year (varies with the CO_2 penalty price)**

  The island is using boilers burning only light heating oil. Thus, the electric energy per year produced burning light heating oil will be:
  Heating energy per year <boilers> = $\sum_{1}^{12}\{(0.4*MWh_a*30)+(0.6*MWh_b*15)/\eta_{HE}\}$ =MWh/year (5-87)
  where MWh_a are the cells of column 5, (*Table 5.13),* 30 is the number of days of the month, MWh_b is the November-cell of column 5, (*Table 5.13*) and 15 is the number of days of November.
  Using the CO_2 calculation method presented in paragraph 5.2.8, the m_{CO2} produced by boilers burning light heating oil will be
  (5-6) $\Rightarrow$ (5-7), (5-8) (*Table 5.7*)$\Rightarrow$ $\mathbf{m_{CO2}}$ **= kgrCO$_2$/y** (η_{DIE}=0.8, APPENDIX B.3)
  Assuming that diesel power plants exceed the CO_2 emission limit at about 3% then the mass of CO_2, which must be accounted for penalty, will be:

$m_{CO2,Di,L,pen}$ = kgrCO_2/year $*$ 0.03 = **kgrCO_2/year**
Thus, **CO_2 emission cost** paid by island consumers **=** $m_{CO2,Di,b,pen}$ $*$ €/1,000kgr = **€/year (varies with the CO_2 penalty price)**
Total CO_2 emission cost = CO_2 emission cost paid by PPC **+** CO_2 emission cost paid by island consumers = **€/year**

1-Shaft GT HE

The only difference is concerning the GT package cost, (+30%).

Scenario 4:*. *Covering the thermal and cooling demand GT,
The useful thermal and cooling output of the GTs, is equal to the 60% demand of thermal and cooling load, of any month. If the generated electricity is higher than the load, surplus electricity is sold to the grid; if it is lower, supplementary electricity is produced by the conventional power plant. The block diagram of the technical configuration is showed in *Fig. 5.14*.

In this mode the following points (key points) must be taken into consideration:

1. selection of the GT power (choosing the TET, R_c from diagrams *Fig. 3.2, 3.4, 3.6* and regulating the mass flow of the engine at the DP performance) in such way to cover the heating and cooling power and energy demand of the most demanded month. The monthly operation is characterised by constant TET (the same with the DP), while the ambient conditions P_a T_a are vary according to the conditions referred in *Table 2.8* (OD performance)

2., 3. 4. and 5. key points are same as scenario 2

Having the above in mind and observing the numbers of the *Tables 5.12* and *5.13*, it can be seen that in the case of the Island the scenario 4 is actually the same with scenario 3. This is because following the criteria of the scenario 3, results in almost equal cooling power of the GTs with the demand in January and December. So there is no change margin in the GTs power to fulfil the criteria of scenario 4.

5.6 Hotel energy scenarios

5.6.1 Conventional case

In this paragraph, the different costs of energy will be analytically presented, for the conventional namely the present energy situation of the Sani Beach Hotel. Essential assistant to that will be given by the data presented in CHAPTER 2. *Tables 2.15* and *2.16* are presenting the energy consumption and the power demand respectively, for a typical day each month of the year. As it has been said in paragraph 2.1, these values do not include either the hypothetical future increase, or the estimation of the worst case for the energy demand point of view. After a relevant discussion with the supervisor, the author decided to multiple by a factor of 1.2 all the prices of the above mentioned *Tables*, in order to include the worst case situation. The results are shown in *Tables 5.15* and *5.16*

Table 5.15: Sani Beach Hotel power demand in kW

MONTHS (30 days per month)	**COOLING** (kW_C)	**LIGHTING & OTHER** (kW_e)	**HEATING** kW_t		**TOTAL POWER**
			ELECTRIC	BOILERS	
JAN	24.00	98.94	0.06	1.94	124.92
FEB	24.00	78.23	0.05	1.93	104.21
MAR	24.00	61.36	0.04	1.94	87.34
APR	94.68	168.00	5.40	182.52	450.60
MAY	240.84	356.28	7.08	240.36	844.44
JUN	293.88	429.24	0.00	229.08	952.20
JUL	347.16	610.32	0.00	251.28	1,208.40
AUG	393.72	672.12	0.00	298.80	1,364.40
SEP	327.60	660.48	6.96	230.64	1,225.20
OCT	257.28	508.20	3.48	175.32	944.28
NOV	24.00	303.60	0.00	1.96	329.52
DEC	24.00	64.56	0.05	1.96	90.55

Table 5.16: Sani Beach Hotel energy demand kWh

MONTHS (30 days per month)	**COOLING** (kWh_C)	**LIGHTING & OTHER** (kWh_e)	**HEATING** kWh_t		**TOTAL POWER**
			ELECTRIC	BOILERS	
JAN	288	1,982	16	0.0	2,286
FEB	288	1,600	16	0.0	1,903
MAR	288	1,288	15	1.2	1,592
APR	1,202	3,682	2,189	66.0	7,140
MAY	4,045	7,024	2,882	86.4	14,038
JUN	5,771	7,579	2,749	0.0	16,099
JUL	7,232	10,445	3,016	0.0	20,692
AUG	7,662	12,014	3,586	0.0	23,262
SEP	6,998	11,288	2,767	84.0	21,138
OCT	5,287	8,867	2,104	42.0	16,300
NOV	288	5,760	16	0.0	6,064
DEC	288	1,348	16	0.0	1,651

1. <u>Cost of electricity (lighting, motion, etc)</u>

The electricity cost of each month (€/month) is

$$(5\text{-}18) \Rightarrow [€/month] = [MWh] * [€/MWh] * 30$$

where MWh is the corresponding to each month value of the cells of column 3, *Table 5.16*, €/MWh is the price of electricity per MWh (see paragraph 5.2.9) and 30 is assumed the number of days of the month.

The electricity cost of the year (12 months) is

$$(5\text{-}19) \Rightarrow \text{Cost of Electricity} = \sum_{1}^{12} (€/\text{month}) = \mathbf{[€/year]}$$

which varies accordingly to the inflation rate

2. <u>Cooling</u>

Electric compression refrigeration system.
The capital cost plus the installation cost -**which has fix value-** is

$$(5\text{-}20) \Rightarrow \text{Capital Cost + Installation Cost} = \text{MW} * €/\text{MW} = \mathbf{€}$$

where MW is the maximum value of the cells of column 2, *Table 5.15*, and €/MW is the price of electricity per MW (see paragraph 5.2.4)

The electric energy per month supplied from the local grid for cooling ($W_{e,\,c}$), can be calculated as follows:

$$(5\text{-}21) \Rightarrow COP = 4.5 \Rightarrow (\text{Eq. } 4\text{-}3) \Rightarrow W_{e,\,c} = \text{MWh} \,/\, 4.5 = \text{MWh}$$

where MWh are the corresponding values of cells of column 2, *Table 5.16*.
Then the operation cost per month is given by the following equation:

$$(5\text{-}22) \Rightarrow \text{Operation Cost per month} = W_{e,\,c} * [€/\text{MWh}] * 30 = €/\text{month}$$

where €/MWh is the price of electricity per MWh (see paragraph 5.2.9) and 30 is assumed the number of days of the month.
The operation cost of the year can easily calculated as

$$(5\text{-}23) \Rightarrow \text{Operation cost} = \sum_{1}^{12} (€/\text{month}) = \mathbf{€/year}$$

which varies accordingly to the inflation rate

Finally, the maintenance cost -**which has fix value, for every year-** is given accordingly to paragraph 5.2.7:

$$(5\text{-}24) \Rightarrow \text{Maintenance Cost} = (\text{Capital cost + Installation cost}) * 0.90 * 0.03 = \mathbf{€/year}$$

We assume that the maintenance of the cooling system is taking place in January (last week)

3. <u>Heating</u>

Heating is coming from two sources: a) from boilers using light heating oil as fuel and b) from electric inverters, heaters etc, which consume electricity and are heating the rooms.
a) Individual boilers using light diesel as fuel (from *Table 5.4*: ρ = 0.86kgr/lt, FCV=42.5MJ/kgr).
The capital cost -**which has fix value-** is given by

$$\text{Capital Cost} = 300\text{kW} * €/\text{kW} = \mathbf{€} \qquad (5\text{-}88)$$

where MW €/MW is the price of boiler per kW (see paragraph 5.2.3)
The installation cost -**which has fix value-** is given by

$$(5\text{-}26) \Rightarrow \text{Installation cost} = \text{Capital Cost} * 0.10 = \mathbf{€}$$

where factor 0.10 is explained in paragraph 5.2.3

The energy provided from the light heating oil per month is calculated as follow:

(5-27) $\Rightarrow \eta_{th,\,b} = 0.8 \Rightarrow$ <Eq. (4-3)> $\Rightarrow Q_{f,\,b}$ = (MWh $*$ 30) / 0.8 = MWh per month

where $\eta_{th,\,b}$ is given in paragraph 5.2.3, MWh are the corresponding values of cells of column 5, *Table 5.16* and 30 is assumed the number of days of the month.

The mass of light heating oil per month and per year are calculated as follow:

$$(5\text{-}28) \Rightarrow \ m_{fm,b} = [(Q_{f,\,b} * 3{,}600) / FCV\ ] = \text{kgr/month}$$

$$(5\text{-}29) \Rightarrow \ m_{fy,b} = \sum_{1}^{12} m_{fm,b} = \text{kgr/year}$$

where FCV is taken from *Table 5.4*.

And so the cost of medium heating oil per year **-which varies accordingly to the international oil prices** can be estimated:

Cost of light heating oil = (($\dot{m}_{fy,b}$ / 0.86) $*$ (2.9t+42.1) / 159) $*$ 1.8 = **€/year** (5-89)

Where 0.92 is the density ρ of the light heating oil (paragraph 5.4), (2.9t+42.1) is equation (5-1), while the factors 159 is the capacity of barrel (paragraph 5.2.7) and 1.8 is due to the isolated consumption area namely Island, paragraph 5.2.7.

Finally, the maintenance cost **-which has fix value, for every year-** is given accordingly to paragraph 5.2.7:

(5-31) $\Rightarrow$ Maintenance Cost = Capital cost $*$ 0.02 **= €/year**

We assume that the maintenance of the heating system is taking place when there is no need for heating, for example when the hotel is close.

b) Electric inverters, heat pumps, heaters etc

Capital cost and installation cost and maintenance cost is negligible due to the fact that the majority of them are used for cooling and heating, so the capital cost of them is already calculated in the previous cooling section

Operation cost.

COP = 3.0 is the average COP of heat pumps heaters, etc, (paragraph 4.6.1). Thus with the help of equation (4-3) in paragraph 4.6.1:

$W_{e,\,h}$ = [Cells of Column 4 (*Table 5.15*)] / 3.0 = kWh (5-90)

Operation cost per month = $W_{e,\,c}$ **X** €/kWh **X** 30<days of the month> = €/month

Operation cost = $\sum_{1}^{12}$ (€/month) **= €/year (varies accordingly to the inflation rate)**

4. <u>Back up generator</u>

Capital Cost and Installation Cost of back up generator = kW $*$ **€**/kW= **€ (fix)** (5-91)

where kW is the maximum sum of cells of columns 2 plus 3 plus 4 (*Table 5.15*) and €/kW is the corresponding price (paragraph 5.2.6).

5. <u>CO_2 emissions estimation and penalty</u>

The energy per year supplied from national grid (PPC) is calculated as follow:

$$(5\text{-}33) \Rightarrow \text{Energy per year} = \sum_{1}^{12} \ \text{values per typical day} * 30 = \text{MWh/year}$$

where the values per typical day are the cells of columns 2 plus 3 plus 4 of *Table 5.16* and 30 is assumed the number of days of the month.

The hotel is connected with the interconnected national grid (continental) of Greece. Taking into account the data from *Table A.5*, it can be estimated how many kWh/year are produced from the available kinds of power plants, assuming that analogical distribution of the total electric energy per year:

Lignite: kWh/year $*$ 0.674 = kWh/year

Natural gas: kWh/year $*$ 0.168 = kWh/year

Diesel (heavy heating oil): kWh/year $*$ 0.056 = kWh/year

Using the CO_2 calculation method presented in paragraph 5.2.8, the m_{CO2} produced from every type of power plant respectively, can be estimated:

(5-6) $\Rightarrow$ (5-7), (5-8) (*Table 5.7*) $\Rightarrow$ $\mathbf{m_{CO2}}$ **= kgrCO$_2$/y** (Assuming η_{LIG}=0.3)

Assuming that lignite power plants exceed the CO_2 emission limit at about 10% then the mass of CO_2, which must be accounted for penalty, will be:

$\mathbf{m_{CO2,Li,pen}}$ **= kgrCO$_2$/year** $*$ 0.1 = **kgrCO$_2$/year**

(5-6) $\Rightarrow$ (5-7), (5-8) (*Table 5.7*) $\Rightarrow$ $\mathbf{m_{CO2}}$ **= kgrCO$_2$/y** (Assuming η_{NG}=0.55)

Assuming that NG power plants exceed the CO_2 emission limit at about 0% then the mass of CO_2, which must be accounted for penalty, will be:

$\mathbf{m_{CO2,NG,pen}}$ **= kgrCO$_2$/year** $*$ 0.0 = **kgrCO$_2$/year**

Finally, (5-6) $\Rightarrow$ (5-7), (5-8) (*Table 5.7*) $\Rightarrow$ $\mathbf{m_{CO2}}$ **= kgrCO$_2$/y** (η_{DIE}=0.36, APPENDIX B.3)

Assuming that diesel power plants exceed the CO_2 emission limit at about 7% then the mass of CO_2, which must be accounted for penalty, will be:

$\mathbf{m_{CO2,Di,pen}}$ **= kgrCO$_2$/year** $*$ 0.07 = **kgrCO$_2$/year**

Thus, **CO_2 emission cost** paid by PPC = ($m_{CO2,Li,pen}$ + $m_{CO2,NG,pen}$ + $m_{CO2,Di,pen}$) $*$ 30€/1,000kgr = **€/year (varies with the CO_2 penalty price)**

The hotel is using boilers burning only light heating oil. Thus the electric energy per year produced burning light heating oil will be:

$$\text{(5-35)} \Rightarrow \text{Heating energy per year of boilers} = \sum_{1}^{12} \text{(values per typical day)} * 30 = \text{MWh/y}$$

where the values per typical day are the cells of column 5, *Table 5*.16 and 30 is assumed the number of days of the month.

Using the CO_2 calculation method presented in paragraph 5.2.8, the m_{CO2} produced by boilers burning light heating oil will be

(5-6) $\Rightarrow$ (5-7), (5-8) (*Table 5.7*)$\Rightarrow$ $\mathbf{m_{CO2}}$ **= kgrCO$_2$/y** (η_{DIE}=0.8, APPENDIX B.3)

Assuming that diesel power plants exceed the CO_2 emission limit at about 3% then the mass of CO_2, which must be accounted for penalty, will be:

$\mathbf{m_{CO2,Di,L,pen}}$ **= kgrCO$_2$/year** $*$ 0.03 = **kgrCO$_2$/year**

Thus, **CO_2 emission cost** paid by the hotel = $m_{CO2,Di,b,pen}$ $*$ €/1,000kgr= **€/year (varies with the CO_2 penalty price)**

Total CO_2 emission cost = CO_2 emission cost paid by PPC **+** CO_2 emission cost paid by the hotel = **€/year**

5.6.2 Hypothetical operation scenarios

A scenario-mode of operation is characterised by the criterion on which the adjustment of the electrical and useful thermal-cooling output of a trigeneration system is based. There are various modes of operation possible, the most distinct of those being the following:

<u>Scenario 1</u>: Maximum capacity GT, operation scenario

There is complete coverage of the electrical, thermal and cooling loads at any instant of time. The possible excess in electric power supplies the national local grid. This is the most expensive strategy, at least from the point of view of initial cost of the system. The block diagram of the technical configuration is showed in *Fig. 5.11*.

In this mode the following points (key points) must be taken into consideration:

1. Selection of the GT power (choosing the TET, R_c from diagrams *Fig. 3.2, 3.4, 3.6*) and regulating the mass flow of the engine at the DP performance in such way to cover the electric, heating and cooling power and energy demand of the most energy-demanded month. The yearly operation is characterised by constant TET (the same with the DP), while the ambient conditions P_a T_a are vary according to the conditions referred in *Table 2.1* (OD performance). The selected engine has the best η_{th} for the maximum TET without cooling system.
2. The proportional factor z is representing the way of the power or the energy of the GT exhaust gasses is split between the heating and cooling demand every month.
3. The availability of the plant is assumed to be about 98%, in other words the plant is assumed to shut down for one week (the first of November, when total needs are minimum) for the annual service of the entire system. During that week electricity is supplied from the local grid while heating is supplied from a stand by boiler.
4. Salaries for extra personnel are assumed to be negligible.

Having the above in mind the economic simulation procedure for the 1-Shaft GT, is as following:

<u>1-Shaft GT</u>, <u>2-Shaft GT</u>

- One GT package cost **-which has fix value-** including the necessary generator, the contribution devices, while the price of the gear box is assumed relatively negligible:

$$(5\text{-}37) \Rightarrow \text{GT Package Cost} = [\text{kW}] * [\$/\text{kW}] / 1.23 = \textbf{€}$$

where MW is the useful work of the GT at the design point, $/kW is the corresponding price of column (see APPENDIX E.1) and finally factor 1.23 is due to transformation from $ to €.

The GT package installation cost **-which has fix value-** is given by the following equation:

$$(5\text{-}38) \Rightarrow \text{GT Package Installation Cost} = \text{GT Package Cost} * 0.1 = \textbf{€}$$

where factor 0.1 is explained in paragraph 5.2.1.

The GT package maintenance cost **-which has fix value, for every year-** is given by the following equation:

$$(5\text{-}39) \Rightarrow \text{G T Package Maintenance Cost} = \text{GT package cost} * 0.01 = \textbf{€/year}$$

where factor 0.01 is explained in paragraph 5.2.7.

- The heat exchanger cost -**which has fix value**- including the installation cost, is given by the following equation:

$$(5\text{-}40) \Rightarrow \text{Heat Exchanger Cost} = [\text{MW}] * [€/\text{MW}] = \mathbf{€}$$

where MW is the maximum cell of columns 2 plus 4 plus 3 (*Table 5.15*) and €/MW is explained in paragraph 5.2.3.

The heat exchanger maintenance cost -**which has fix value, for every year-** is given by the following equation:

(5-41)⇒Heat Exchanger Maint. Cost = 0.9 * [Heat Exchanger Cost] * 0.02 = **€/year**

where factors 0.9 and 0.02 are explained in paragraphs 5.2.3 and 5.2.7 respectively

- The district heating installation cost -**which has fix value**- is given by the following equation:

  District Heating Installation Cost = [kW] * [€/kW] * ([kW]/120,000) = **€** (5-92)

  where MW is the maximum cell of columns 2, 4 plus (*Table 5.15*), €/MW is the corresponding price (paragraph 5.2.5), kW is the maximum cell of columns 2 plus 3 (*Table 5.10*) and the factor 120,000 is due to the relatively small system (paragraph 5.2.5).

- The absorption chiller cost -**which has fix value**- including the installation cost is given by the following equation:

  $$(5\text{-}43) \Rightarrow \text{Absorption Chiller Cost} = [\text{kW}] * [€/\text{kW}_c] = \mathbf{€}$$

  where kW is the maximum cell of column 2 (*Table 5.15*) and €/kW_c is the corresponding price (paragraph 5.2.4).

  The absorption chiller maintenance cost -**which has fix value, for every year-** is given by the following equation:

  (5-44)⇒Absorption Chiller Maint. Cost=0.8*Absorption chiller cost*0.031= **€/year**

  where factors 0.8 and 0.031 are explained in paragraphs 5.2.4 and 5.2.7 respectively.

- Cost of back up cooling.

  The electric compression refrigeration system capital cost including the installation cost -**which has fix value**- is given by the following equation:

  (5-45) ⇒ Electric Compression Refrigeration System Capital Cost+Installation Cost = 0.5 * [kW] * [€/kW] = **€**

  where the factor 0.5 is due to the assumption that the back up cooling power is the 50% of the maximum cooling demand power in kW, kW is the maximum value of cells of column 2, (*Table 5.15*) and €/kW is the corresponding price (paragraph 5.2.4)

  The operation cost of electric compression refrigeration system is calculated as follows:

  Operation back up cooling energy: COP=4.5⇒(Eq. 4-3)⇒$W_{e,c}$= [kWh]/4.5=kWh

  where $W_{e,c}$ is the electric energy supplied from the local grid for cooling and kWh corresponds to the December -cells of column 2 (*Table 5.16*)-.

  Operation cost in December = $W_{e,c}$ * [€/kWh] <paragraph 5.2.9> * 7 = €/month

  where €/kW is the corresponding price (paragraph 5.2.9), and 7 is the number of November days when the back cooling system works.

  $$(5\text{-}46) \Rightarrow \text{Operation cost} = \sum_{1}^{12} (€/\text{month}) = \mathbf{€/year}$$

which varies accordingly to the inflation rate.

The maintenance cost -**which has fix value, for every year-** is given by the following equation:

(5-47) ⇒Maint. Cost=([Capital Cost]+[Installation Cost])$*$0.90$*$0.03$*$(7/360)**=€/year**
where factors 0.90 and 0.03 are discussed in paragraphs 5.2.4, 5.2.7, while the factor (7/360) is simulates the relative duration of the operation
We assume that the maintenance of the cooling system is taking place in December (last week)

- The NG mass flow per month is given by

  (5-48) ⇒NG mass flow per month = $\dot{m}_f * 60 * 60 * 24 * 30 = \dot{m}_{fm}$ kgr/month

  where first factor 60 is for the conversion of seconds to minutes, second factor 60 is for conversion of minutes to hours, factor 24 is for the conversion of hours to days, while factor 30 corresponds to the number of days of the month, except from November which is assumed to operate 23 days, due to shut down for annual service
  Thus, the NG mass flow per year is

  $$(5\text{-}49) \Rightarrow \text{NG mass flow per year} = \dot{m}_{fy} = \sum_{1}^{12} \dot{m}_{fm} \text{ kgr/year}$$

  The cost of NG per year -**which varies accordingly to the international oil prices-** is given by the equation:

  Cost of NG per year = $\dot{m}_{fy} * 1.11 * 48.6 * 0.2778 * (2.39t+15.61)$ = **€/year** (5-93)

  where factors 1.11 and 48.6MJ/kgr is accordingly to paragraph 5.2.7, while the factor 0.2778 is due to transformation from MJ to kWh. Finally, the factor (2.39t+15.61) is equation (5-4).

- Cost of back up boilers (assume 50% of the maximum heating demand power in MW) using light heating oil. *Table 5.4*: ρ =0.86kgr/lt, FCV=42.5MJ/kgr
  Using the same methodology as in paragraph 5.4.1 and especially the part labeled heating. The capital cost (**fix value**) and the installation cost (**fix value**) can be calculated with the help of equations (5-25) and (5-26) respectively, while the maintenance cost (**fix value, for every year)** is given by the equation (5-31) with a slight modulation:

  (5-51) ⇒ Maintenance Cost = [capital cost] $*$ 0.02 $*$ (7/360) = **€/year**

  where the factor (7/360) is due to the fact that operates regularly only 7 days per year. Assume **no operation cost**.

- The connection to the grid cost -**which has fix value**- is given by the following equation:

  (5-54) ⇒ Connection to the grid Cost = kW $*$ [€/kW$_e$] = **€**

  where kW is the maximum difference between the monthly GT power production and the corresponding cell of column 3 (*Table 5.15*), because that is the maximum power difference that might be sold to the local grid and €/kW$_e$ is the corresponding price (paragraph 5.2.6).

- The electricity cost **-which varies accordingly to the inflation rate**- is given by the equation

$$\text{Electricity cost} = \text{kWh} * €/\text{kWh} * 7 = \mathbf{€/year} \qquad (5\text{-}94)$$

- Where kWh are the December-cells of column 2+3+4, (*Table 5.16*), factor €/kWh is the corresponding price (paragraph 5.29) and because no electric energy is imported, except from the 1 week = 7 days in December, when service works are in process.

- Excess of electric energy per month = (energy produced by the GT when it is operating 24 hours per day for 30 days per month, -23 for December-) **–** (cells for each month of column 3, *Table 5.16*) = kWh

  Electricity profit per year = $\sum_{1}^{12}$ Excess of electric energy per month $*$ 30 * 68 €/kWh<paragraph 5.2.9> = **€/month (varies accordingly to the inflation rate)**

- Greek government is offering financial support -**which has fix value**- is given by the following equation:
  (5-57)⇒Greek Government Financial Support = 0.4 $*$ (GT package cost + GT package installation cost + Heat exchanger cost + District heating installation cost + Absorption chiller cost + Capital cost of back up boiler + Installation cost of boiler + Connection to the grid cost) = **€**
  where factor 0.4 is explained in paragraph 5.2.10

- CO_2 emissions estimation penalty.
  Total CO_2 emission cost = 0€ because it is assumed that trigeneration plants do not exceeds the official limits of CO_2 emissions. Emissions from the operation of the back up boiler or from the power plants of PPC to produce the electricity during the shut down period of the GT, are assumed negligible.

- CO_2 emissions estimation profit.
  The excess of energy per year is supplied to national grid is given by the equation
  (5-58)⇒Excess of energy per y= $\sum_{1}^{12}$ Excess of electric energy per month **=** MWh/year
  The hotel is connected with the interconnected national grid (continental) of Greece. Taking into account the data from *Table A.5*, it can be estimated how many kWh/year are produced from the available kinds of power plants, assuming that analogical distribution of the total electric energy per year:
  Lignite: MWh/year $*$ 0.674
  Natural gas: MWh/year $*$ 0.168
  Diesel (heavy heating oil): MWh/year $*$ 0.056
  Using the CO_2 calculation method presented in paragraph 5.2.8, the m_{CO2} produced from every type of power plant respectively, can be estimated:
  Eq.(5-6) ⇒ [Eqs (5-7), (5-8) (*Table 5.7*)] ⇒ $\mathbf{m_{CO2}}$ **= kgrCO$_2$/year** (Assuming η_{LIG}=0.3)
  Assuming that lignite power plants exceed the CO_2 emission limit at about 10% then the mass of CO_2, which must be accounted for penalty, will be:
  $\mathbf{m_{CO2,Li,pen}}$ **= kgrCO$_2$/year** $*$ 0.1 **= kgrCO$_2$/year**

Similarly (5-6) $\Rightarrow$ (5-7), (5-8) (*Table 5.7*) $\Rightarrow$ $\mathbf{m_{CO2}}$ **= kgr$\mathbf{CO_2}$/year** (Assuming η_{NG}=0.55)
Assuming that NG power plants exceed the CO_2 emission limit at about 0% then the mass of CO_2, which must be accounted for penalty, will be:
$\mathbf{m_{CO2,NG,pen}}$ **= kgr$\mathbf{CO_2}$/year** $*$ 0.0 = **kgr$\mathbf{CO_2}$/year**
Finally, (5-6) $\Rightarrow$ (5-7), (5-8) (*Table 5.7*) $\Rightarrow$ $\mathbf{m_{CO2}}$ **= kgr$\mathbf{CO_2}$/y** (η_{DIE}=0.36, APPENDIX B.3)
Assuming that diesel (heavy) power plants exceed the CO_2 emission limit at about 7% then the mass of CO_2, which must be accounted for penalty, will be:
$\mathbf{m_{CO2,Di,pen}}$ **= kgr$\mathbf{CO_2}$/year** $*$ 0.07 = **kgr$\mathbf{CO_2}$/year**
Thus, **$\mathbf{CO_2}$ emission cost -which varies with the $\mathbf{CO_2}$ penalty price-** paid by PPC is
(5-59)$\Rightarrow$$CO_2$ emission cost paid by PPC = ($m_{CO2,Li,pen}$ + $m_{CO2,NG,pen}$ + $m_{CO2,DiH,pen}$) $*$ (€/1,000) = **€/year**
where factor 1,000 is for units similarity (paragraph 5.2.8)

1-Shaft GT, HE

The only difference is concerning the GT package cost, (+30%).

<u>Scenario 2:</u> Maximum capacity GT, following the total demand load scenario

It is the same scenario with the previous, but the system is always working only to cover all its needs at any time. The distribution of the power demand is such (cooling power is higher than heating and electric) that covering the cooling power there is an excess of electric energy to export to the local national grid.The block diagram of the technical configuration is showed in *Fig. 5.12*.

In this mode the following points (key points) must be taken into consideration:

1. selection of the GT power (choosing the TET, Rc from diagrams *Fig. 3.2, 3.4, 3.6* and regulating the mass flow of the engine at the DP performance) in such way to cover the electric, heating and cooling power and energy demand at any month. The monthly operation is characterised by variant TET (less than that of the DP performance,), while the ambient conditions P_a T_a are vary according to the conditions referred in *Table 2.1* (This is actually part load performance of the GT)

The selected engine has the best η_{th} for the maximum TET without cooling system.
2. 3. and 4. key points are same as scenario1

Having the above in mind the economic simulation procedure is as following:

<u>1-Shaft GT</u>, <u>2-Shaft GT</u>

- One GT package cost (generator, included) = **€ [same as in scenario 1,** (5-37)**]**
  GT package installation cost = **€ [scenario 1,** (5-38)**]**
  GT package maintenance cost = **€/year [scenario 1,** (5-39)**]**

- Heat exchanger cost (installation cost, included) = **€ [scenario 1,** (5-40)**]**
  Heat exchanger maintenance cost = **€/year [scenario 1,** (5-41)**]**

- District heating installation cost = **€ [scenario 1,** (5-92)**]**

- Absorption chiller cost (installation cost, included) = **€ [scenario 1,** (5-43)**]**

Absorption chiller maintenance cost = **€/year [scenario 1,** (5-44)**]**

- Cost of back up cooling
  The electric compression refrigeration system capital cost **-which has fix value-** including the installation cost can be calculated with the help of equation (5-45) **[scenario 1,** (5-45)**]** where again the factor 0.5 is due to the assumption that the back up cooling power is the 50% of the maximum cooling demand power in kW, kW is the maximum value of cells of column 2, (*Table 5.15*) and €/kW is the corresponding price (paragraph 5.2.4)
  Operation cost = **€/year [scenario 1,** (5-46)**]**
  Maintenance cost = **€/year [scenario 1,** (5-47)**]**
  We assume that the maintenance of the cooling system is taking place in December (one week)

- NG mass flow per month = **[scenario 1,** (5-48)**]**
  NG mass flow per year = **[scenario 1,** (5-49)**]**
  Cost of NG per year = **[scenario 1,** (5-50)**]**

- Cost of back up boilers (assume 50% of the maximum heating demand power in MW) using light heating oil. *Table 5.4*: ρ =0.86kgr/lt, FCV=42.5MJ/kgr>
  Capital cost = **€ [scenario 1,** (5-25)**]**
  Installation cost = **€ [scenario 1,** (5-26)**]**
  Maintenance cost = **€/year [scenario 1,** (5-51)**]**
  Cost of medium heating oil = **€/year [scenario 1,** (5-53)**]**

- Connection to the grid cost = **€ [scenario 1,** (5-54)**]**

- Electricity cost = **€/year [scenario 1,** (5-55)**]**

- Electricity profit = **€/year [[scenario 1,** (5-56)**]**

- Greek government is offering financial support = **€ [scenario 1,** (5-57)**]**

- CO_2 emissions estimation penalty = **€/year [scenario 1]**

<u>**1-Shaft GT HE**</u>

The only difference is concerning the GT package cost, (+30%).

***Scenario 3:* Minimum electric capacity GT**

In this scenario the GT has the power, of the minimum electric power between the months of the year (minimum value of columns 3, -March- *Table 5.15).* Thus, the needed surplus electric energy is supplied from the local national grid. This also means that if there is a lack of heating energy, which is necessary for the proper operation of absorption chiller system, will be covered with conventional air conditions. Finally, the possible lack of heating power will be covered from the use of boilers. The block diagram of the technical configuration is showed in *Fig. 5.13*.

In this mode the following points (key points) must be taken into consideration:

1. Selection of the GT power (choosing the TET, R_c from diagrams *Fig. 3.2, 3.4, 3.6* and regulating the mass flow of the engine at the DP performance) in such way to cover only the electric, power and energy demand of the lowest energy demand month. The monthly operation is characterised by constant TET (the same with the DP performance), while the ambient conditions P_a T_a are vary according to the conditions referred in *Table 2.1* (OD performance)

The selected engine has the best η_{th} for the maximum TET without cooling system.

2. 3. and 4. key points are same as scenario1

Having the above in mind the economic simulation procedure is as following:

1-Shaft GT, 2-Shaft GT

- One GT package cost **-which has fix value-** including the necessary generator, the contribution devices, while the price of the gear box is assumed relatively negligible:

  (5-37) $\Rightarrow$ GT Package Cost = [kW] $*$ [$/W] $*$ 1000 / 1.23 = **€**

  where MW is minimum of cells of column 3 (*Table 5.15*), $/kW is the corresponding price of column (see APPENDIX E.1) and finally factor 1.23 is due to transformation from $ to €.

  GT package installation cost = **€ [scenario 1,** (5-38)]

  GT package maintenance cost = **€/year [scenario 1,** (5-39)**]**

- The heat exchanger cost **-which has fix value-** including the installation cost, is given by the following equation:

  (5-40) $\Rightarrow$ Heat Exchanger Cost = [kW] $*$ [€/kW] = **€**

where kW is the maximum capability of GT heat power production and €/kW is explained in paragraph 5.2.3.

  Heat exchanger maintenance cost = **€/year [scenario 1,** (5-41)**]**

- The district heating installation cost **-which has fix value-** is given by the following equation:

  (5-42) $\Rightarrow$ District Heating Installation Cost = [kW] $*$ [€/kW] $*$ ([kW]/120,000) = **€**

  where kW is the capability of GT heat power production, €/kW is the corresponding price (paragraph 5.2.5), kW is the maximum capability of GT heat power production and the factor 120,000 is due to the relatively small system (paragraph 5.2.5).

- The absorption chiller cost -**which has fix value**- including the installation cost is given by the following equation:

  Absorption Chiller Cost = [kW] $*$ [1/(z+1)] $*$ [€/kW$_c$] = **€** (5-94)

  where kW is the maximum capability of GT heat power production, factor 1/(z+1) is the portion of GT exhaust heat going for cooling and €/kW$_c$ is the corresponding price (paragraph 5.2.4).

  The absorption chiller maintenance cost -**which has fix value, for every year-** is given by the following equation:

  (5-44)$\Rightarrow$Absorption Chiller Maint. Cost =0.8$*$Absorpt. chiller cost $*$ 0.031= **€/year**

  where factors 0.8 and 0.031 are explained in paragraphs 5.2.4 and 5.2.7 respectively.

- Cost of back up cooling.

  The electric compression refrigeration system capital cost including the installation cost -**which has fix value**- is given by the following equation:

  (5-45) $\Rightarrow$Electric Compression Refrigeration System Capital Cost + Installation Cost = [kW] $*$ [€/MW] = **€**

  where **k**W is the maximum value of cells of column 2 (*Table 5.15*) and €/kW is the corresponding price (paragraph 5.2.4)

  The operation cost of electric compression refrigeration system is calculated as follows:

  Operation cooling energy: COP = 4.5 $\Rightarrow$ (Eq. 4-3) $\Rightarrow$ $W_{e,\,c}$= {kWh $*$ 30 **-** kW $*$ [1/(z+1)] $*$ η_{HE} $*$ 24 $*$ 30} **/** 4.5 = kWh

  where $W_{e,\,c}$ is the electric energy supplied from the local grid for cooling and kWh corresponds to the December-cell of column 2 (*Table 5.16*), kW is the capability of GT heat power production and [1/(z+1)] is the portion of GT exhaust heat, going for cooling.

  Operation cost in December = $W_{e,\,c}$ $*$ [€/kWh] = €/month

  where €/MW is the corresponding price (paragraph 5.2.9).

  $$(5\text{-}46) \Rightarrow \text{Operation cost} = \sum_{1}^{12} (\text{€/month}) = \textbf{€/year}$$

  which varies accordingly to the inflation rate.

The maintenance cost -**which has fix value, for every year-** is given by the following equation:

(5-47) $\Rightarrow$ Maint. Cost=([Capital Cost]+[Installation Cost])$*$0.90$*$0.03**=€/year**

where factors 0.90 and 0.03 are discussed in paragraphs 5.2.4, 5.2.7. We assume that the maintenance of the cooling system is taking place in December (last week)

- The NG mass flow per month is given by

  (5-48) $\Rightarrow$NG mass flow per month = $\dot{m}_f * 60 * 60 * 24 * 30 = \dot{m}_{fm}$ kgr/month

  where first factor 60 is for the conversion of seconds to minutes, second factor 60 is for conversion of minutes to hours, factor 24 is for the conversion of hours to days, while factor 30 corresponds to the number of days of the month, except from December which is assumed to operate 23 days, due to shut down for annual service Thus, the NG mass flow per year is

  $$(5\text{-}49) \Rightarrow \text{NG mass flow per year} = \dot{m}_{fy} = \sum_{1}^{12} \dot{m}_{fm} \text{ kgr/year}$$

The cost of NG per year **-which varies accordingly to the international oil prices-** is given by the equation:

(5-93) $\Rightarrow$Cost of NG per year = $\dot{m}_{fy} * 1.11 * 48.6 * 0.2778 * (2.39t+15.61)$ = **€/year**

where factors 1.11 and 48.6MJ/kgr is accordingly to paragraph 5.2.7, while the factor 0.2778 is due to transformation from MJ to kWh. Finally the factor (2.39t+15.61) is equation (5-4).

- The heating power of the GT is proved to produce not enough heat to cover the heating demand in every month. (Cost of boilers using light heating oil <from Table 5.4: ρ =0.86kgr/lt, FCV=42.5MJ/kgr>)

The capital cost **-which has fix value-** of boilers**,** is given by

$$\text{Capital Cost of boilers} = \{kW_a - kW_b * [z/(z+1]\} * €/kW = € \qquad (5\text{-}95)$$

where kW_a are the cells of column 4 plus 5 (*Table 5.15*), kW_b is the capability of GT heat power production, [z/(z+1)] is the portion of GT exhaust heat, going for heating and €/MW is the price of boiler per MW (see paragraph 5.2.3)

The installation cost **-which has fix value-** is given by

(5-26) $\Rightarrow$ Installation cost = Capital Cost $*$ 0.10 **= €**

where factor 0.10 is explained in paragraph 5.2.3

Finally, the maintenance cost **-which has fix value, for every year-** is given accordingly to paragraph 5.2.7:

(5-31) $\Rightarrow$ Maintenance Cost = Capital cost $*$ 0.02 **= €/year**

We assume that the maintenance of the heating system is taking place in December

Energy provided from the light heating oil is calculated as follow:

$\eta_{th,\,b} = 0.8 \Rightarrow$ <Eq. (4-3)> $\Rightarrow Q_{f,\,b} = \{(kWh * 30) + (kW * 24 * 30)\}/\ \eta_{HE}$ = kWh/m per month.

where $\eta_{th,\,b}$ is given in paragraph 5.2.3, kWh are the corresponding values of cells of column 4 plus 5 (*Table 5.16)*, kW is the GT production capability of heat energy and 30 is assumed the number of days of the month (23 for December).

The mass of light diesel per month and per year are calculated as follow:

$$(5\text{-}28) \Rightarrow m_{fm,b} = [(Q_{f,\,b} * 3{,}600) / FCV] = \text{kgr/month}$$

$$(5\text{-}29) \Rightarrow m_{fy,b} = \sum_{1}^{12} m_{fm,b} = \text{kgr/year}$$

where FCV is taken from *Table 5.4*.

And so the cost of light diesel per year **-which varies accordingly to the international oil prices,** can be estimated:

(5-79) $\Rightarrow$ Cost of light diesel = $((\dot{m}_{fy,b} / 0.86) * (2.9t+42.1)$ **/** $159) * 1.8$ = **€/year**

where 0.86 is the density ρ of the light diesel (paragraph 5.), (2.9t+42.1) is equation (5-1), while the factor 159 is the capacity of barrel (paragraph 5.2.7) and 1.8 is due to relatively large consumption (paragraph 5.2.7).

- Cost of electricity for the months April – October. **(varies accordingly to the inflation rate) =** $\sum_{1}^{12}$ [kWh **-** kW$*$ 24 **X** 30] **X** €/kWh**= €/year** (5-96)

where kWh are cells of column 3, *Table 5.16,* kW is the electric power produced every month from GT, 30 is the number of days of the month -December 23-.

- Connection to the grid cost = kW$*$ €/kW$_e$ = **€ (fix)**
  where kW is the maximum difference between the monthly GT power production and the corresponding cell of column 3, *Table 5.15*, because that is the maximum power difference that might be sold to the local grid and €/kW$_e$ is the corresponding price (paragraph 5.2.6).

- Greek government is offering financial support -**which has fix value**- is given by the following equation:
  (5-57)$\Rightarrow$Greek Government Financial Support = 0.4 $*$ (GT package cost + GT package installation cost + Heat exchanger cost + District heating installation cost + Absorption chiller cost + Compression refrigeration system cost + Capital cost of boilers + Installation cost of boilers) = **€**
  where factor 0.4 is explained in paragraph 5.2.10

- CO_2 emissions estimation and penalty
  The energy per year supplied from national grid (PPC) is calculated as follow:

$$\text{Energy per year} = \sum_{1}^{12} [\text{ kWh - kW} * 24 * 30] = \text{kWh/year} \qquad (5\text{-}97)$$

  where kWh are the cells of column 3 (*Table 5*.16), kW electric power produced every month from GT and 30 is assumed the number of days of the month.

  The hotel is connected with the interconnected national grid (continental) of Greece. Taking into account the data from *Table A.5*, it can be estimated how many kWh/year are produced from the available kinds of power plants, assuming that analogical distribution of the total electric energy per year:
  Lignite: kWh/year $*$ 0.674 = kWh/year
  Natural gas: kWh/year $*$ 0.168 = kWh/year
  Diesel (heavy heating oil): kWh/year $*$ 0.056 = kWh/year
  Using the CO_2 calculation method presented in paragraph 5.2.8, the m_{CO2} produced from every type of power plant respectively, can be estimated:
  (5-6) $\Rightarrow$ (5-7), (5-8) (*Table 5.7*) $\Rightarrow$ $\mathbf{m_{CO2}}$ **= kgrCO$_2$/year** (Assuming η_{LIG}=0.3)
  Assuming that lignite power plants exceed the CO_2 emission limit at about 10% then the mass of CO_2, which must be accounted for penalty, will be:
  $\mathbf{m_{CO2,Li,pen}}$ **= kgrCO$_2$/year**
  Similarly (5-6) $\Rightarrow$ (5-7), (5-8) (*Table 5.7*) $\Rightarrow$ $\mathbf{m_{CO2}}$ **= kgrCO$_2$/year** (Assuming η_{NG}=0.55)
  Assuming that NG power plants exceed the CO_2 emission limit at about 0% then the mass of CO_2, which must be accounted for penalty, will be:
  $\mathbf{m_{CO2,NG,pen}}$ **= kgrCO$_2$/year**
  Finally, (5-6) $\Rightarrow$ (5-7), (5-8) (*Table 5.7*) $\Rightarrow$ $\mathbf{m_{CO2}}$ **= kgrCO$_2$/y** (η_{DIE}=0.36, APPENDIX B.3)
  Assuming that diesel (heavy) power plants exceed the CO_2 emission limit at about 7% then the mass of CO_2, which must be accounted for penalty, will be:
  $\mathbf{m_{CO2,Di,pen}}$ **= kgrCO$_2$/year**
  Thus, **CO_2 emission cost** paid by PPC = ($m_{CO2,Li,pen}$ + $m_{CO2,NG,pen}$ + $m_{CO2,DiH,pen}$) $*$ €/1,000kgr <paragraph 5.2.8> = **€/year (varies with the CO_2 penalty price)**

<u>1-Shaft GT HE</u>

The only difference is concerning the GT package cost, (+30%).

***Scenario 4:* Covering the thermal and cooling demand GT,**

The useful thermal and cooling output of the GT, is equal to the 100% demand of thermal and cooling load, at any instant of time. If the generated electricity is higher than the load, surplus electricity is sold to the grid; if it is lower, supplementary electricity is produced by the conventional power plant. The block diagram of the technical configuration is showed in *Fig. 5.14*.

In this mode the following points (key points) must be taken into consideration:

1. selection of the GT power (choosing the TET, R_c from diagrams *Fig. 3.2, 3.4, 3.6* and regulating the mass flow of the engine at the DP performance) in such way to cover the heating and cooling power and energy demand of the most demand month. The monthly operation is characterised by constant TET (the same with the DP), while the ambient conditions P_a T_a are vary according to the conditions referred in *Table 2.8* (OD performance)

2., 3. 4. and 5. key points are same as scenario 2

Having the above in mind and observing the numbers of the *Tables 5.15* and *5.16*, it can be seen that in the case of the Island the scenario 4 is actually the same with scenario 1. This is because following the criteria of the scenario 4, when covering the heating and cooling demand results in exactly covering of the lighting demand.

5.7 Economic evaluation

5.7.1 The Airport Case

The basic characteristics of the different operational modes and the economic results of the economic simulation are presented in *Tables 5.17* and *5.18* respectively.

Table 5.17: Summary of the GT design point basic characteristics of the different operational modes of the Airport case

MODE	1-shaft simple cycle	2-shaft simple cycle	1-shaft cycle with HE
Scenario 1			
TET (K)	1,300	1,300	1,300
R_C	20	25	16.3
$\dot{m}$ (kgr/sec)	36.7	42.7	34.8
UW_{DP} (MW)	9.1	10.0	8.9
η_{DP} (%)	33.04	33.57	33.02
$\eta_{TOTAL,AVER}$ (%) (1)	70,5	70,13	69.05
Scenario 2			
TET (K)	1,300	1,300	1,300
R_C	20	25	16.3
$\dot{m}$ (kgr/sec)	36.7	42.7	34.8
UW_{DP} (MW)	9.1	10.0	8.9
η_{DP} (%)	33.04	33.57	33.02
$\eta_{TOTAL,AVER}$ (%)	69.01	69.35	76.49
Scenario 3			
TET (K)	1,300	1,300	1,300
R_C	20.0	25	16.3
$\dot{m}$ (kgr/sec)	7.9	8.35	7.8
UW (MW)	2	2	2
η_{DP} (%)	33.04	33.57	33.02
$\eta_{TOTAL,AVER}$ (%)	70.5	70.13	69.05
Scenario 4			
TET (K)	-	-	-
R_C	-	-	-
$\dot{m}$ (kgr/sec)	-	-	-
UW_{DP} (MW)	-	-	-
η_{DP} (%)	-	-	-
$\eta_{TOTAL,AVER}$ (%)	-	-	-

(1) Shows the net indicative average coefficient of performance of the 12 months of the year, taking into account only the performance in producing power, heat and cooling. The profits from selling the excess of electricity to the local grid and from reducing the CO_2 emissions are not included.

Table 5.18: Summary of the economic evaluation results of the different operational modes for the Airport case

	Net Present Value (NPV) x 10^3 €			
MODE		**1-shaft simple cycle**	**2-shaft simple cycle**	**1-shaft cycle with HE**
Conventional	-73,770	-	-	-
Scenario 1	-	-64,249	-57,242	-65,684
Scenario 2	-	-68,054	-59,688 (1)	-72,494
Scenario 3	-	-63,248	-61,384	-64,638
Scenario 4 (2)	-	-	-	-

(1) This value comes from a case where the assumptions of scenario 2 are not fulfilled totally. Actually, the TET is not reduced as low as it should, due to the program code restriction of operating with choked turbines at any time.
(2) The reason that there is no data for the scenario 4 is explained in paragraph 5.4.2.

Fig. 5.15 shows the cost distribution of the conventional case. Similarly, *Figs 5.16-5.24* show the cost distribution of the hypothetical modes.

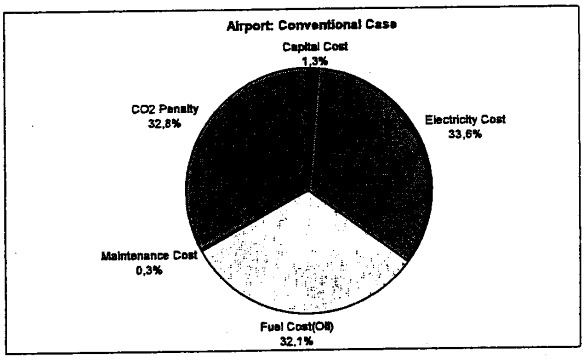


Fig. 5.15: Airport: Conventional Case

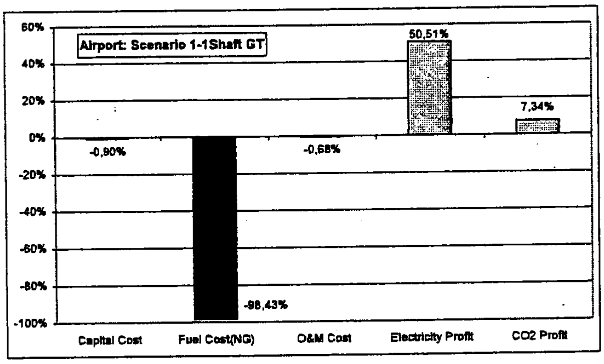


Fig. 5.16: Airport: Scenario 1-1 Shaft GT

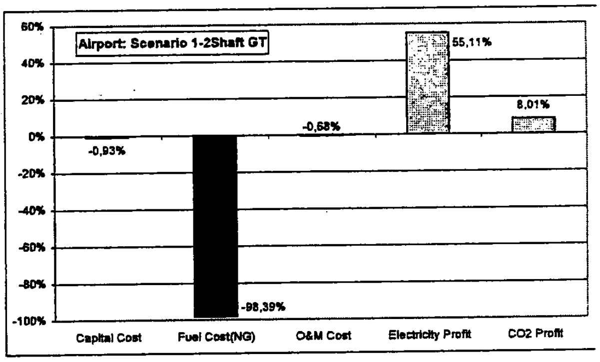


Fig. 5.17: Airport: Scenario 1-2 Shaft GT

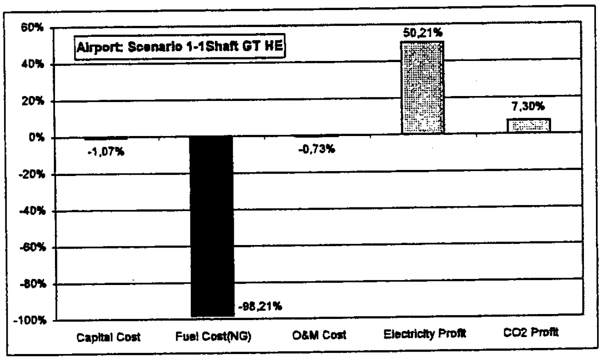


Fig. 5.18: Airport: Scenario 1-1 Shaft GT HE

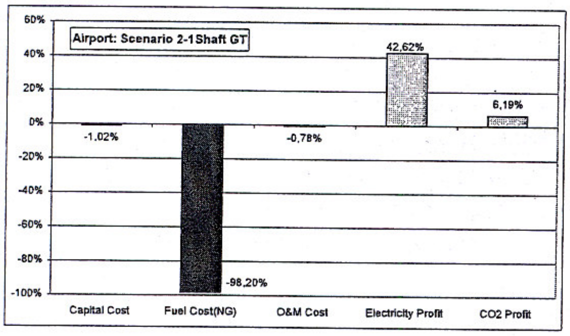


Fig. 5.19: Airport: Scenario 2-1 Shaft GT

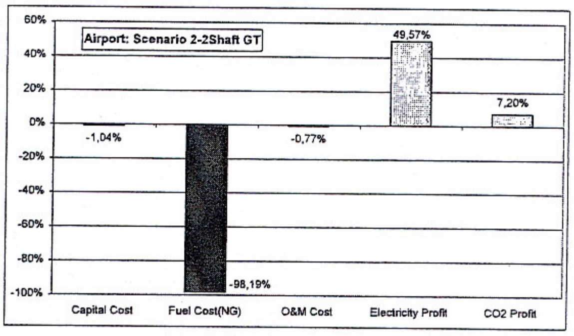


Fig. 5.20: Airport: Scenario 2-2 Shaft GT

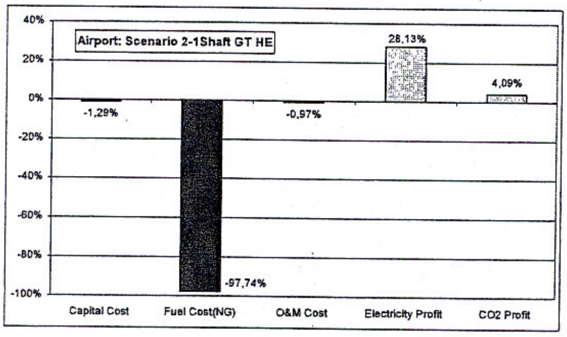


Fig. 5.21: Airport: Scenario 2-1 Shaft GT HE

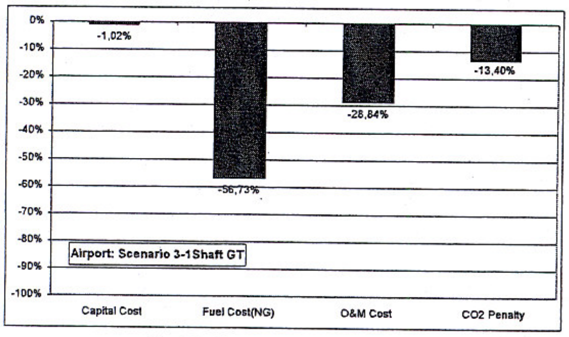


Fig. 5.22: Airport: Scenario 3-1 Shaft GT

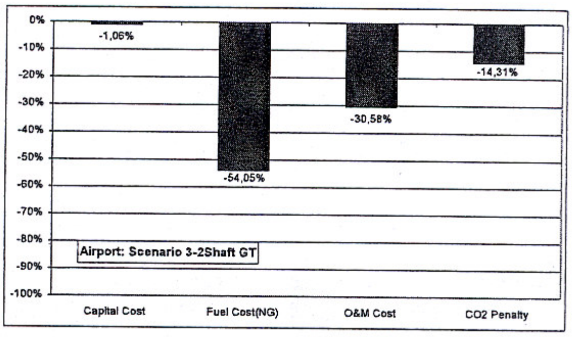


Fig. 5.23: Airport: Scenario 3-2 Shaft GT

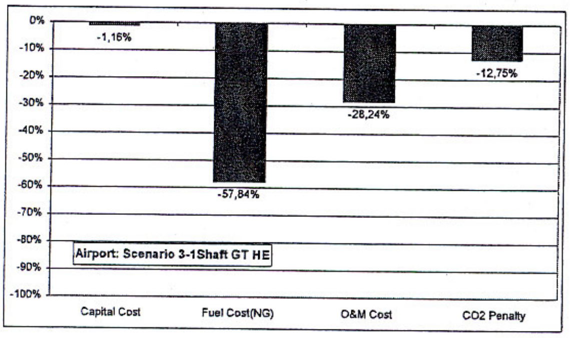


Fig. 5.24: Airport: Scenario 3-1 Shaft GT HE

Notice that in some of these figures, there are positive percentages. They show the relative individual profit (electricity or CO_2 profit) in percentage, which we gain in accordance with the sum of the negative costs (capital+fuel+O&M).

The economic evaluation is based on a twenty-year hypothetical period of operation of the CHCP plant.

In the conventional case it is observed that the main costs are the electricity, fuel and CO_2 penalty. Their contribution to the total cost is almost equal and approximately 32%. As is made obvious by the following sensitivity analysis (*Table 5.19*) the oil price, the electricity price and the CO_2 penalty variations almost equally affect the operation cost of the conventional case.

The aim of this thesis is to optimise a CHCP system. In order to achieve this, different cases of a CHCP system were studied. The **best** case was revealed to be **Scenario 1 which uses a 2-shaft simple cycle**, with an overall economic **savings of 22.5%**.

<u>Sensitivity analysis of the best case</u>
The sensitivity analysis (the results are shown in *Table 5.19*), is carried out independently based on the following basic assumptions:

1. Match the national electricity price to the EU average price (paragraph 5.2.9) and relatively equal matching of the electricity-selling price of the excess of electricity to the local grid.
2. 30% increase of the oil price and NG price (paragraph 5.2.9).
3. 30% decrease of the oil price and NG price (paragraph 5.2.9).
4. Double the CO_2 penalty price (paragraph 5.2.9).
5. Simultaneous stand of the 1, 2 and 4 assumptions.
6. Simultaneous stand of the 1, 3 and 4 assumptions

Table 5.19: Sensitivity analysis results of the best airport case

Assumption	**1**	**2**	**3**	**4**	**5**	**6**
Conventional. NPV (x 10^3 €)	-78,586	-81,165	-66,374	-99,016	-111,228	-96,437
Scenario 1 2-shaft simple cycle	-41,047	-101,331	-13,152	-45,319	-73,214	14,965

The economic analysis leads to some important conclusions:

- The results shown that all the investment options were profitable compared with the conventional case. (*Table 5.18*)
- It is observed that in all three scenarios, the profitable order of engine configurations (namely, 1-shaft simple cycle, 2 shafts simple cycle and 1-Shaft simple cycle with HE) is sustained. This can be explained by the superiority of the 2-shaft engine in terms of off design performance and especially by the increased UW_{od}, and Q_{outod} (resulting from the increased exhaust mass flow, which outweighs the slight increase of EGT of 1-shaft GT). As far the 1-shaft GT with HE concerned the poor Q_{outod} is the dominant reason for the less efficient overall operation of the trigeneration plant
- The cases of scenario 3 are not so competitive, mainly due to the lack of profit coming from the exports of electricity to the local grid. The electricity price at

which the state buys the excess of electricity seems to be satisfactory. So, in the cases of scenario 2 and especially of scenario 2 with configuration of 2-shaft GT more revenues can be obtained from selling the surplus electric energy to the grid.

- The NG price is the dominant cost-effective factor. The price of NG is generally follows the fluctuations of crude oil. Thus, these variations similarly affect the NPV of conventional and the other cases.
- CO_2 penalty is critical for the conventional case, while for the other cases it is relatively low.
- Capital cost and O&M in all cases seem to be very low. Thus, it cannot affect the final decision.

5.7.2 Rhodes Island case

The basic characteristics of the different operational modes and the economic results of the economic simulation are presented in *Tables 5.20* and *5.21*.

Table 5.20: Summary of the GT design point basic characteristics of the different operational modes for Rhodes Island

MODE	1-Shaft simple cycle	2-Shaft simple cycle	1-Shaft cycle with HE
Scenario 1			
TET (K)	-	-	-
R_C	-	-	-
$\dot{m}$ (kgr/sec)	-	-	-
UW_{DP} (MW)	-	-	-
η_{DP} (%)	-	-	-
$\eta_{TOTAL,AVER}$ (%) (1)	-	-	-
Scenario 2			
TET (K)	1,300	1,300	1,300
R_C	20	25	16.3
$\dot{m}$ (kgr/sec)	139.6	157.5	$2 \cdot 131,4$
UW_{DP} (MW)	$2 \cdot 34.7$	$2 \cdot 37$	$2 \cdot 33.9$
η_{DP} (%)	33.04	33.57	33.02
$\eta_{TOTAL,AVER}$ (%)	71.88	72.76	75.94
Scenario 3			
TET (K)	1,300	1,300	1,300
R_C	20	25	16.3
$\dot{m}$ (kgr/sec)	$2 \cdot 78,5$	$2 \cdot 82,7$	$2 \cdot 73$
UW_{DP} (MW)	$2 \cdot 19,5$	$2 \cdot 19,4$	$2 \cdot 18,7$
η_{DP} (%)	33.04	33.57	33.02
$\eta_{TOTAL,AVER}$ (%)	73.69	73.25	72.49
Scenario 4			
TET (K)	-	-	-
R_C	-	-	-
$\dot{m}$ (kgr/sec)	-	-	-
UW_{DP} (MW)	-	-	-
η_{DP} (%)	-	-	-
$\eta_{TOTAL,AVER}$ (%)	-	-	-

(1) Shows the net indicative average coefficient of performance of the 12 months of the year, taking into account only the performance in producing power, heat and cooling. The profits from selling the excess of electricity to the local grid and from reducing the CO_2 emissions are not included.

Table 5.21: Summary of the economic evaluation of the different operational modes for Rhodes Island case

	Net Present Value (NPV) x 10^3 €			
MODE		**1-Shaft simple cycle**	**2-Shaft simple cycle**	**1-Shaft cycle with HE**
Conventional	-1,078,135	-	-	-
Scenario 1 (2)	-	-	-	-
Scenario 2	-	-545,373	-570,428 (1)	-638,481
Scenario 3	-	-593.621	-584,131	-602,008
Scenario 4 (2)	-	-	-	-

(1) This value comes from a case where the assumptions of scenario 2 are not fulfilled totally. Actually, the TET is not reduced as low as it should, due to the program code restriction of operating with choked turbines at any time.
(2) The reason that there is no data for the scenario 1 and 4 is explained in paragraph 5.5.2.

Fig. 5.25 shows the cost distribution of the conventional case. Similarly, *Figs 5.26-5.31* show the cost distribution of the hypothetical modes. Notice that in these figures, there are no positive percentages, due to the autonomous local grid of the island.

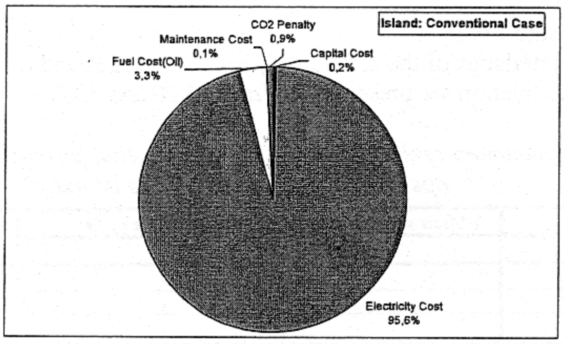


Fig. 5.25: Island: Conventional Case

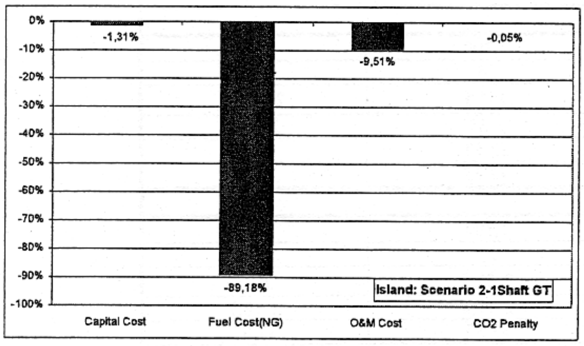


Fig. 5.26: Island: Scenario 2-1 Shaft GT

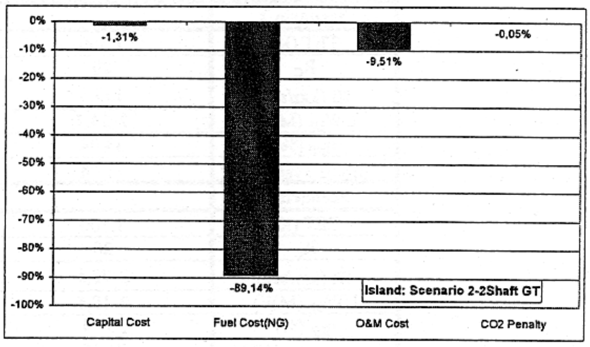


Fig. 5.27: Island: Scenario 2-2 Shaft GT

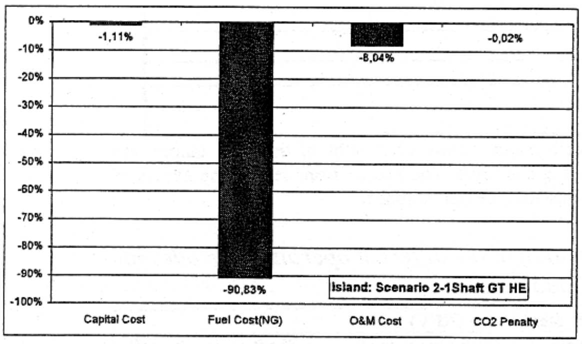


Fig. 5.28: Island: Scenario 2-1 Shaft GT HE

T

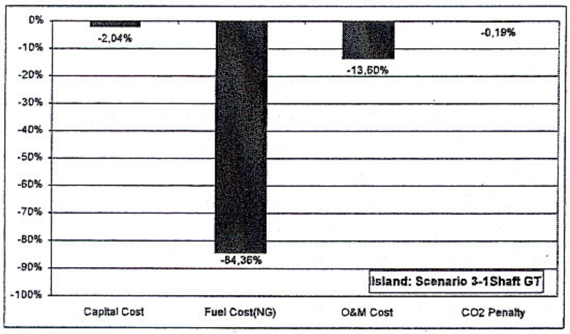


Fig. 5.29: Island: Scenario 3-1 Shaft GT

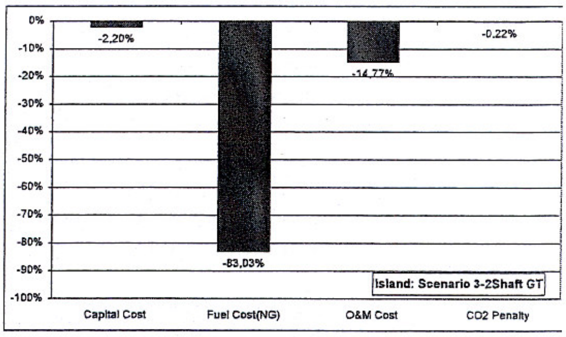


Fig. 5.30: Island: Scenario 3-2 Shaft GT

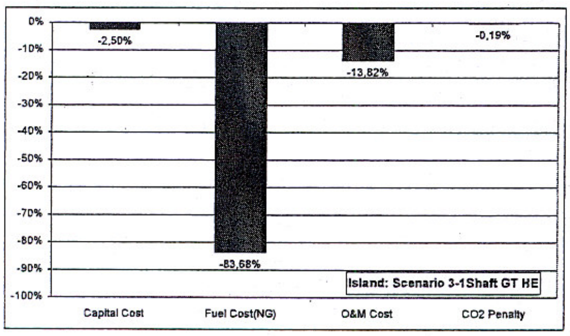


Fig. 5.31: Island: Scenario 3-1 Shaft GT HE

The economic evaluation is based on a twenty-year hypothetical period of operation of the CHCP plant.

In the conventional case it is observed that the dominant cost is the fuel cost. As it will be obvious and from the following sensitivity analysis (*Table 5.22*) the electricity price variations affect almost exclusively the operation cost of the conventional case.

The aim of this thesis is to optimise a CHCP system. In order to achieve this, different cases of a CHCP system were studied. The **best** case was revealed to be **Scenario 2 using 1-shaft simple cycle**, with an overall economic **savings of 47%**.

Sensitivity analysis of the best case

The sensitivity analysis (the results are shown in *Table 5.22*), is carried out independently based on the following basic assumptions:

1. Match the national electricity price to the EU average price (paragraph 5.2.9)
2. 30% increase of the oil price and NG price (paragraph 5.2.9).
3. 30% decrease of the oil price and NG price (paragraph 5.2.9).
4. Double the CO_2 penalty price (paragraph 5.2.9).
5. Simultaneous stand of the 1, 2 and 4 assumptions.
6. Simultaneous stand of the 1, 3 and 4 assumptions

Table 5.22: Sensitivity analysis results of the best island case

Assumption	**1**	**2**	**3**	**4**	**5**	**6**
Conventional. NPV (x 10^3 €)	-1,176,811	-1,099,987	-1,056,282	-1,090,076	-1,210,605	-1,166,899
Scenario 2 1-shaft simple cycle	-550,170	-689,657	-401,090	-545.605	-694,686	-406,119

The economic analysis leads to some important conclusions:

- The results show that all the investment options were profitable compared with the conventional case. (*Table 5.18*)
- Due to the restriction of operating with the turbine unchoked uncertainty results concerning which configuration is the best of scenario 2. As has been said in the airport case (paragrph 5.7.1), we would expect the 2-shaft case to be more profitable. By comparing the NPVs of scenario 3, we would expect the corresponding values of the two configurations of scenario 2 to be very close.
- The cases of scenario 3 are not so competitive, mainly due to the lack of profit coming from the exports of electricity to the local grid.
- The fuel price is the dominant cost-effective factor. The price of NG is generally follows the fluctuations of crude oil. Thus, these variations similarly affect the NPV of conventional and the other cases.
- CO_2 penalty, capital cost and O&M are not critical either for the conventional case, or for the rest of the cases

5.7.3 The Hotel case

The basic characteristics of the different operational modes and the economic results of the economic simulation are presented in *Tables 5.23* and *5.24*.

Table 5.23: Summary of the GT design point basic characteristics of the different operational modes for the case of the Hotel

MODE	1-shaft simple cycle	2-shaft simple cycle	1-shaft cycle with HE
Scenario 1			
TET (K)	1,300	1,300	1,300
R_C	20	25	16.3
$\dot{m}$ (kgr/sec)	2.47	2.77	2.33
UW_{DP} (MW)	0.6	0.7	0.6
η_{DP} (%)	33.04	33.57	33.02
$\eta_{TOTAL,AVER}$ (%) (1)	71.03	70.68	69.69
Scenario 2			
TET (K)	1,300	1,300	1,300
R_C	20	25	16.3
$\dot{m}$ (kgr/sec)	2.47	2.77	2.33
UW_{DP} (MW)	0.6	0.7	0.6
η_{DP} (%)	33.04	33.57	33.02
$\eta_{TOTAL,AVER}$ (%)	66.83	69.78	76.97
Scenario 3			
TET (K)	1,300	1,300	1,300
R_C	20	25	16.3
$\dot{m}$ (kgr/sec)	0.25	0.26	0.24
UW_{DP} (MW)	0.1	0.1	0.1
η_{DP} (%)	33.04	33.57	33.02
$\eta_{TOTAL,AVER}$ (%)	71.03	70.68	69.69
Scenario 4			
TET (K)	-	-	-
R_C	-	-	-
$\dot{m}$ (kgr/sec)	-	-	-
UW_{DP} (MW)	-	-	-
η_{DP} (%)	-	-	-
$\eta_{TOTAL,AVER}$ (%)	-	-	-

(1) Showsthe net indicative average coefficient of performance of the 12 months of the year, taking into account only the performance in producing power, heat and cooling. The profits from selling the excess of electricity to the local grid and from reducing the CO_2 emissions are not included.

Table 5.24: Summary of the economic evaluation of the different operational modes for the case of the Hotel

	Net Present Value (NPV) x 10^3 €			
MODE		**1-shaft simple cycle**	**2-shaft simple cycle**	**1-shaft cycle with HE**
Conventional	-3,574	-	-	-
Scenario 1	-	-5,457	-4,905	-5,580
Scenario 2	-	-5,378	-5,079 (1)	-5,859
Scenario 3	-	-2,877	-2,828	-2,904
Scenario 4 (2)	-	-	-	-

(1) This value comes from a case where the assumptions of scenario 2 are not fulfilled totally. Actually, the TET is not reduced as low as it should, due to the program code restriction of operating with choked turbines at any time.

(2) The reason that there is no data for the scenario 4 is explained in paragraph 5.6.2.

Fig. 5.32 shows the cost distribution of the conventional case. Similarly, *Figs 5.33-5.41* show the cost distribution of the hypothetical modes.

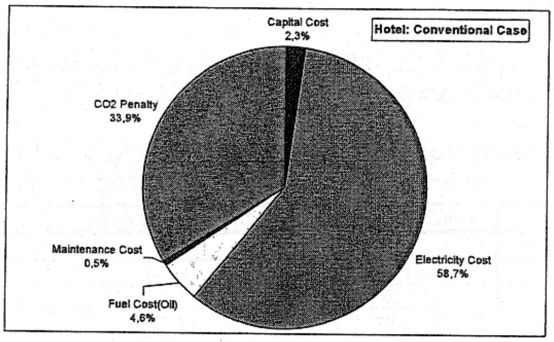


Fig. 5.32: Airport: Conventional Case

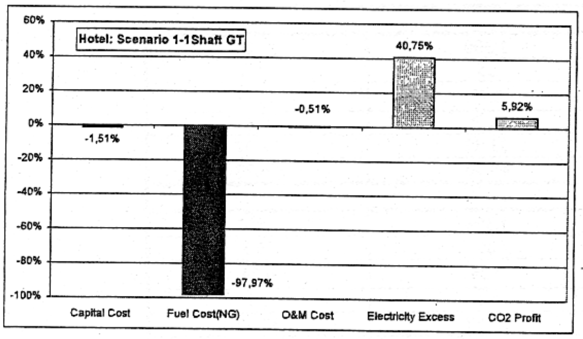


Fig. 5.33: Hotel: Scenario 1-1 Shaft GT

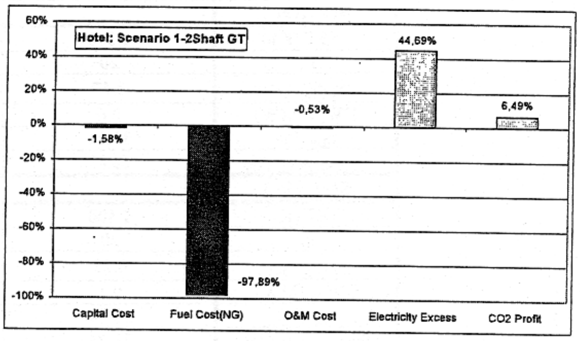


Fig. 5.34: Hotel: Scenario 1-2 Shaft GT

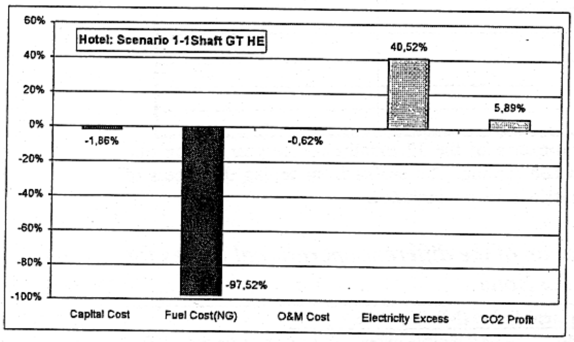


Fig. 5.35: Hotel: Scenario 1-1 Shaft GT HE

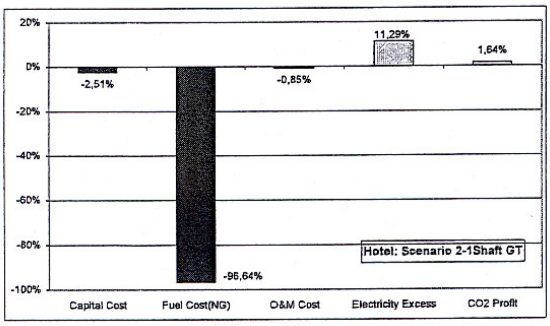


Fig. 5.36: Hotel: Scenario 2-1 Shaft GT

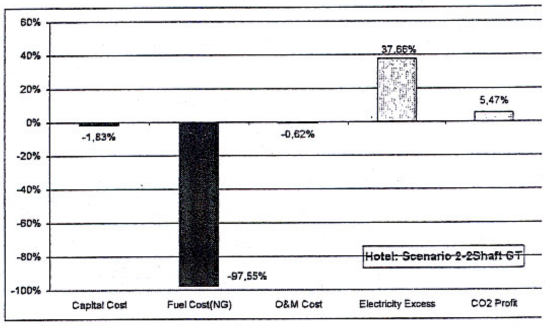


Fig. 5.37: Hotel: Scenario 2-2 Shaft GT

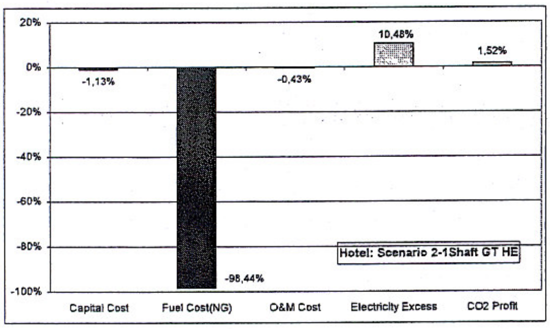


Fig. 5.38: Hotel: Scenario 2-1 Shaft GT HE

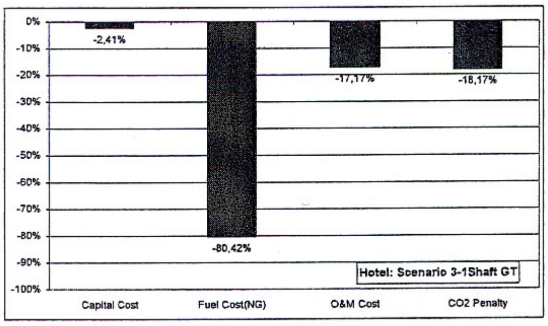


Fig. 5.39: Hotel: Scenario 3-1 Shaft GT

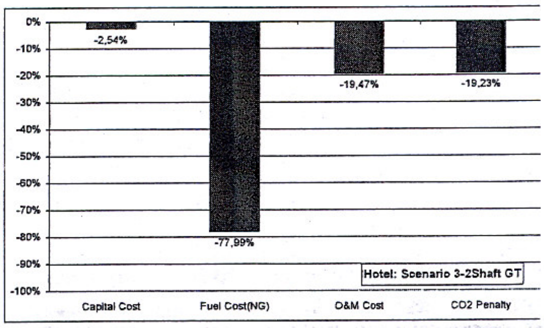


Fig. 5.40: Hotel: Scenario 3-2 Shaft GT

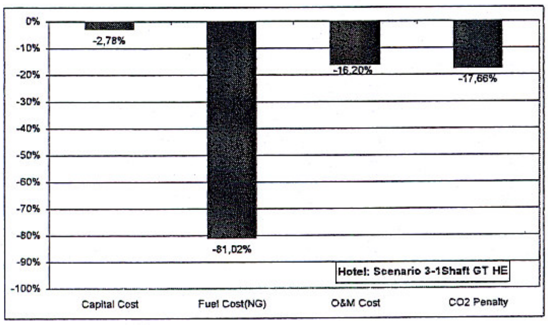


Fig. 5.41: Hotel: Scenario 3-1 Shaft GT HE

Notice that in some of these figures, there are positive percentages. They show the relative individual profit (electricity or CO_2 profit) in percentage, which we gain in accordance with the sum of the negative costs (capital+fuel+O&M).

The economic evaluation is based on a twenty-year hypothetical period of operation of the CHCP plant.

In the conventional case it is observed that the main costs are firstly the electricity (58,7%) and secondly the CO_2 penalty (33,9%). As is made obvious from the following sensitivity analysis (*Table 5.25*) the electricity price and the CO_2 penalty variations considerably affect the operation cost of the conventional case.

The aim of this thesis is to optimise a CHCP system. In order to achieve this, different cases of a CHCP system were studied. The **best** case was revealed to be **Scenario 3 using 2-shaft simple cycle**, with an overall economic **savings of 20.8%**.

<u>Sensitivity analysis of the best case</u>
The sensitivity analysis (the results are shown in *Table 5.25*), is carried out independently based on the following basic assumptions:
1. Match the national electricity price to the EU average price (paragraph 5.2.9) and relatively equal matching of the electricity-selling price of the excess of electricity to the local grid.
2. 30% increase of the oil price and NG price (paragraph 5.2.9).
3. 30% decrease of the oil price and NG price (paragraph 5.2.9).
4. Double the CO_2 penalty price (paragraph 5.2.9).
5. Simultaneous stand of the 1, 2 and 4 assumptions.
6. Simultaneous stand of the 1, 3 and 4 assumptions

Table 5.25: Sensitivity analysis results of the best hotel case

Assumption	1	2	3	4	5	6
Conventional. NPV (x 10^3 €)	-5,422	-3,357	-3,352	-4,467	-6,537	-6,532
Scenario 3 2 shafts simple cycle	-4,234	-3,165	-2,490	-3,052	-4,795	-4,120

The economic analysis leads to some important conclusions:
- The results shown that all the investment options were profitable compared with the conventional case. (*Table 5.18*)
- It is observed that in all three scenarios, the profitable order of engine configurations (namely, 1-shaft simple cycle, 2-shaft simple cycle and 1-shaft simple cycle with HE) is sustained. This can be explained by the superiority of the 2-shaft engine in terms of off design performance and especially by the increased UW_{od}, and Q_{outod} (resulting from the increased exhaust mass flow, which outweighs the slightly increased of EGT of 1-shaft GT). As far the 1-shaft GT with HE concerned the poor Q_{outod} is the dominant reason for the less efficient overall operation of the trigeneration plant
- The cases of scenario 3 are far the more competitive, mainly due to the lack of electricity excess. The electricity price at which the state buys the excess of

electricity seems to be satisfactory. But the electric energy to be export is not capable of giving enough profit, to justify the operation of a large GT.

- The electricity price is critical for all the cases and especially the conventional case as already mentioned. This is due to the large consumption of electricity for cooling for air-conditioning purposes.
- The fuel price is the dominant cost-effective factor. The price of NG is generally follows the fluctuations of crude oil. Thus, these variations of them similarly affect the NPV of conventional and the others cases.
- CO_2 penalty is critical for the conventional case, while for the rest of the cases it is relatively low.
- Capital cost and O&M in modes seem to have an appreciable effect on the overall performance.

5.7.4 General remarks of the three cases

Observing columns 2 and 3, referring to the 1-shaft and 2-shaft GTs, in *Tables 5.17*, *5.20* and *5.23* it can be seen that same of the basic characteristics (such as R_c, $\dot{m}$, UW_{DP} η_{DP}) are considerably different. To explain this difference it must underlined that these particular design point characteristics of the two GT-types have been chosen having in mind not only the best design point but also the off deign performance. As it is explained in paragraph 3.7 these two engines perform differently at off design point conditions and this is the reason for different choose of design point characteristics.

Concluding it can be seen from paragraphs 5.7.1, 5.7.2 and 5.7.3, that the suggested from this thesis tri-generation technology is more economic favourable than the conventional technology, at least when some particular scenarios are followed. (*Table 5.26*)

Table 5. 26: Overall economic evaluation results of the different operational modes for the three cases

	Net Present Value (NPV) x 10^3 € for the Airport Case			
MODE		**1-shaft simple cycle**	**2-shaft simple cycle**	**1-shaft cycle with HE**
Conventional	-73,770	-	-	-
Scenario 1	-	-64,249	-57,242	-65,684
Scenario 2	-	-68,054	-59,688	-72,494
Scenario 3	-	-63,248	-61,384	-64,638
Scenario 4	-	-	-	-
	Net Present Value (NPV) x 10^3 € for Rhodes Island Case			
MODE		**1-Shaft simple cycle**	**2-Shaft simple cycle**	**1-Shaft cycle with HE**
Conventional	-1,078,135	-	-	-
Scenario 1[(2)]	-	-	-	-
Scenario 2	-	-545,373	-570,428	-638,481
Scenario 3	-	-593.621	-584,131	-602,008
Scenario 4	-	-	-	-
	Net Present Value (NPV) x 10^3 € for the Hotel Case			
MODE		**1-shaft simple cycle**	**2-shaft simple cycle**	**1-shaft cycle with HE**
Conventional	-3,574	-	-	-
Scenario 1	-	-5,457	-4,905	-5,580
Scenario 2	-	-5,378	-5,079	-5,859
Scenario 3	-	-2,877	-2,828	-2,904
Scenario 4	-	-	-	-

6. DISCUSSION - CONCLUSIONS

6.1 Overview of thesis procedure

In this thesis an evaluation tool of specified trigeneration systems under certain simplifying assumptions have been presented. An effort has been made to solve a difficult problem: to determine which is the best trigeneration technology and system design for a particular application and which is the best operation mode at any moment in time. Emphasis have been given this determination of the best trigeneration to take into account not only the design point of the GT, but also the also the off design performance (variation of the ambient condition or the load).

Before any consideration of trigeneration, potential changes in energy requirements must be investigated. The selection of the optimum trigeneration system should be based on criteria specified by the investor and user of the system, considering economic performance, energy efficiency, uninterrupted operation or other performance measures. The problem posed in the introductory paragraph can be stated more explicitly as a set of decisions that have to be made regarding

the type of trigeneration technology (gas turbine, combined cycle etc.)

number of prime movers and nominal power of each one,

heat recovery equipment,

absorption cooling system,

need of thermal or electric storage,

interconnection with the grid (one-way, two-way, no connection at all),

operation mode of the system (i.e. operating electrical and thermal power at any moment).

Any decisions should also take into consideration legal and regulatory requirements, which may impose limits on design and operation parameters such as noise level, emission of pollutants and total operating efficiency.

The whole activity from the initial conception to the final design can be divided in three stages:

<u>Preliminary assessment</u>: An energy audit of the site is performed in order to reach a first assessment on whether or not the technical conditions are such that cogeneration could be economically viable. Aspects, which are examined, include the following:

Level and duration of electrical and thermal-cooling loads.

Energy saving measures that could be implemented before trigeneration.

Any plans for changes in processes, which would affect electrical and thermal- cooling loads.

Compatibility of thermal loads with the heat provided by available trigeneration technologies.

Availability of space for installing the trigeneration system.

Ability to interconnect with the electrical and thermal-cooling system of the facility.

Effect that cogeneration may have on the need to install and on the operation of other equipment such as boilers, emergency generator and absorption chillers.

Even though the aforementioned are referring to an existing facility, similar aspects are also examined when a new facility (either building or industry) is under design. In fact in such a case, the integration of the trigeneration system with the rest of the installation is much easier and it has greater potential for improving the economic viability. In large projects, a pre-feasibility study might be advisable for a better assessment at this stage.

Feasibility study and system selection: It is the crucial stage, which will determine whether trigeneration is viable and which is the best system for the particular application. It includes the following actions.

- Collection of data and drawing of load profiles for the various energy forms needed: electricity, heat in the form of steam at various pressure and temperature levels, heat in the form of hot water at various temperatures, cooling requirements, etc.
- Collection of information about electricity and fuel tariffs, as well as about legal and regulatory issues.
- Selection of trigeneration technology that can provide the quality of heat (medium, pressure, temperature) required. The power to heat ratio might be an additional criterion for selection but not very strict, because it can be changed either by additional equipment (e.g. augmented heat recovery, supplementary firing, thermal storage) or by a decision to cover part of the electrical or thermal load.
- Selection of the number of units and of the capacity of each unit. From the point of view of energy efficiency, the selection should be such that the cogenerated heat is used, avoiding rejection to the environment.
- Selection of the operation mode and calculation of the energy and economic measures of performance. Calculations can be repeated for various operation modes.
- Actions 3, 4 and 5 are repeated for other combinations of technology, number and capacity of units, additional equipment and operation mode.
- The system with the best performance is selected. A single- or multi-criteria approach can be followed.
- A study of the environmental, social and other effects of the selected system is performed.

In cases where there is a strong phase shift between the electrical and thermal load, it is useful to examine the technical and economic feasibility of thermal storage or (not so common) electrical storage, in order to increase the utilization of cogenerated electricity and heat.

The multitude of variations of system structure and operation mode makes an exhaustive search very difficult, if at all possible, by conventional means. Computer program has been developed by the author to aid the designer and are commercially available. They differ from each other with respect to the range of applicability and depth of analysis

Detailed design: For the system selected in Stage II a detailed study follows. There may be a need to collect more accurate and detailed information about load profiles and repeat actions 4 and 5 of Stage II at a higher depth, in order to either verify or slightly modify the main characteristics of the system. Detailed technical specifications of the main unit(s) are recorded, including not only capacity, efficiency and controls, but also emissions, noise and vibration levels. Specifications for other major components are also prepared.

6.2 Gas turbine considerations

Recently much attention has been paid to the trigeneration (CHCP) system, due to its inherent highly effective energy utilization, and various power-generating machines are used as prime movers. The gas turbine has relatively lower efficiency, while it releases large amounts of thermal energy by exhaust gas. For this reason, the gas turbine is suitable for the topping cycle application in the trigeneration and combined cycle systems. There have been many efforts to fully use the advantages of the gas turbine and make the trigeneration system compact and efficient. There are also many high-performance gas turbine engines which have been constructed with the prime purpose of application to cogeneration and combined cycle power generation systems. The major advantages of GTs are:

- High reliability which permits - long-term unattended operation.
- High grade heat available.
- Constant high speed enabling - close frequency control of electrical output.
- High power/weight ratio.
- No cooling water required.
- Relatively low investment cost per kW_e electrical output.
- Wide fuel range capability (NG, LPG, diesel, naphtha, associated gas, biomass).
- Multi fuel capability.
- Low emissions.

On the other hand there are some disadvantages, which must be taken into consideration:

- Limited number of unit sizes within the output range.
- Lower thermal efficiency than reciprocating engines.
- If gas fired, requires high-pressure supply or in-house boosters.
- High noise levels (those of high frequency can be easily alternated).
- Poor efficiency at low loading (but they can operate continuously at low loads).
- Can operate on premium fuels but need to be cleaned of dried.
- The performance of a gas turbine engine is greatly affected by the component performance and the efficiency decrease sharply at off-design conditions, especially at part load.
- May need long overhaul periods.

For power generation purposes, two fundamental engine configurations are most commonly used. These are the 1-shaft engines and the 2-shaft ones. These two engine types demonstrate different performance characteristics in terms of electricity generation production.

In the 1-shaft configuration case, the engine rotates at the same speed as the load is varied. Thus, the transient performance of such an engine in terms of electrical frequency stability (produced by, the directly connected generator) is considered to be good.

Two-shaft engines respond differently to a load variation. In this case, the requirement to increase or decrease power is satisfied by the variation of the hot gas flow to the power turbine. For instance, increasing the demanded power output in such an engine, results in increased production of hot gas at higher pressures. This is achieved by increasing the speed of the gas generator. Fuel flow is also increased in accordance with the higher speed. The whole process to be completed requires a finite time and thus the response of the two shaft engine in terms of load variation and produced electrical stability is not as good as the single shaft one.

During part load operation though, the 2-shaft engine performs better than the single shaft one in both efficiency and produced output torque due mainly to the independence of the power turbine from the gas generator. The gas generator is able to operate at higher turbine entry temperatures and rotate at higher speeds irrespective of the power turbine, which can be set to supply the demanded outcome

For power generation purposes, two fundamental engine configurations are most commonly used. These are the 1-shaft engines and the 2-shaft ones. These two engine types demonstrate different performance characteristics in terms of electricity generation production.

Also, the purpose of this study is to analyze the performance characteristics of recuperated gas turbines operating at part load conditions. Differences in part load performance, due to various factors in design and operation, have been investigated. Maintaining high turbine exhaust temperature (and thus, the recuperator inlet gas temperature) enhances the part load efficiency considerably. In particular, the variable speed operation of the single-shaft configuration provides the most efficient part load operation. As the design turbine inlet temperature increases, the relative part load efficiency - becomes higher. A higher design pressure ratio exhibits better part load efficiency characteristics

Gas turbine performance modeling forms a powerful tool, available to the gas turbine engineer in order to validate any sort of developments related to the advance or introduction of new technologies. Computer technology has made possible, the accurate and fast solution of complex equations possible. This process may be repeat for all different stages that an engine may follow. Such modeling activities are able to contribute greatly to an engine development, at many different levels of generality or detail. In the gas turbine engine concept, the importance of modeling is even greater, since the cost and availability of test rigs and engines is considered to vary greatly. Engine modeling provides the means on which certain predictions of an engine performance are made possible to be conducted, and could form the basis on which development

6.3 Conclusions

Initially, a study was carried out concerning the energy demands of different actual cases. The research includes sourcing, collecting, classification and evaluation of the available information. The main outputs stemming from this research are power, cooling and heating loads. The data were multiplied by a factor of 1.2 in order to include the worst-case situation. The case studies are chosen from different locations with different climatic and geographic characteristics. The cases covered a wide range of economic life and the resulting data specifies the energy needs which the purposed tri-generation power plant needed to cover.

The **case studies**, which were chosen, were:

6. The new International "Macedonia" Airport of Thessaloniki, Greece
7. Lemnos island
8. Rhodes island
9. Hotel in Rethimno-Crete
10. Hotel in northern Greece

The second part dealt with the prime mover (namely the Gas Turbine, GT) modelling and simulation. The technical part of the assessment included the Design Point (DP) and Off Design (OD) analysis of the GT. In other words, the performance analysis simulated different thermodynamic cycles (Simple, or with Heat Exchanger), and different configurations (one or two-shaft). The computer programming code, is capable of simulating the effects of the use of different types of fuel, ambient conditions, part load conditions, degradation, or the extraction of power for district heating or for absorption cooling.

The design point and off design computer simulation program results were also compared with the corresponding Turbomatch results, showing a satisfactory similarity -3-5%- in the results, mainly caused by the use of compressor and turbine maps in the case of Turbomatch.[132] The performance results also verify the conclusions of the GT theory ([14]-[16]).

The third part included the simulation of the absorption cooling system alone and/or in co-operation with the prime mover. The simulation was based upon the premise that the original prime mover is replaceable. In the analysis of the absorption cooling reasonable assumptions for thermodynamic and geometric conditions were employed to simulate the real absorption cooling system. The simulation showed that with the particular assumption the COP was 0.64, which is the average value that the major manufactures give for similar systems.[73], [74],[75]

Finally, an evaluation methodology of tri-generation plants, was introduced, taking into account, both technical facts and economic data -based on certain cases from Greek reality- helping the potential users to decide whether it is profitable to use such technology or not. The economic analysis included the basic economic facts such as initial cost, handling and operational cost (fuel prices, maintenance etc), using methodology based on Net Present Value (NPV).

Three case studies were evaluated, using **four** different **scenarios**:

1. Complete coverage of the electrical, thermal and cooling loads at any instant of time. The possible excess in electric power supplies the national local grid.
2. Similar to the previous scenario, but the system is always working to exactly cover all its needed power at any time. The distribution of the power demand is such (cooling power is relatively higher than heating and electric), that when the cooling power is covered, then there is an excess of electric energy to export to the local national grid.
3. The GT at the design point has the power equal to that month which has the minimum electric power between the months of the year. Thus, the needed surplus electric energy is supplied from the local national grid. This also means that if there is a lack of heating energy, which is necessary for the proper operation of an

absorption chiller system, it will be covered by conventional air conditioners. Finally, the possible lack of heating power will be covered by the use of boilers.
4. The useful thermal and cooling output of the GT, is equal to the demand of thermal and cooling load, at any instant of time. If the generated electricity is higher than the load, surplus electricity is sold to the grid; if it is lower, supplementary electricity is purchased from the local national grid.

and three **GT configurations**:
1. 1-shaft GT
2. 2-shaft GT
3. 1-shaft GT with heat exchanger

The results of this analysis were that all of the suggested modes were economically profitable despite the relatively low electricity price in Greece (due to the utilization in of cheap lignite as raw material in the power plants -65%-). Particularly the most profitable combination scenarios and configurations were:
1. Airport case. Scenario 1 using 2-shaft simple cycle, (saving of 22.5%)
2. Rhodes Island. Scenario 2 using 1-shaft simple cycle, (saving of 47%).
3. Hotel in northern Greece. Scenario 3 using 2-shaft simple cycle, (saving of 20.8%).

A sensitivity analysis was also carried out concerning: national electricity price, purchase electricity price, oil price, NG price, and CO_2 penalty price.

The overall performance of the various trigeneration plant configurations was compared and the following results have been obtained.
- No significant difference in part load thermal efficiency is observed between 1 and 2-shaft engines.
- The 2-shaft system offers larger heat recovery than a 1-shaft system at part load. In the 1-shaft system, power reduction accompanies a continuous decrease in total trigeneration efficiency.
- The superiority of the 2-shaft engine can be found in the application of trigeneration systems rather than in the gas turbine system alone.

In general, the heat-match mode results in the highest fuel utilization rate (fuel energy savings ratio) and perhaps provides the best economic performance for cogeneration in the industrial and building sectors. In the utility sector, the mode of operation depends on the total network load, the availability of power plants and the commitments of the utility with its customers regarding supply of electricity and heat.

However, applying general rules is not the most prudent approach in trigeneration. A number of factors must be considered:
- There is a variety of trigeneration systems (type of the technology, size, and configuration).
- The design of a trigeneration system can be tailored to the needs of the user;
- The design of a trigeneration system affects the possible modes of operation, and vice versa.
- The technical and economic parameters may change with the day and time during the operation of the system.

All these aspects make it necessary to reach decisions not by rules of thump only, but by systematic optimisation procedures, based on mathematical programming, for both the design and operation of the system.

For each particular site, energy and economic performance measures are calculated for various configurations of cogeneration systems (number of units, capacity of each unit, heat recovery equipment, etc.). For each configuration, the calculations can be repeated with various models of operation, as well as with various assumptions on the values of technical and economic parameters, in particular those subject to an uncertainty. Based on the results, decisions can be reached on which of the examined systems is the most appropriate for the particular application.

For the operation of trigeneration systems, in particular, microprocessor-based control systems are available. They can provide the capability to operate in a base load mode, to track either electrical or thermal loads, or to operate in an economic dispatch mode (mixed-match mode). In the latter mode, the microprocessor can be used to monitor trigeneration system performance; specifically:
the system efficiency and the amount of useful heat available,
the electrical and thermal requirements of the user, the amount of excess electricity which has to be exported to the grid, and the amount of heat that must be rejected to the environment;
the cost of purchased electricity and the value of electricity sales, as they may vary with the time of the day, the day of the week, or season.

Using the aforementioned data, the microprocessor can determine which operating mode is the most economical or even whether the unit should be shut down. Moreover, by monitoring operational parameters such as efficiency, operating hours, exhaust gas temperature, coolant water temperatures, the microprocessor can help in maintenance scheduling. If the system is unattended, the microprocessor can be linked by a telephone line with a remote monitoring center, where the computer analysis of the data may notify the skilled staff about an impending need for scheduled or unscheduled maintenance. Furthermore, as part of a data acquisition system, the microprocessor can produce reports of the systems technical and economic performance.

6.4 Trigeneration potential and future prospects

The construction and operation of cogeneration systems may affect the national economy in several ways either direct or indirect (creation of new job positions, increased production of goods and services, etc.).

The EU Council of Ministers agreed on the directive to liberalise the electricity market at the end of 1996, after six years of negotiations. The directive obliged a market opening of at least 25% of the European Electricity market by 19 February 1999. This was to be progressively increased to 28% by 2002 and 33% by 2005.

This directive to liberalise the gas market was agreed about one and a half-years later than the electricity directive. The liberalisation process has been similar: Member states had to liberalise their gas market gradually and partially. The deadline for the first step was 10 August 2000. As with the Electricity Directive, the tendency was to liberalise faster and further than what the Directive required.

As far as the environmental protection is concerned, the most important issue at the moment is climate change. The contribution of the electricity industry to greenhouse gas emissions is enormous, and it is easier to regulate than the transport and the building sector -the other two large contributors.

In December 1997, within the framework of the climate change negotiations in Kyoto, the EU committed itself to reduce its greenhouse gas emissions by 8% for the period between 2008-2010 in relation to its 1990 levels. This commitment wass then distributed with different targets among the EU member states. Cogeneration has been widely recognised -both at EU and member state level- as a technology that can make a major contribution to achieving these targets.

Very briefly, EU policy documents recognising the importance of cogeneration to achieve the climate change commitments and defining possible instruments to promote the technology at the EU level. When the EU Energy Strategy was issued in 1997, the share of electricity produced from cogeneration in the EU was about 19%. The Strategy sets a target of achieving 18% by 2010. *Fig. 6.1* shows the percentage of electricity produced from cogeneration in the EU in 1999. As already described, the share of electricity produced from cogeneration ίo the EU is around 10%. The European Commission has established a target to achieve a share of 18% by 2010. COGEN Europe estimates that this potential is in fact at least 30%. In fact, three countries have already achieved a higher share. However, in the current situation of market stagnation due to the liberalisation process and the uncertainties arising from it, even the 18% target is unlikely to be achieved. It is very important that both the EU and the member states establish clear policies and actions aimed at achieving these targets if the climate change commitments are to be met.

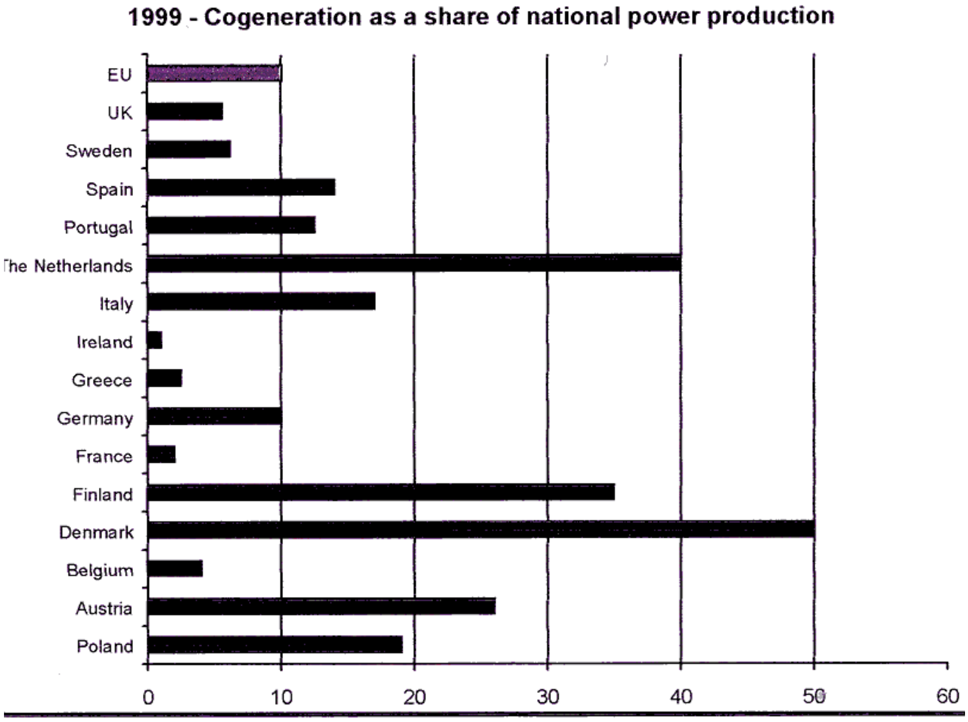


Fig. 6.1: Cogeneration as share of national power production in EU countries (1999)[80]

As already pointed out, it is universally recognised that cogeneration is one of the most important -techniques for more efficient use of fuels, savings in natural and economical resources, and protection of the environment. Attempts have been made in many countries to remove the barriers and promote cogeneration. Various incentives have been used, such as relatively high prices for excess electricity sold to the grid and grants on investments. Other measures have included dissemination of relavent information, energy auditing and analysis of data, support of research and development, etc.

Most of these measures were designed at a time when most of the barriers to the development of cogeneration derived from the existence of monopolistic electricity and gas markets. The most frequently mentioned barriers to cogeneration in the EU when the markets were not liberalised were:

- low price paid for the surplus of electricity to the grid,
- high fees for top-up and back-up supplies, .
- lack of freedom to 'wheel' (third party access) or, when allowed, too expensive to consider;
- predatory pricing against possible competition.
- technical barriers. Cogeneration schemes need to fulfill certain technical and safety requirements for proper operation. Sometimes the procedures take too long and are not transparent enough.

These barriers should be lifted in a truly liberalised market.

However, at this moment are in a transition a situation due to the liberalisation of the electricity and the gas markets. The process has major consequences both in terms of barriers to cogeneration and promotional actions. The liberalisation process is far from being completed and therefore the best word to describe the current market situation is uncertainty, which has hindering effect in terms of ι investment. Further, the first effect of electricity liberalisation in many countries has been a sharp decline in electricity prices, sometimes below production costs. This is not sustainable in the long term and so electricity prices are starting to increase again. Liberalisation should, in principle, have beneficial effects for cogeneration development, but only if environmental costs are fully included in energy prices. There is hope however, and many governments acknowledge the need to continue promoting cogeneration in a liberalised market, in recognition to its environmental benefits.

6.5 Future Work

The potential for further work in this field of study is considerable. The future work should be focus on two important guidelines: **accuracy** and **improved complexity**

Thus, future work could involve various aspects concerning:

- Collection of greater detail and accurate data from the potentials sources. (Creation of detailed energy records for the last 2-4 years would be very useful).
- The simulation work could be also improved, with fewer assumptions used (namely, accurate compressor and turbine maps, the program could also operate with the turbines unchoked, use of variable geometry, and simulation procedure for every day of the year, ect).
- Conduction of the entire plant simulation using different fuels (especially biomass)
- Simulation of a double effect absorption cooling system.
- Simulation of the engine performance, when excess of cooling power coming from the absorption cooling system is used to decrease the inlet temperature of the air entering the engine.
- Consideration of potential emissions penalties (NO_x), other than CO_2.
- Application of "shadow" price to trigeneration projects.
- Optimization of trigeneration systems, using linear programming methods.

In this thesis, the cost evaluation results concerning the trigeneration modes might underestimate. Namely, if we take into account additional emissions penalties such is the NO_x, or the shadow price, the cost performance of the trigeneration systems is going to be even better. But again this is the case of future work.

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APPENDIX *A:*
The Greek electric energy production system and market

A.1 General

Extent of Greece: 131,957km^2. Population: 10,538,086 (1999). Residents per km^2: 79.9. In the *Table A.1* a summary of energy balance of Greece is presented.

Table A.21: Greece: summary energy balance [8]

Mtoe	1985	1988	1990	1995	1996	1997	90/85	95/90	96/95	97/96	97/90
							Annual % Change				
Primary Production	7.34	8.63	9.15	9.71	10.14	9.95	4.5%	1.2%	4.5%	-1.9%	1.2%
Solids	4.84	6.29	7.08	7.91	8.20	8.07	7.9%	2.3%	3.7%	-1.6%	1.9%
Oil	1.32	1.12	0.83	0.46	0.51	0.47	-8.8%	-11.2%	12.2%	-9.4%	-7.9%
Natural gas	0.07	0.13	0.14	0.04	0.05	0.04	14.0%	-20.4%	5.5%	-3.1%	-14.8%
Nuclear	0.00	0.00	0.00	0.00	0.00	0.00	-	-	-	-	-
Hydro & Wind	0.24	0.20	0.15	0.31	0.38	0.34	-8.8%	15.0%	23.1%	-10.7%	12.0%
Geothermal	0.00	0.00	0.00	0.00	0.00	0.00	14.9%	1.3%	0.0%	-14.8%	-1.4%
Other renewable energy sources	0.86	0.88	0.95	0.98	1.00	1.02	1.9%	0.7%	1.9%	2.2%	1.1%
Net Imports	11.81	13.62	15.37	18.21	18.83	19.19	5.4%	3.4%	3.4%	1.9%	3.2%
Solids	1.23	0.86	0.99	0.92	1.17	0.76	-4.3%	-1.3%	26.2%	-34.5%	-3.6%
Oil	10.52	12.74	14.32	17.21	17.54	18.10	6.4%	3.7%	1.9%	3.2%	3.4%
Crude oil	10.54	14.39	14.71	16.95	18.32	18.40	6.9%	2.9%	8.1%	0.4%	3.2%
Oil products	-0.02	-1.65	-0.39	0.26	-0.78	-0.29	83.4%	-	-	-62.7%	-3.9%
Natural gas	0.00	0.00	0.00	0.00	0.01	0.13	-	-	-	1585.4%	-
Electricity	0.06	0.03	0.06	0.07	0.12	0.20	-0.7%	2.3%	69.4%	69.9%	18.2%
Gross Inland Consumption	18.34	20.16	22.24	24.14	25.41	25.61	3.9%	1.6%	5.3%	0.8%	2.0%
Solids	6.08	7.42	8.09	8.78	8.95	8.82	5.9%	1.7%	1.9%	-1.5%	1.2%
Oil	11.01	11.50	12.85	13.95	14.91	15.06	3.1%	1.7%	6.9%	1.0%	2.3%
Natural gas	0.07	0.13	0.14	0.04	0.05	0.17	14.0%	-20.4%	12.3%	246.9%	3.1%
Other (1)	1.17	1.11	1.17	1.36	1.50	1.56	-0.1%	3.1%	10.1%	4.2%	4.3%
Electricity Generation in TWh	27.74	33.40	34.99	41.54	42.55	43.50	4.8%	3.5%	2.4%	2.2%	3.2%
Nuclear	0.00	0.00	0.00	0.00	0.00	0.00	-	-	-	-	-
Hydro & wind (including pumping)	2.80	2.60	2.00	3.82	4.54	4.13	-6.6%	13.8%	19.0%	-9.0%	10.9%
Thermal	24.93	30.79	33.00	37.73	38.01	39.37	5.8%	2.7%	0.7%	3.6%	2.6%
Generation Capacity in GWe	7.13	8.12	8.51	8.94	9.12	9.57	3.6%	1.0%	2.0%	5.0%	1.7%
Nuclear	0.00	0.00	0.00	0.00	0.00	0.00	-	-	-	-	-
Hydro & wind	2.03	2.15	2.41	2.55	2.55	2.75	3.5%	1.1%	0.0%	8.0%	1.9%
Thermal	5.10	5.97	6.10	6.39	6.57	6.82	3.7%	0.9%	2.8%	3.8%	1.6%
Average Load Factor in %	44.4	46.9	46.9	53.0	53.2	51.9	1.1%	2.5%	0.4%	-2.6%	1.4%
Fuel Inputs for Thermal Power Generation	6.44	7.72	8.72	9.88	10.01	9.16	6.2%	2.5%	1.3%	-8.5%	0.7%
Solids	4.81	6.23	6.89	7.79	7.97	7.11	7.5%	2.5%	2.4%	-10.8%	0.5%
Oil	1.63	1.47	1.80	2.08	2.02	1.96	1.9%	2.9%	-2.6%	-2.8%	1.3%
Gas	0.00	0.02	0.03	0.01	0.02	0.09	-	-14.9%	17.9%	426.9%	15.7%
Geothermal	0.00	0.00	0.00	0.00	0.00	0.00	-	-	-	-	-
Biomass	0.00	0.00	0.00	0.00	0.00	0.00	-	-	-	-	-
Average Thermal Efficiency in %	33.3	34.3	32.5	32.8	32.7	36.9	-0.4%	0.2%	-0.6%	13.2%	1.8%
Non-Energy Uses	0.54	0.52	0.64	0.44	0.45	0.43	3.2%	-7.1%	2.5%	-4.7%	-5.4%
Total Final Energy Demand	12.52	13.72	14.54	15.82	16.88	17.25	3.0%	1.7%	6.7%	2.2%	2.5%
Solids	1.28	1.20	1.07	1.08	1.08	0.96	-3.5%	0.3%	-0.6%	-10.6%	-1.5%
Oil	8.29	9.29	10.05	10.80	11.72	12.06	3.9%	1.5%	8.5%	2.9%	2.6%
Gas	0.01	0.01	0.01	0.01	0.02	0.04	11.2%	-0.6%	24.0%	150.4%	17.1%
Electricity	2.05	2.31	2.45	2.93	3.06	3.15	3.6%	3.7%	4.3%	3.1%	3.7%
Heat	0.00	0.00	0.00	0.00	0.00	0.00	14.9%	1.3%	0.0%	-14.8%	-1.4%
Renewable energy sources	0.89	0.91	0.95	0.99	1.01	1.03	1.4%	0.7%	2.0%	2.2%	1.1%
CO_2 Emissions in Mt of CO_2 (2)	56.7	65.5	70.9	77.9	81.7	78.8	4.6%	1.9%	5.0%	-3.6%	1.5%
Indicators											
Population (Million)	9.93	10.04	10.16	10.45	10.48	10.51	0.5%	0.6%	0.2%	0.4%	0.5%
GDP (bil. EUR 1990)	59.5	62.8	65.3	69.4	71.1	73.3	1.9%	1.2%	2.4%	3.2%	1.7%
Gross Inl Cons./GDP (toe/1990 MEUR)	308.4	321.0	340.9	347.7	357.5	349.2	2.0%	0.4%	2.8%	-2.3%	0.3%
Gross Inl Cons./Capita (Kgoe/inhabitant)	1845.9	2009.0	2189.3	2309.1	2425.8	2435.8	3.5%	1.1%	5.1%	0.4%	1.5%
Electricity Generated/Capita (kWh/inhabitant)	2791.8	3327.6	3444.2	3973.9	4061.6	4137.6	4.3%	2.9%	2.2%	1.9%	2.7%
CO_2 Emissions/Capita (kg of CO_2/inhabitant)	5706.6	6521.0	6979.6	7450.6	7803.6	7493.6	4.1%	1.3%	4.7%	-4.0%	1.0%
Import Dependency (%)	60.7	61.3	62.1	65.8	66.0	66.8	0.4%	1.2%	0.4%	1.2%	1.1%

(1) Includes nuclear, hydro and wind, net imports of electricity, and other energy sources.
(2) Given on an indicative basis; calculated using common emission factors accross all countries in the world

The following diagram (*Fig.A.2*), presents the structure of Greek market of electric energy, as it was shaped afterwards the application of law 2773/1999 of the release of market of electric energy:

The Operational Units of Transport and Distribution are compelled to transport and to distribute respectively electric current on behalf of the Public Power Company, PPC other and the rests of independent producers and suppliers.

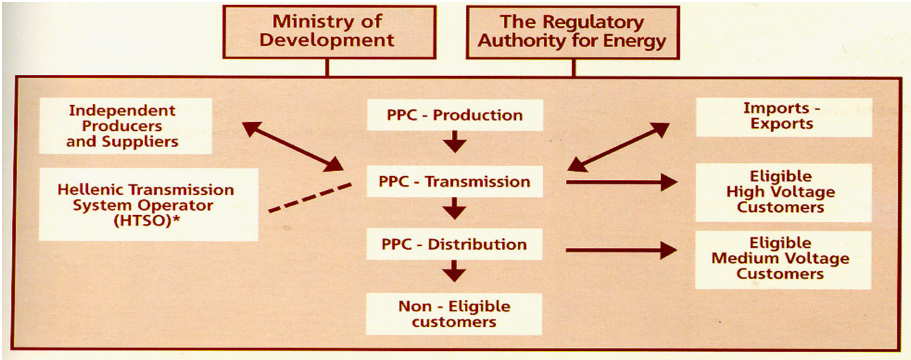


Fig.A.20: Structure of the electric energy market in Greece [2]

The PPC is the bigger company of production, the unique company that has in the property her system of transmission and distribution of electric energy and, on the present, the unique company of distribution of electric energy in Greece, which, 31 December 2002, provided electric energy in 6.7 millions customers. At the duration of year 2002 the Company produced roughly the 97% from the 50,572GWh of electric energy that were produced in Greece.

She is the bigger industrial enterprise in Greece as for the constant energetic elements. The first annual use that expired on 31 December 2002 it presented total sales of height of e 6,497 million and functional result of height of e 2,093 million (pre dumping, with base the Greek accountant models). The 31 December 2002 the Company had total installed force 11,739MW. *Table A.2* presents certain elements with regard to the functional activity of Company at the last three-year period:

Table A.22: Operation data for the years 2000, 2001, 2002 [4]

31 DECEMBER	**2000**	**2001**	**2002**
Installed Load (MW)	11,121	11,158	11,739
Net Electricity Production (GWh) (1)	48,483	48,054	48,902
Electricity sales (TWh) (2)	42.9	44.5	46.6
Number of Customers at the end of the year (in millions)	6.5	6.6	6.7

(1) The *clean production of electric energy counterbalances with the total production of electric energy minus the internal consumption of electric energy that is owed in the process of production.*
(2) Including *the sales in the mines of **PUBLIC POWER COMPANY** and in customers in the abroad.*

The Public Power Company is the main Greek energy manager. The Greece's ever-rising demand for electricity, in the year 2001, raised 4,237kWh per capita from if average of 88kWh per capita in 1950. (*Table A.3, A.4, Figs A.3-A.4*). In July 2002, moreover, peak load rose to 8,924MW - the largest increase ever recorded since the beginning of the company.

Table A.23: Increase in energy demand (2001 in comparison to 2000) [5]

Electricity consumption	+ 3.6%
Peak load	8,600 MW (July 2001 In July 2002 there was an increase by 3.8% in comparison to July 2001)
Per capita consumption	+ 2.3%
Customers	+ 105,000

Table A.24: Changes in the operation data of PPC for the coverage of the aforementioned needs (2001 in comparison to 2000) [5]

Availability of thermal stations for 2001	87%
Availability of hydroelectric stations for 2001	97%
Lignite production	+ 4.5%
Transmissions lines	+ 350km
Distribution network	+6,240km

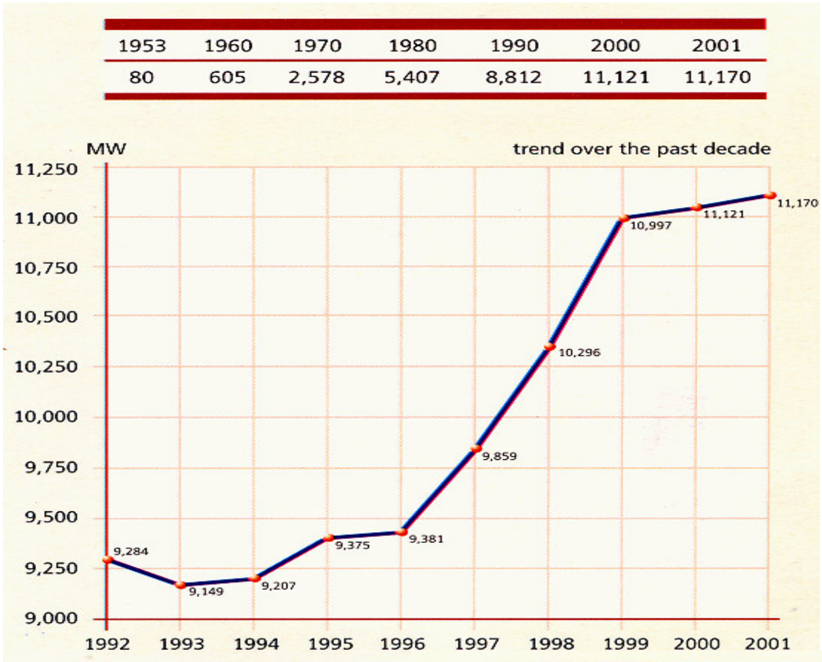


Fig.A.1:
Installed capacity (MW) [3]

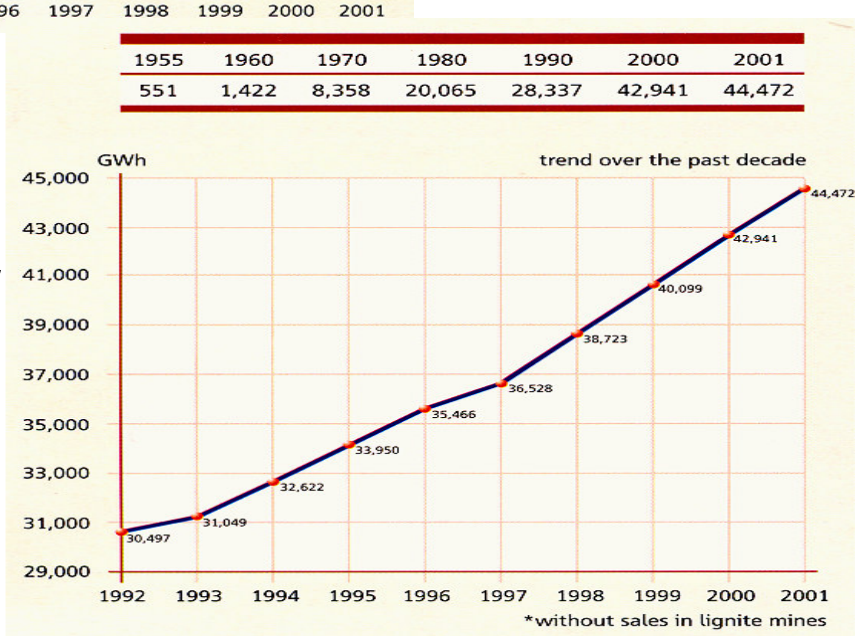


Fig.A.2:
Sales of electricity (GWh) [3]

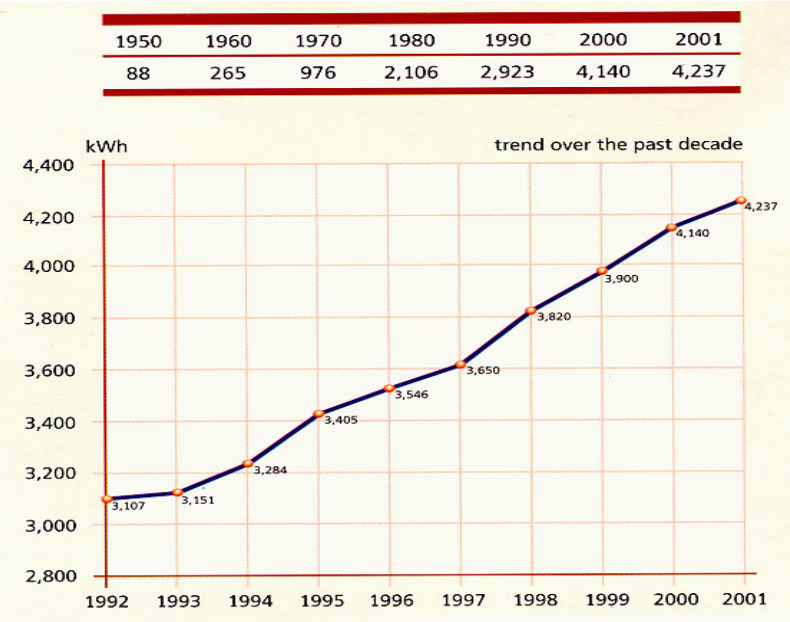


Fig.A.3: Annual percentage of increase in E.U. energy consumption, by country [3]

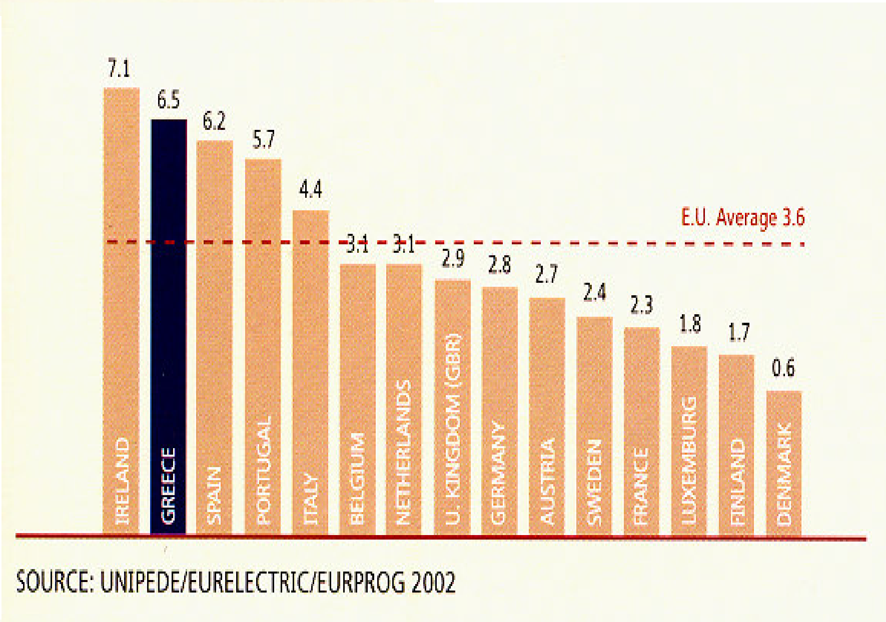


Fig.A.4: Yearly per capita consumption in kWh [4]

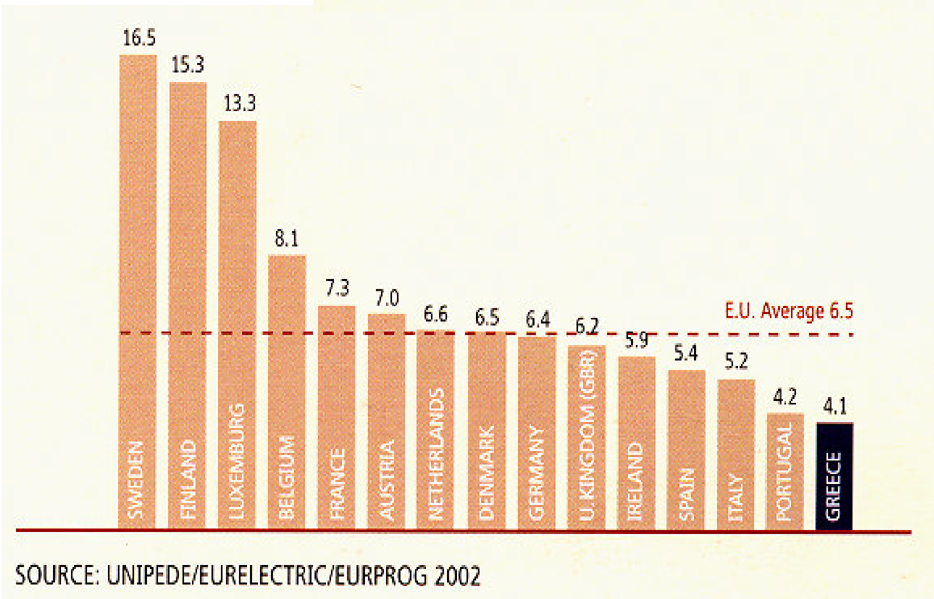


Fig.A.5: Electricity consumption per capita by country (MWh)[4]

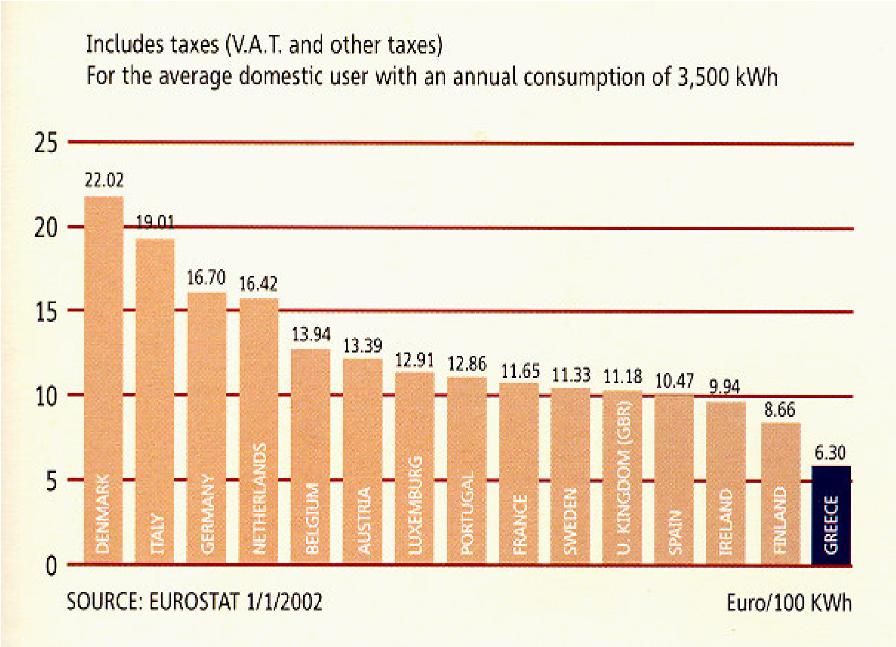


Fig.A.6: Comparison of E.U. Domestic electricity prices [4]

The demand and the production of electric current differ per regions of Greece. In continental Greece, the biggest productive power he is assembled in the northern part of country, where the majority of the lignite mines is. On the islands, the production of electric energy depends from the distance of islands from continental Greece, as well as from the possibility of connection of the islands with the continental transmission system. The islands of Ionian, as also and certain islands of Aegean, are connected with the system of transport of electric energy of continental Greece and with this system constitute the "**interconnected system**". The remainder islands are served by autonomous stations of production of electric energy, which function mainly with oil and wind energy. The islands this are reported as "**not interconnected islands**". Most stations of production in the not interconnected islands of are small size, according to the population that they serve. The stations of production of Crete and Rhodes are considered big stations.

A.2 Production of Electric Energy

On 31 December 2002, in the interconnected system and in the islands Crete and Rhodes functioned seven lignite stations of production, four petrol stations of production and two petrol units in the station of production that is found in Lavrio, a station of natural gas in Keratsini, one unit of Combined Circle of Natural Gas in Komotini (beginning of operation in 2002) and two in Lavrio, as well as 24 hydroelectric power stations

Besides, in the rests not interconnected islands are functioning 33 autonomous thermoelectric power stations in total, 21 wind parks and 5 solar (panel) stations. The total installed capacity of stations is 11.739MW. From the total of installed capacity, the 10.354MW constitute the capacity of stations connected in the interconnected system, which it supplies electric energy in continental Greece and certain near islands, connected between themselves or with the interconnected system via submarine cables. The systems of production of Crete and Rhodes have installed capacity 590MW and 206MW respectively. The total installed capacity of rests of not linked islands is 589MW. In *Fig.A.8 is* impressed the geographic distribution of stations of production.

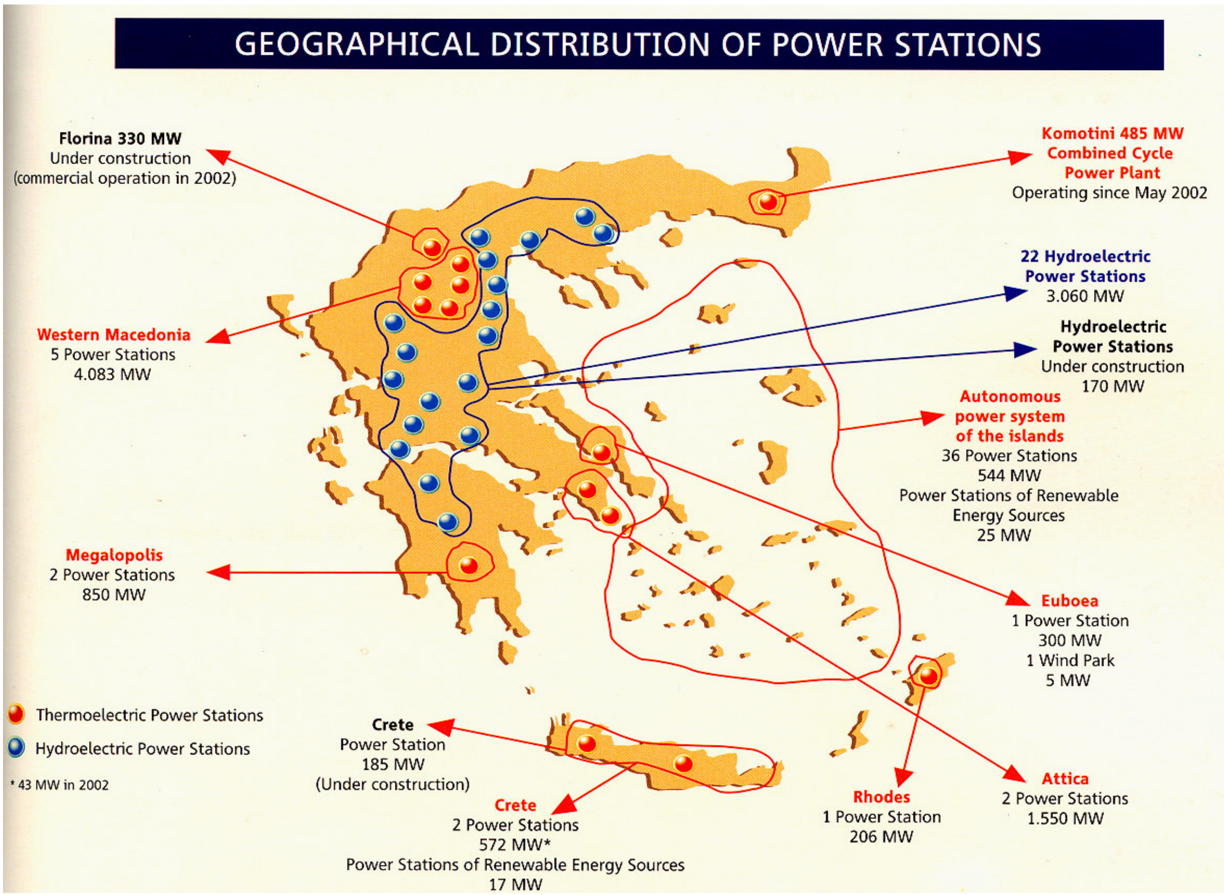


Fig.A.7: Geographical distribution of power stations [5]

In *Table A.5* are mentioned the installed capacity in MW with base the primary source of energy (use of fuel) for three-year period 2000-2002, as well as the total net production in GWh for the same period.

All the lignite stations are located close the mines of Company so that is decreased the cost of transport of **lignite**, the bigger quantity of which is transported on conveyor belts. The Company realizes that the excavation of lignite from self-belonging mines is the most important cost of this thermal source of energy production. Relative with **oil** as fuel, the company "Greek Oils S.A." or "ELPE", that is controlled by the Greek State, is on the present the unique supplier of oil, the prices of delivery of liquid fuels are format in weekly base and are based on the mean of high prices of relative petroleum products at the duration of previous week, as these they are published in the Platt's Oilgram Marketscan.

The PPC is the bigger purchaser of **natural gas** in Greece. It buys roughly the 75% of quantity of gas that traffics in the company "Public Enterprise of Gas S.A", or "DEPA" according to the convention of purchase of natural gas that was placed in force in 1994 and expires 2016. Besides, the Company uses energy produced from hydroelectric power stations in periods of peak of charge. Because the services of common utility that is compelled the PPC to provide, as for example the supplies of **water** of irrigation, certain from the hydroelectric power stations of Company they even function in periods not-peak. The hydroelectric power stations need usually lower levels of maintenance and less personnel than that the other stations of production

Table A.25: The installed and the total net power production for the years 2000, 2001, 2002 [11]

	INSTALLED CAPACITY (MW)			TOTAL NET PRODUCTION (GWH)		
31 December	**2000**	**2001**	**2002**	**2000**	**2001**	**2002**
Interconnected System						
Thermoelectrical Power Plants						
Lignite Power Plants	4.908	4.933	4.958	30.943	32.042	31.197
Oil power Plants	777	750	750	4.143	3.543	3.394
Natural Gas Power Plants	1.100	1.100	1.581	5.572	5.814	6.725
Total Thermoelectrical Power Plants	6.785	6.783	7.289	40.658	41.399	41.316
Hydroelectric Power Plants	3.060	3.060	3.060	4.055	2.666	3.381
Wind and other Renewable Power Plants	5	5	5	14	11	14
Total Interconnected System	**9.850**	**9.848**	**10.354**	**44.727**	**44.076**	**44.711**
Non-Interconnected Islands						
Thermoelectrical Power Plants						
Lignite Power Plants	-	-	-	-	-	-
Oil power Plants	1.238	1.277	1.352	3.678	3.886	4.122
Natural Gas Power Plants	-	-	-	-	-	-
Total Thermoelectrical Power Plants	1.238	1.277	1.352	3.678	3.886	4.122
Hydroelectric Power Plants	1	1	1	1	1	1
Wind and other Renewable Power Plants	32	32	32	77	91	68
Total Non-Interconnected Islands	**1.271**	**1.310**	**1.385**	**3.756**	**3.978**	**4.191**
Total Interconnected System & Total Non-Interconnected Islands						
Total Thermoelectrical Power Plants	8.023	8.060	8.641	44.336	45.283	45.438
Total Hydroelectric Power Plants	3.061	3.061	3.061	4.056	2.667	3.382
Total Wind and other Renewable Power Plants	37	37	37	91	102	82
TOTAL	**11.121**	**11.158**	**11.739**	**48.483**	**48.054**	**48.902**

Table A.26: Production percentages per fuel (2004)[10]

National Grid	Lignite	Natural gas	Water	Diesel	Renewable
Interconnected	67.4%	16.8%	10.2%	5.6%	0.03%
Non-Interconnected				98.5%	1.5%

The Company has installed 157 **wind** generators, total installed force 37MW, with annual production about 100,000MWh. Also, has installed 5 **solar** panel stations as well as crowd of individual solar panel units in small and isolated islands. Her affiliated company "PPC Renewable", has installed with other company of production of electric energy from renewable sources, two wind parks of total installed force 8.4MW, From February 2001, the PPC submitted applications in the Ministry of Development and in the RAE for issuing of authorizations of production for 25 wind parks, three **geothermal** stations and a solar station. The total power of the above it amounts in the 380MW roughly. Up to today the Company has received authorization of production for the creation of 1 wind parks of total installed power 26MW and for the growth of geothermal force 8MW, Eight from the above wind parks of total installed force 17MW as well as the growth of geothermal field, already have been included in the Operational Program of Development. Also, her have been engaged the rights of research and exploitation of three still geothermal fields with recoverable geothermal 150MW. The timetable of growth of into account of fields will depend main from the consent of local societies. (*Fig.A.9, A.10*)

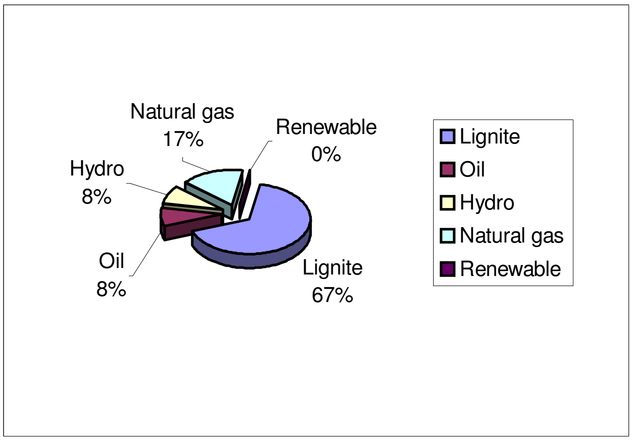


Fig.A.8: Production percentage in the interconnected system per type of fuel [9]

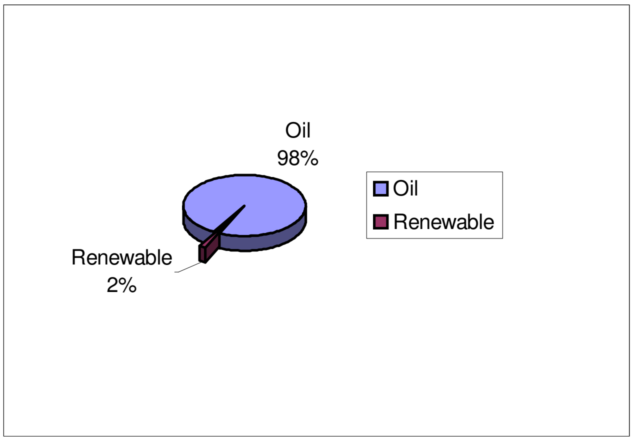


Fig.A.9: Production percentage in the non-interconnected islands per type of fuel [9]

The below stations or units of production already have been manufactured or they are found under manufacture and they are expected to be placed in commercial operation between 2003 and 2005:

- Station of production with fuel of lignite and installed capacity 330MW in Florina. The commercial operation of station is expected in the first half-year period 2003, while his operation it began from beginning May 2003.
- Hydroelectric power station with installed capacity 162MW in Mesohora. The commercial operation of station is expected at 2005.
- Station of production constituted from two oil units total installed capacity 102MW, in the Atherinolako, Lasithi. This station has been programmed is placed in commercial operation the second half-year period 2004.
- Station of production with fuel the oil (with parallel faculty of combustion of natural gas) in the Atherinolako, Lasithi, which will be constituted from two power plants total

installed capacity 90-100MW. The above station is programmed to begin its operation in 2006.

- Two units of production of electric energy with fuel diesel of installed capacity 28MW the every, are to install itself in the stations Chania and Linoperamoto. The commercial operation of units in question is appreciated that it will begin in June 2003.

From the total cost of budget for the manufacture of above stations or units, roughly the 70% it had been spent until 31.12.2002.

In November 2002 the Company submitted application in the Energy Regulation Authority, RAE for the manufacture of new unit of natural gas of combined circle in the station of Lavrion of total installed capacity 400MW roughly. In case where the RAE approves the application, the manufacture of unit in question will be completed in 28 months by the signature of convention.

A.3 Transmission of Electric Energy

The Operational Unit of Transmission has in her property the electric system of transmission of continental Greece by which is transmitted electric energy, via the lines high voltage, in entire the country. The operation of system of transmission is under the responsibility RAE.

The produced electric energy, by the stations of PPC or by independent producers and in the case of imported current from the points of interconnection with neighboring electricity systems, is transmitted in the big industrial consumers and in the network of distribution by where it is then distributed in the continental country.

The vertebral column of linked system of transport they constitute the three lines of double circuit of 400kV, that transmit electric energy, mainly from more important for our country energy center of production of Western Macedonia. In this region, are produced roughly the 70% of total electric energy of country that is then transmitted in the big centers of consumption Central and Southern Greece, where are consumed roughly the 65% of current. The system of transmission allocates moreover lines of 400kV, also overhead, underground lines and submarine cables of 150kV, as well as submarine cables 66kV that connect the islands of Western Greece, with the interconnected system.

Moreover, the system of transmission is connected with neighboring electricity systems of Albania, FYROM, and Bulgaria as well as with direct submarine cable of 400kV of continuous current with the electric system of Italy.

In the *Fig.A.11* are presented basic elements of network of transport 400kV. 31 December 2002 the linked system of transport included 10,330km of lines, as it appears in *Table A.7*.

In the dues 2002, the system of transmission included also 493 transformers and autotransformers with total nominal power 36,845MVA. Today, the Operational Unit of transmission executes the daily natural operation, the maintenance and the development of interconnected system of transmission, according to the indications HTSO.

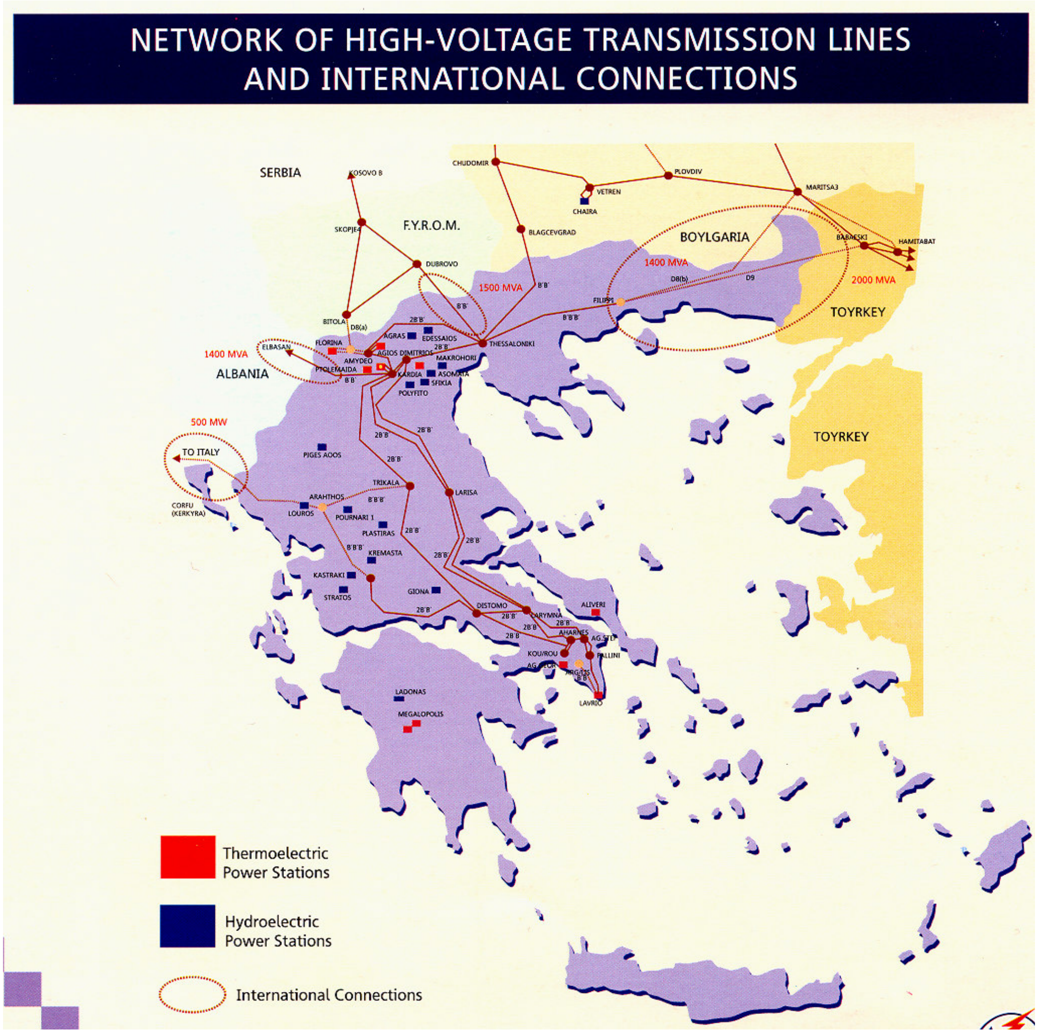


Fig.A.10: Network of high-voltage transmission lines and international connections [5]

Table A.27: High voltage transmission lines (km)[4]

	400 KV	DC 400 KV	150 KV	66 KV	Total
Overhead lines	2,272.17	105.95	7,761.19	39.05	10,178.36
Submarine lines	-	-	107.84	15.00	122.84
Underground lines	-	-	28.37	-	28.37
Total	**2,272.17**	**105.95**	**7,897.40**	**54.05**	**10,329.57**

The Electric System of Greece is characterized by big concentration of power plants in the North (lignite production of region Ptolemaida, hydroelectric power stations) and big concentration of consumption in the South (region of Capital). International interconnections are found in the North and consequently that means more severe unbalance, in any case of intense phenomenon of mass transmission of energy from North to South. (*Fig. A.12*)

The transmission system can be separated in two sub systems. The **main transmission system** constituted by network Hyper High Voltage (400kV) that has been drawn precisely in order to it ensures the economic and sure mass transport in direction N-S. The network of High Voltage (150kV, 66kV) can be considered as network **hypo-transmission.** It ensures the further transmission from the High Voltage Centers in the Low Voltage distribution network.

Because of the form of Production-Transmission System is presented the phenomenon of unfair division tendencies between N-S and consequently problems of increased losses and need of support of voltage in the South. For the confrontation of precisely these problems has been drawn with the above way System Production-Transmission. Firstly has been developed capable production in the South, or with lignite units (Megalopolis), or with petrol (Aliveri, Lavrio, Keratsini). At the same time the network of Transmission North-South particularly has been strengthened (3 lines 400kV of double circuit). In this way, it has been ensured, that for the present conditions and for the next decade, the losses are kept in reasonable level and the support of tendency of System in the South is the adequate. (*Fig. A.12*)

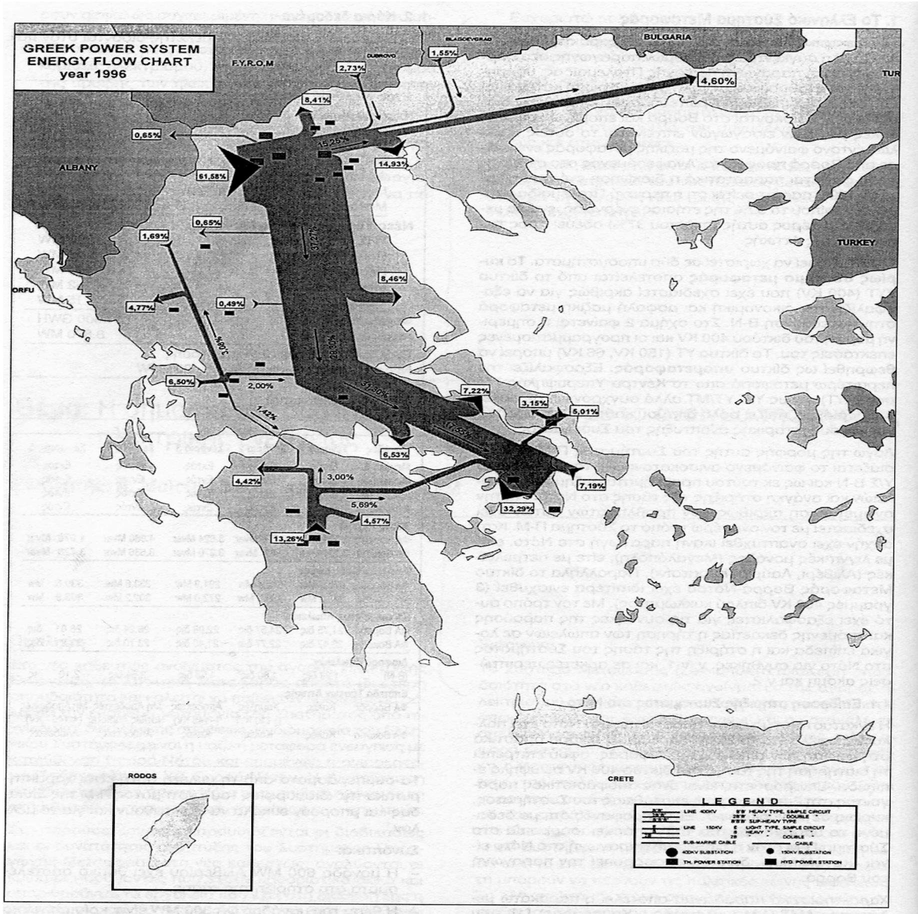
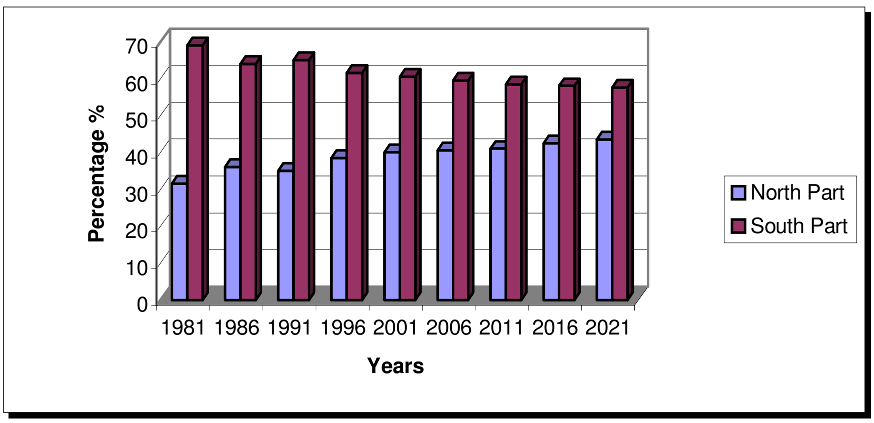


Fig.A.11: Electric energy flow 1996 (The region of Ptolemaida produces roughly the 62% of annual energy, while the bigger part of this (roughly 37%) leads to the region of Attica). [5]

Fig.A.12: Progress of the participation of the north and south part to the total energy consumption [5]



A.4 Distribution of Electric Energy

The Operational Unit of Distribution is person in charge for the distribution of electric energy in all the Greek territory, so much in the region of linked system what in the not linked islands, supplying thus with electric current all the customers of PPC (includes remaining the customers medium and high tendency). With the term "distribution" is meant the transport of electric energy from the system of transport in the final consumer.

According to the N.2773/1999, the PPC as the unique distributor of electric energy in Greece, on the present, apart from the obligation of distribution of electric energy in the customers, is compelled to provide access in the network of distribution in all the holders of authorizations of production and supply of electric energy, as well as in Selecting Customers.

So that it provides the access in question in the network of distribution, the PPC has right to debit the producers, their customers and the suppliers with an end of connection, which is approved by the Minister of Growth, after consultation of RAE. *Table A.8* presents the network of distribution in Greece at the 31.12.2002:

Table A.28: Total distribution lines (Interconnected system and non interconnected islands (km))[4]

	150, 66KV	22, 20, 15, 6.6 KV	230-400 V	TOTAL
Overhead lines	653	86,122	96,107	12,229
Submarine lines	-	1,024	1	1,025
Underground lines	144	7,112	9,826	17,082
Total	797	94,258	105,934	200,989

Also, with date 31.12.2002 the network of distribution includes 130,924 transformers of middle of so much total force 20,783MVA. In *Table A.9* are presented the quantities of sold electric energy, above category of customer in the linked system and the total of income from the each category at uses 2000 until 2002.

Table A.29: Sales of Electric Energy in the Interconnected System[4]

1/1 – 31/12	**2000**		**2001**		**2002**	
	GWh	**Million €**	**GWh**	**Million €**	**GWh**	**Million €**
Industrial Sector						
High Voltage	6,585	235	6,719	232	7,028	244
Medium & Low Voltage	8,626	393	6,819	414	6,921	435
Commercial	8,726	775	9,462	866	10,023	953
Domestic	12,907	916	13,207	954	14,280	1,071
Agricultural	2,676	88	2,562	88	2,266	83
Others	1,859	141	1,953	150	1,99	161
TOTAL	39,379	2,547	40,715	2,703	42,516	2,947

In the *Table A.10* are presented the quantities of sold electric energy per category of customer in the not linked islands and the total of income from the each category at uses 2000 until 2002.

Table A.30: Sales of Electric Energy in the non Interconnected System [4]

1/1 – 31/12	2000		2001		2002	
	GWh	Million €	GWh	Million €	GWh	Million €
Industrial Sector						
High Voltage	-	-	-	-	-	-
Medium & Low Voltage	255	18	275	21	288	21
Commercial	1,420	126	1,551	141	1,645	159
Domestic	1,300	97	1,339	103	1,495	120
Agricultural	234	12	218	9	218	8
Others	354	26	374	29	390	32
TOTAL	3,563	279	3,757	302	4,036	339

A.5 PPC's supply of lignite

PPC's lignite mines in Ptolemaida and Megalopolis provide the Greek economy with its most important source of fuel for electrical generation -lignite- on which the electrification of the country has depended since the founding of the Public Power Corporation. Lignite is found in great abundance in Greece's subsoil in terms of ignite production, our country is second in the European Union and sixth worldwide. On the basis of Greece's total deposits and anticipated future rate of consumption, it is estimated that the domestic supply of ignite is enough to last for more than 50 years.

Up to date, a total of 1.2 billion tons of lignite have already been mined, while exploitable reserves total approximately 3.5 billion tons. In 2001, a total of 66.2 million tons were mined, a record since the beginning of the mines operation. Today, PPC's lignite power stations comprise 44% of the country's total installed capacity and produce nearly 67% of the country's electrical energy.

The utilization of lignite in generating electricity offers Greece enormous savings in foreign currency reserves (approximately 1 billion dollars annually). Lignite is of strategic importance for PPC, because of the Low cost of extraction; it guarantees a stable and easily monitored price, and offers both stability and security in the availability of fuel supplies.

At the same time, the utilization of Lignite provides thousands of jobs throughout the Greek countryside, where high rates of unemployment prevail. In all of these ways, Lignite has contributed significantly to the growth of the Greek National Product. *Table A.11* it mentions the production of mines of Company at three-year period 2000-2002. (*Fig. A.14)*

Table A.31: Production per Lignite center (in million tons)[4]

	2000	2001	2002
Lignite Center of West Macedonia	50.83	51.72	55.83
Lignite Center of Megalopoli	12.48	14.45	14.51

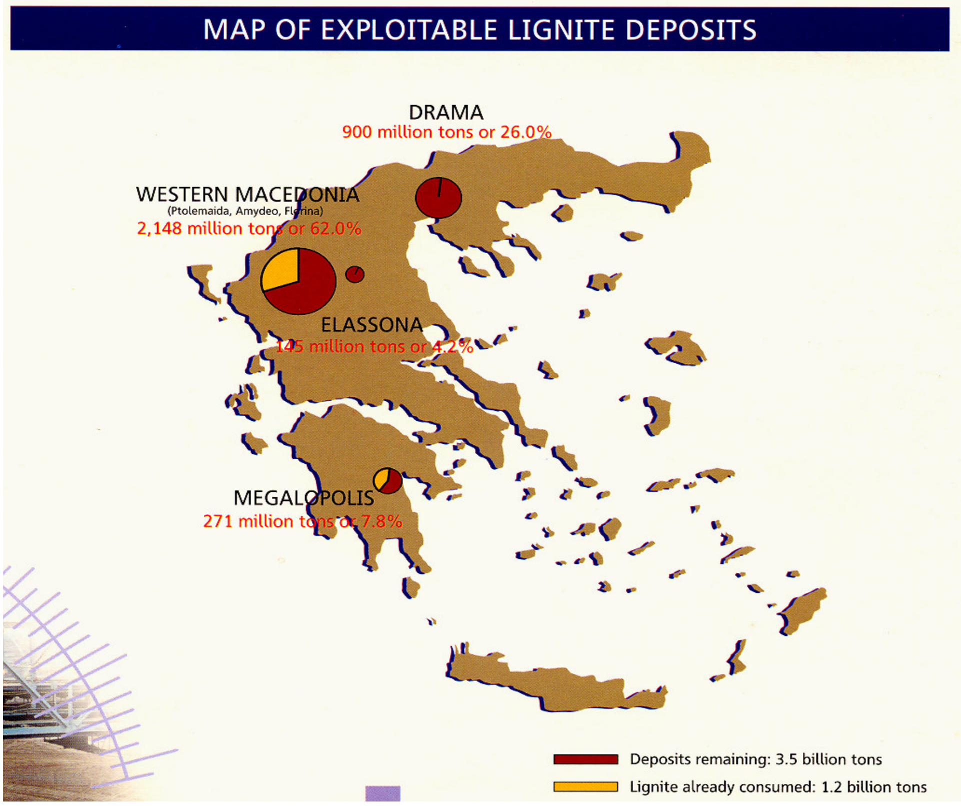


Fig.A.13: Map of exploitable lignite deposits [5]

A.6 Environmental issues

The main activities of PPC, is excavation of lignite, production, transmission and distribution of electric energy are regulated by a wide environmental legislative frame, which includes laws and remaining lawful provisions what has been adopted in the Greek legislation with the incorporation of Community Directives and corresponding international agreements.

The environmental legislation that influences the operations of PPC concerns mainly in the emissions of gases, in the pollution of aquatic resources, in the disposal of waste and in the electromagnetic fields. The main by-products of production of electric energy from mining fuels are the emissions of dioxide of sulfur (SO_2), oxides of nitrogen (NO_X), carbon dioxide (CO_2) and hovering particles as dust and ash.

A.7 Handling of the electric consumption peaks

The latest years is observed internationally an important increase of consumption of electricity. This increase and particularly the increase of peaks of electricity demand have taken worrying dimensions. The electric systems of countries with warm climate (Mediterranean) face with difficulty the demand of electricity of heat summer days.

Objective of the following paragraphs is it presents and it analyzes the existing situation. The problem of peaks in the demand of electricity is not faced only with increase of installed force of stations of generation of electricity or with imports of electricity. Is proposed a line of solutions, or rational use of energy or management of charge, which if they are applied, they blunt considerably the problem.

A.8 Analysis of existing situation

The problem of peaks of electric demand in Greece has emerged the last years. It is common, in bigger or smaller degree, in all the countries of Mediterranean and the USA, while it has begun to be observed even in Scandinavian countries.

For the comprehension of problem it is useful is examined the consumption of electricity in the country. For reasons of brevity, will only be mentioned the elements which concern the consumption of continental Greece and interconnected islands. The islands with autonomous electric networks have, obviously the same and perhaps increased problems.

At the last 30 years is observed a continuous and linear increase of electric consumption (*Fig. A.15*). The mean annual rate of increase of total consumption for the examined interval is roughly 6.1%. It is obvious that the attendance of industry (High Voltage) in the increase of consumption is very small. Also, it has been observed that a lot of big industrial units control their consumption and use abundance of advanced systems of saving of electric energy. The rate of increase of electric consumption is owed main in the consumption of distribution, in users that is to say except the heavy industry.

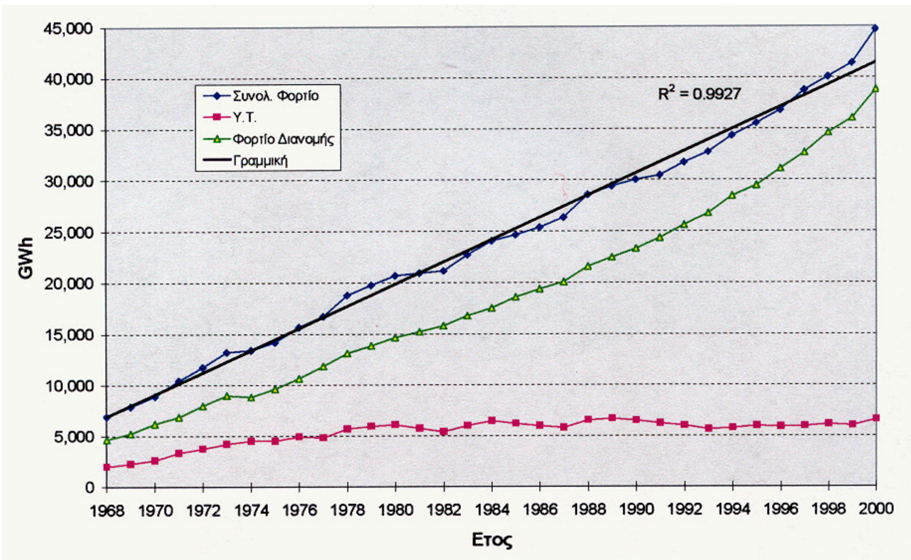


Fig.A.14: Annual consumption of Electrical energy [6]

The increase in the consumption at the examined period is owed in the substantially complete electrification of country, and the increase of biotic level. It has been observed internationally, that the increase of energy consumption is proportional with the increase of Gross National Income, something that is not in effect absolutely for Greece at the examined interval. The increase of electric consumption of is bigger that of Gross National Income.

The *Fig. A.16*, describes the increase of peaks of demand at time interval (1968-2000). The rate of increase of peaks for the examined time interval is 6.25%, similar that is to say with that of consumption. Examining the primary elements, it appears that the peaks afterwards 1992 were transported by the winter (end December) in the summertime (July or August). The peaks in the high voltage (heavy industry) are checked and their behavior follows that of consumption. On the contrary in the distribution the observed peaks at the last five-year period are increased with rate of order the 8-9%, putting in danger the stability of electric system of country. The electric system it is not possible and economically acceptable, to follow with facility such rate of increase in the demand of peak. The peaks are faced with import in the system of all the units of PPC, as well as with imports from third countries.

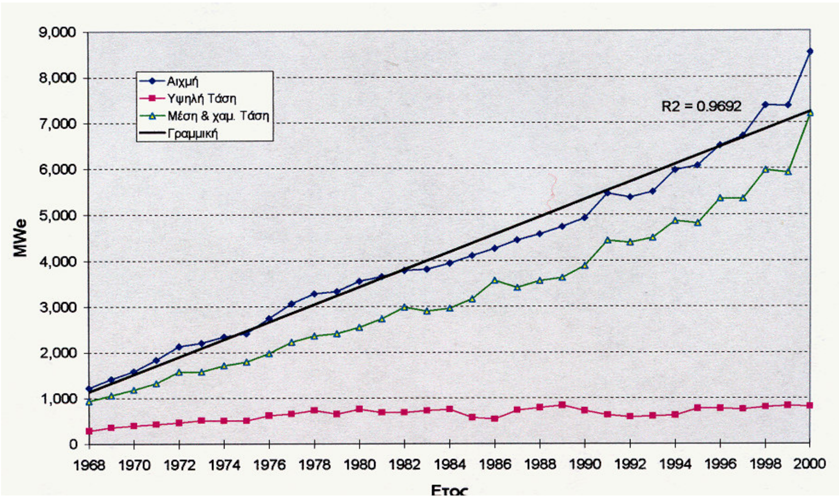


Fig.A.15: Peaks of demand of electric energy [6]

The PPC's measurements of consumption, allow the analysis of distribution in big teams of consumers. This analysis appears in *Fig.A.17*.

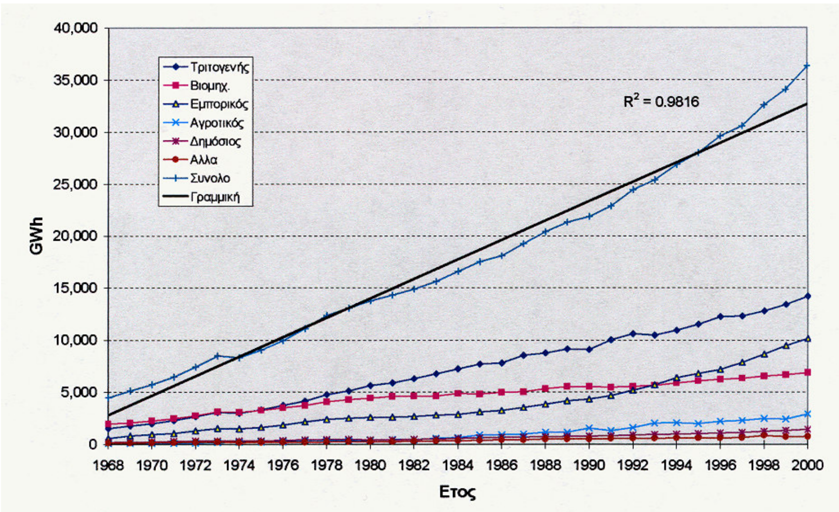


Fig.A.16: Analysis of electric consumption (low and medium electric voltage)[6]

Examining the rate of increase of examined time interval, (accepting that the increase is linear) results the *Table A.12* for the various teams of consumers.

Table A.32: Percentage of different sectors [8]

Sector	Tertiary	Commercial	Industrial	Agricultural	Public	Total
Increase %	7.25	9.4	4	11.8	6.5	6.75

The annual increase the public and industrial sector is near in the mean and does not inspire concern. The bigger rate of increase, which should be examined and is analyzed, is observed in agricultural as well as in the commercial sector.
The big increase of consumption of electricity in the agricultural production is owed in following reasons. These reasons include the low infiltration of electricity at the beginning of examined period (1968), the intensifying of production, the low price of electricity (50% domestic or 30% of commercial) as well as the low infiltration of systems of control in the agricultural cultures and mainly in the pumping.

The commercial sector requires particular attention because the high absolute price of consumption but also the rate of increase at the last decade. The big commercial chains have the technical infrastructure and are interested, in general lines, for the energy consumption of their installations. The experience shows that the problem is located mainly in small shops, where coexist unacceptable high levels of lighting with medium or bad quality lightning systems, and air-conditioning systems in full operation with the entries of shops remaining permanently open. The designers of such installations interest itself for the attracting of customers, the presentation of products and usually ignore the parameter of energy consumption.

The *Fig. A.18* shows the daily peaks at the duration year (1997). The observed circles are the weeks of time. As it is expected, the consumption during one week is smaller at the Weekends and it is increased in the weekday days. It is obvious that the behavior of peaks is almost constant at the duration of time, with the exception of the summer period. There is observed an increase of order the 20%, which, for the examined year lasts roughly 8 weeks.

This increase implies, obviously, the effect of air conditioning in the consumption of electricity. Analysis in hourly prices for 2 formal weeks of year they are attached in the end of article.

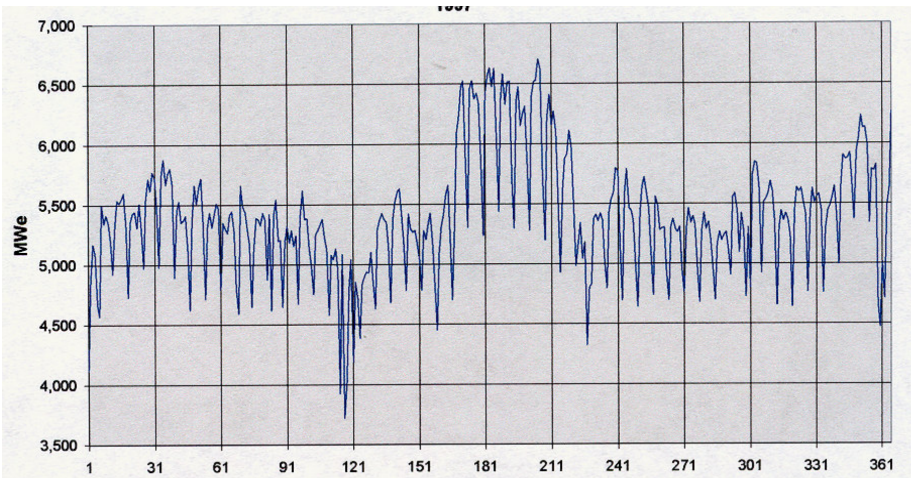


Fig.A.17: July Daily Peaks – Continental network (Interlinked)[6]

A.9 Air conditioning

The wide spread of air conditioning in Greece began afterwards the summertime 1988. The *Fig. A.19* gives the increase of sales of air-conditions at the five-year period 1991-1996, for which exist official statistical data.

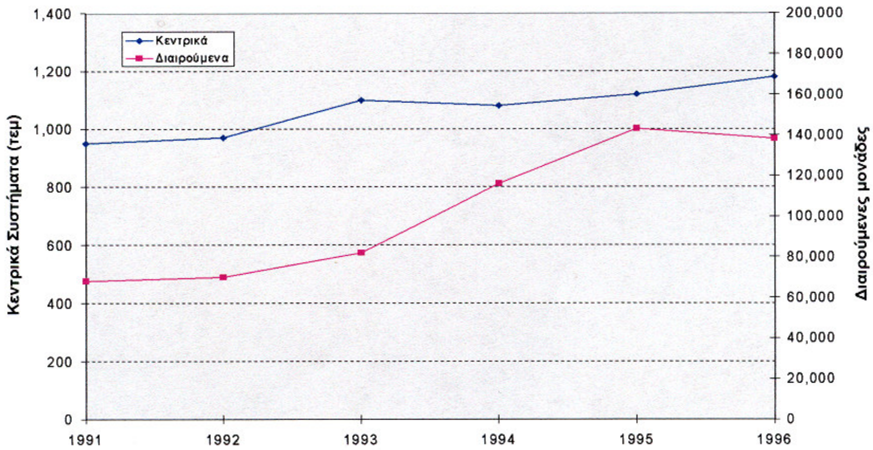


Fig.A.18: Sales of Air-conditions [6]

The increase of sales of central instruments is only about 20% at the five-year period - from 950 in 1180 item. On the contrary the increase of sales of small air-conditioners of is order the 100% - from 68,000 in 138,000 items.

With base estimations, the distribution of the increase of installed capacity for air conditioning in 1996 has as follows: (*Table A.13*)

Table A.33: The increase of installed capacity for air conditioning in 1996 [6]

Systems	Installed capacity	Percentage %
Central	46	15
Semi-central	70	22
Split	196	63
Total	310	100

The 2/3 of the capacity that installed in 1996 was in small units, while only the 1/3 was for semi-central or central units. This relation has not changed in favor the central units up to today.

A.10 Proposed solutions of rationalization of charges.

From the above, it results that exists an important problem of increase of consumption and particularly her peaks. This increase does not appear to decrease itself in the direct future. On the contrary it is expected to be accentuated front it begins to be blunted. The problem of peaks is owed main (above 60%) in the increase of installed force of air-conditioning instruments.

Next, will be presented certain solutions about what can be done to decrease the problem.

A.10.1 Manufacture of new power plants

For the confrontation of increase of capacity (*Fig. A.15*), are required new electrical power units of capacity about 500-600MW each year. The mean cost of such station (depending on the technology) is about 100,000-150,000€. Examining the *Fig. A.18,* we observe that this station will function about 2 months/year. The pay off time is expected to be six times bigger in compare to a station which functions in annual base. It is expected exceeds the 25 years, interval which is equal or bigger than the time of life of instruments.

The manufacture of such stations with alone objective the confrontation of peaks, is obviously economic disadvantageous.

A.10.2 Systems of control electrical motors (regulators of revolutions - inverters)

These systems are applied in electric engines with altered charge. The electric consumption of engines follows the charge. The pay off time of such systems becomes in few years. Their main applications are in

- Industry. The rate of infiltration in big units is very high; with result dynamic new industries check almost the total of their engines.
- Agricultural sector. Particular accent should be given in pump units, where the level of water horizon changes permanently and the consumption of pump will be supposed him it follows.
- Building sector. These appliances find application mainly in parts of systems of air conditioning and heating

In European Level, is feasible the reduction of consumption at 150TWh in the industry and 120TWh in the building sector.

A.10.3 Planning of buildings

The more important stage in the energy efficiency of building is his initial planning. The erroneous planning is in many times not reversible and leads to over-consumption during life of building. Initially accent should be given in the planning of structure.

Particular importance it is the proportion of openings/walls. In *Table A.14* are given indicative cooling loads for $1m^2$ of elements of nutshell in Southern and Western orientation. The openings are all in metal frame. The Northern and Eastern orientation do not add important loads in the time periods at which exists problem in the electric network.

Table A.34: Indicative cooling loads for $1m^2$ of elements of nutshell in Southern and Western orientation [6]

Element of nutshell	Southern	Western
Isolated wall	8	12
Single panel window	311	700
Double panel window	265	677
Double panel window with reflection	160	354
Double panel window with internal shading	185	419
Double panel window with external shading	65	129

The above loads are indicative and it will be supposed they are examined mainly as for qualitative characteristically. The biggest loads are presented in the Southern orientation in period 11:00-13:00, while in the Westerner between 15:00- 17:00.

Obviously the overwhelming majority of cooling loads from the nutshell of building emanates from the openings (panels). The contribution of walls is very small. The double panes offer very small reduction of loads compared with alone. The reflective panes offer almost the same protection with the internal shading of double panes. As he is expected, the exterior shading offers the optimal results.

The shading of buildings should be such that it allows the biggest possible reduction of loads of refrigeration without it increases the loads of heating and the needs for artificial lighting. The *Table A.14* shows that the shading should be exterior and variable, so that it allows the bigger possible infiltration of natural lighting in the building. A lot of buildings and particular intended for professional use are based to a large extent on the artificial lighting, which apart from direct electric consumption increases also the loads of air conditioning, because the emitted heat from the lightings.

A.10.4 Air conditioning
It is obvious (*Fig. A.18*) that the bigger increase of peaks is owed in the systems of air conditioning. The increase infiltration of these systems is inevitable. There are techniques and technologies, which are essential to be applied immediately, in order to moderate the negative effect of air conditioning in the electric system and in the energy consumption of buildings.

A.10.5 Planning and use of efficient systems
The logic of energy planning of nutshell of building follows the planning of electromechanical systems of most optimal energy efficiency. These systems will change and transport energy with most optimal efficiency. Simultaneous electromechanical systems of equipment, as lightings, machines of office, should be efficient so that they do not add cooling charges.

The engineer air conditioning in Greece, but also more generally in the countries of Mediterranean, is a relative new activity. The present situation is transient and leads to the complete air conditioning of buildings. A lot of buildings do not have total confrontation of subject of refrigeration (contrary to the heating which is central). The increasing needs for refrigeration in existing buildings are faced with the placement of small divided units. These units even if preferred by the users for various reasons, do not offer high degree of comfort compared with central systems, while in favor they consume energy until 35%. Their decreased initial cost of installation, concerning the central systems, she is lost in few years of operation.

In level of country and European Union, have not been established publics acceptable specifications in the cooling instruments, while they exist dark in the knowledge of craftsmen that deals with the small systems. These problems will be untied progressively, as it happened with the systems of central heating. Will be supposed however the involved institutions to accelerate their solution. It is necessary while exist strategy in the refrigeration of building (as it exists in the heating) and the study and concretization to become from experienced personnel. The systems they are energy efficient.

A.10.6 Tri-generation

The application of method is dated by the dues of 19^{th} century. The cycle is based on the production of refrigeration by heat (steam or direct combustion) with cooling means the ammonia, or mix of lithium-Bromium etc. This refrigeration is produced autonomously or as by-product of units of co-production. The existing systems nets with absorption, they are currently bulky, with big initial cost. They have however minimal electric consumption and particularly quiet operation. Become efforts for creation of small systems.

The energy consumption for air conditioning with the application of this method, it is transported by the networks of electricity in the networks of liquids and gases of fuels. The Natural Gas, NG, could constitute ideal fuel. The profits concern companies of electricity generation, which see the peaks in load decreasing. In Attica was established finally tariff of use Natural Gas for refrigeration, which is expected to be applied. Indicatively we will report that in Japan of the 17% of air-conditioning systems of are absorption. The infiltration of such systems is big where the suitable mix of energy billing policy exists.

Advantages of co-production

Successful installation leads to reduction of fuel consumption of order 25%. The total efficiency of stations CHCP exits 85%. The reduction of atmospheric pollution is proportional. Also, with the use of natural gas the emissions SO_2, and soot are eliminated.

The reliability of energy distribution is increased. CHCP units which are linked with the electric network, where it gives or takes electricity guarantees unhindered operation in level of unit, in case of interruption of operation of station or electrification from the network. In level of country, it decreases the need of installation of big stations of generation of electricity and increases the stability of electric system of country. The co-production can be achieved with use of renewable sources of energy (biomass) substituting conventional fuels.

Technologies of co-production

The main part of installation CHCP is the machine, which produces heat and electricity.
The basic known technologies are:

1. **Gas turbine** (cycle Brayton). Known from his use in the planes. The air compressed up to the booth of combustion and following is defused
2. **Steam Turbine** (cycle Rankine). It defuses steam of high enthalpy, and produces mechanic work as well as steam of lower enthalpy
3. **Combined circle**. It is a combination of the above, with the use of recuperation boiler between them.
4. **Piston engine**. (Cycle Diesel or Otto).
5. **Fuel Cells**. The principal of these machines is the production of heat and electricity without combustion. With electrochemical activities in the fuel (mainly natural gas) are split and by the chemical reactions are produced heat and electricity (under form of ions).

Machines 1-4 produce electricity with generator coupled on their axis. Recuperation boilers, with or without additional combustion, produce the heat. The refrigeration is produced with the circle of absorption or adsorption.

Election of System

The first stage in the decision of installation of CHCP unit is the recording of energy requirements. The choice of system will become after have been taken all the other measures saving of the energy. The fuels that were consumed at the previous years are analytically recorded, hot water or steam that is used. Daily fluctuations they give the possibility of exploitation of unit. Become forecasts for future consumes also uses.

Economic analysis
The economic analysis is the one that will give clue for whether the co-production is acceptable and who technology will be applied.

The CHP (Fig. A.20) system will be connected with the electric network of country and it will:
1. provide (will sell) the electric energy excess
2. absorb (will buy) the necessary electric energy during the peak load. The cost of installation is constituted from: The cost of the investment, is the sum of cost of basic instruments of production of heat, cooling and electricity, installations of storage of fuel, likely filters of cleaning of fumes combustion, working, building installations, pipes, wirings, systems of control and finally the mechanical studies and supervisions.

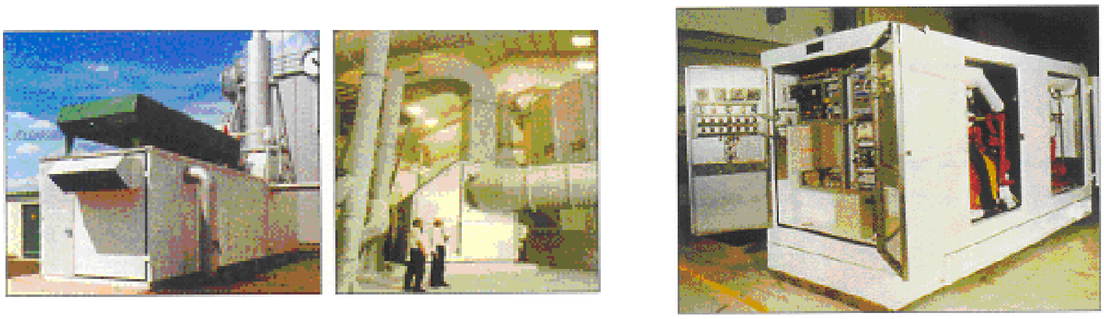


Fig.A.19: Small CHP unit [6]

The cost of operation and maintenance consist of:
- the cost of fuel for the main machine CHCP. Is added income from the sale of electricity in the network and is removed
- expenses of purchase of electricity from the network whish are balanced with the income from the sale of electricity in the network and are removed
- salaries and the cost of parts for the periodical maintenance of system

Existing situation
In Greece exist today roughly 20 units in industrial mainly installations, with attendance 2% in the generation of electricity of country. The installation of new units moves with slow pace.

A.10.7 Storage of refrigeration
The use of refrigeration storage systems decreases the high loads at hours of peak, transporting big part of load in the evening hours for the charge of reservoirs of storage. It is applied in big buildings with central systems of refrigeration. With this technique is probably not achieved saving of energy, exists a small increase of consumption of energy, which however appears in periods out of peak. The profits for the user are economically also arise from the reduction of biggest demand of load. The results are beneficial for the system of generation of electricity, because shift of peak loads in nightly hours.

It can be shown that in Greece has for 1996 the appreciated reduction of peak with use storage of is in the order of 40MW. This reduction acts in total each year and is proportional with the installed force of central units.

A.10.8 Use underground or marine waters

The concentrators between, the pumps of heat, the systems of absorption can be frozen the summertime or be heated the winter, where it is possible, from underground waters or water of sea. This waters have also in the two seasons much better temperatures than the air of environment which they use the air-cooled air-conditioners. Result of this is the important increase of efficiency and consequently the saving of energy for air conditioning. The factor of efficiency an air-cooled system of compaction is increased from roughly 3, in 4-6 when this is become water -cooled.

A.11 Conclusions

In the last paragraphs was examined the problem of increase of consumption of electricity and particularly the peaks.(Figs A.21,A.22). Were examined various sectors of economy and it was found that agricultural and commercial require attention and further analysis because the big rate of increase of consumption. The peaks of demand are observed the summertime and are owed in very large percentage (on the 50%) in the air conditioning. These problems are not faced only with the increase of installed force of generation of electricity and with imports of energy. Ways of reduction of peaks were proposed via systems of saving of energy and management of load. In the methods and the systems of saving of energy are included the energy planning of buildings and systems of air conditioning, the regulators of turns, the efficient systems of lighting and the use underground or marine waters in the air conditioning. It is to the common interest of users but also companies of energy become management of load, with systems of storage of refrigeration and refrigeration with absorption. In order to be promoted these systems, it will be supposed to exist change of tariffs of electricity in multi-zone for big buildings and exists competitive pricing of natural gas for summer use or in units' productions or in direct refrigeration.

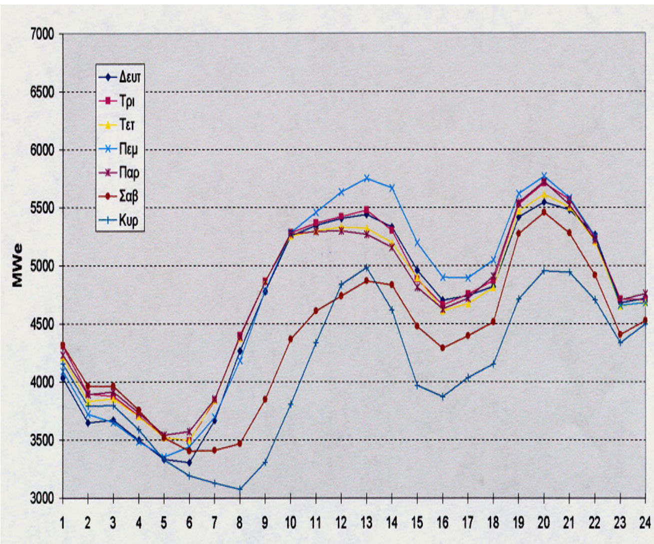


Fig.A.20: Consumption of electric energy in formal one wintry week (day 28-34) [6]

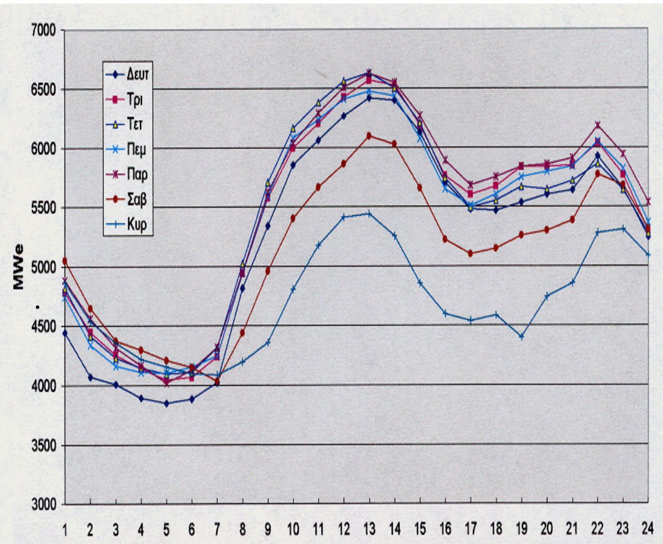


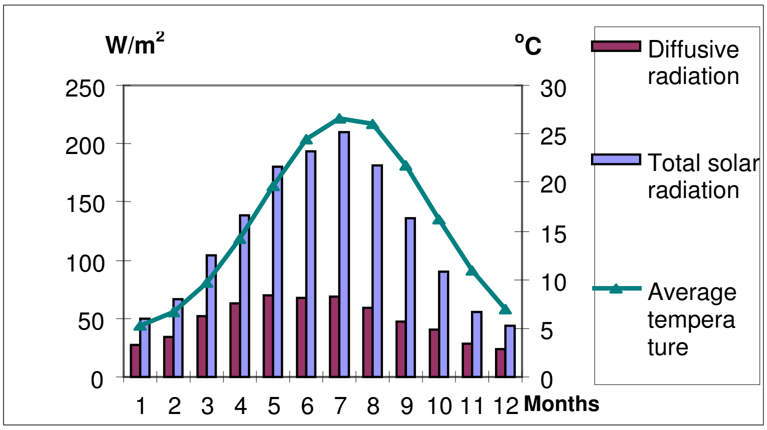
Fig.A.21: Consumption of electric energy in formal one summer week (day 192-188) [6]

APPENDIX B

B.1 Airport processing procedure

<u>Climate data (Thessaloniki region)</u>

Fig.B.1: Diffusive, total solar radiation and average temperature



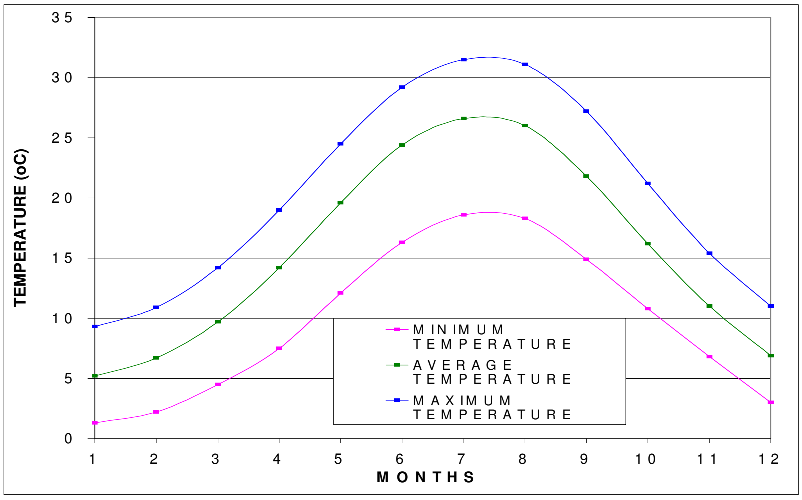
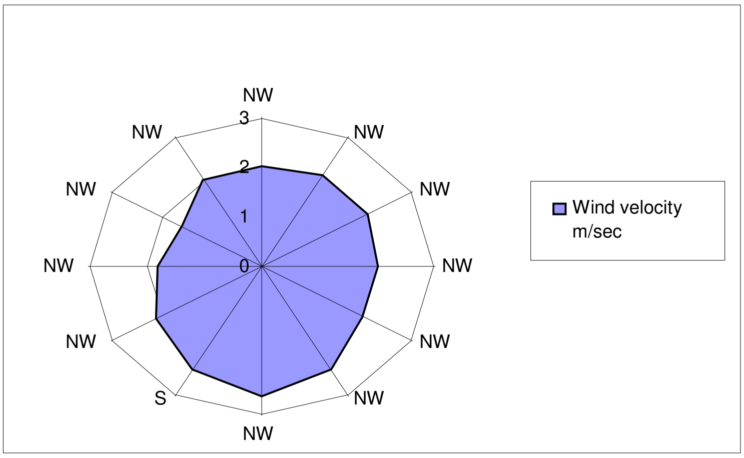


Fig.B.1: Absolute minimum, maximum and average temperatures

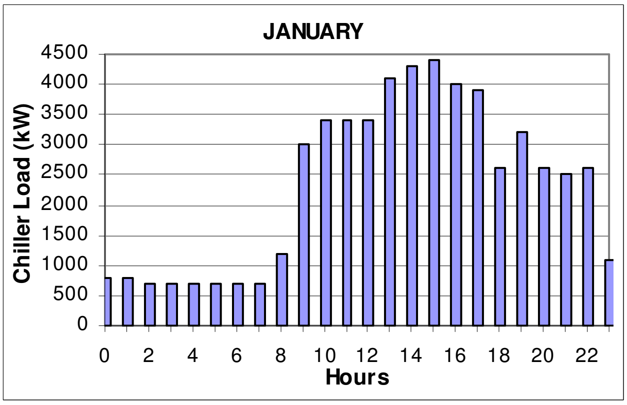
Fig.B.2: Average velocity and direction of the wind during the year

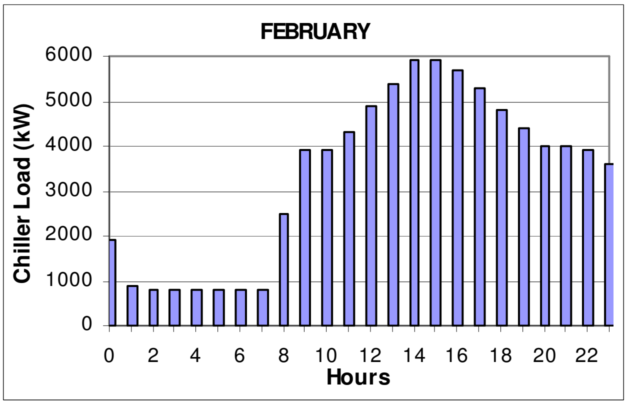


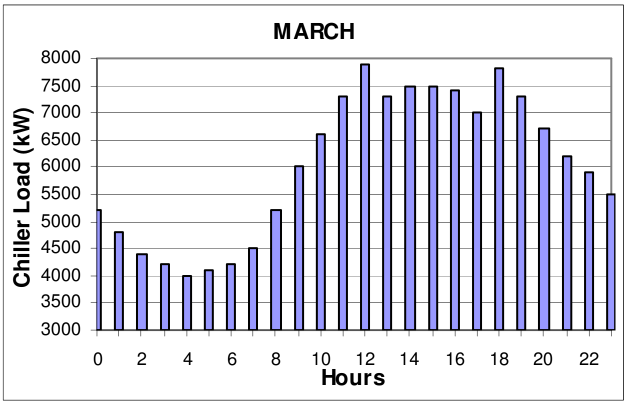
Cooling

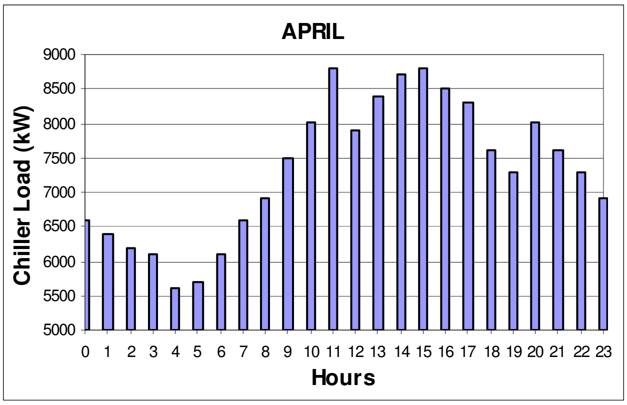
Table B.1: Daily profile of the cooling power, for all months of the year, (typical day)

Hour of Day	Chiller Load (kW)											
	JAN	FEB	MAR	APR	MAY	JUN	JUL	AUG	SEP	OCT	NOV	DEC
0:00	800	1900	5200	6600	8300	10000	9900	9800	9000	6700	4700	1100
1:00	800	900	4800	6400	8100	10000	9800	9900	8700	6500	4400	800
2:00	700	800	4400	6200	8000	9800	9700	9700	8600	6400	4100	800
3:00	700	800	4200	6100	7900	9700	9800	9700	8600	6300	4000	800
4:00	700	800	4000	5600	7700	9500	9600	9500	8500	6100	3600	700
5:00	700	800	4100	5700	7800	9400	9500	9700	8300	6200	3700	700
6:00	700	800	4200	6100	8100	9800	9700	9900	8600	6300	3800	700
7:00	700	800	4500	6600	8400	10100	10100	10100	9100	6700	4000	800
8:00	1200	2500	5200	6900	8700	10500	10500	10500	9300	7000	4900	1500
9:00	3000	3900	6000	7500	9300	8900	9100	8900	9900	7500	5800	3700
10:00	3400	3900	6600	8000	8200	9300	9800	9900	8900	8100	6300	4200
11:00	3400	4300	7300	8800	8700	10000	10000	10100	9200	7500	7000	4400
12:00	3400	4900	7900	7900	9300	10400	10600	10500	9800	8000	7700	4800
13:00	4100	5400	7300	8400	9800	11000	11000	11100	10100	8600	6900	5400
14:00	4300	5900	7500	8700	10100	11300	11300	11300	10400	8800	7100	5700
15:00	4400	5900	7500	8800	10200	11400	11600	11400	10500	8800	7100	5800
16:00	4000	5700	7400	8500	9900	11200	11300	11100	10300	8600	6800	5500
17:00	3900	5300	7000	8300	9700	11400	11100	10900	10000	8200	6700	5200
18:00	2600	4800	7800	7600	8900	10100	10400	10100	9300	7500	7500	4700
19:00	3200	4400	7300	7300	8600	9700	9700	9700	8900	7300	7100	4400
20:00	2600	4000	6700	8000	7800	9000	9200	9000	8300	7300	6500	4100
21:00	2500	4000	6200	7600	7400	8600	8700	8700	8000	7700	6000	4000
22:00	2600	3900	5900	7300	8900	8300	8300	8300	7500	7300	5600	3800
23:00	1100	3600	5500	6900	8700	9600	7800	8000	9200	7000	5300	2900
TOTAL	**55500**	**80000**	**144500**	**175800**	**208500**	**239000**	**238500**	**237800**	**219000**	**176400**	**136600**	**76500**
AVERAGE	**2313**	**3333**	**6021**	**7325**	**8688**	**9958**	**9938**	**9908**	**9125**	**7350**	**5692**	**3188**









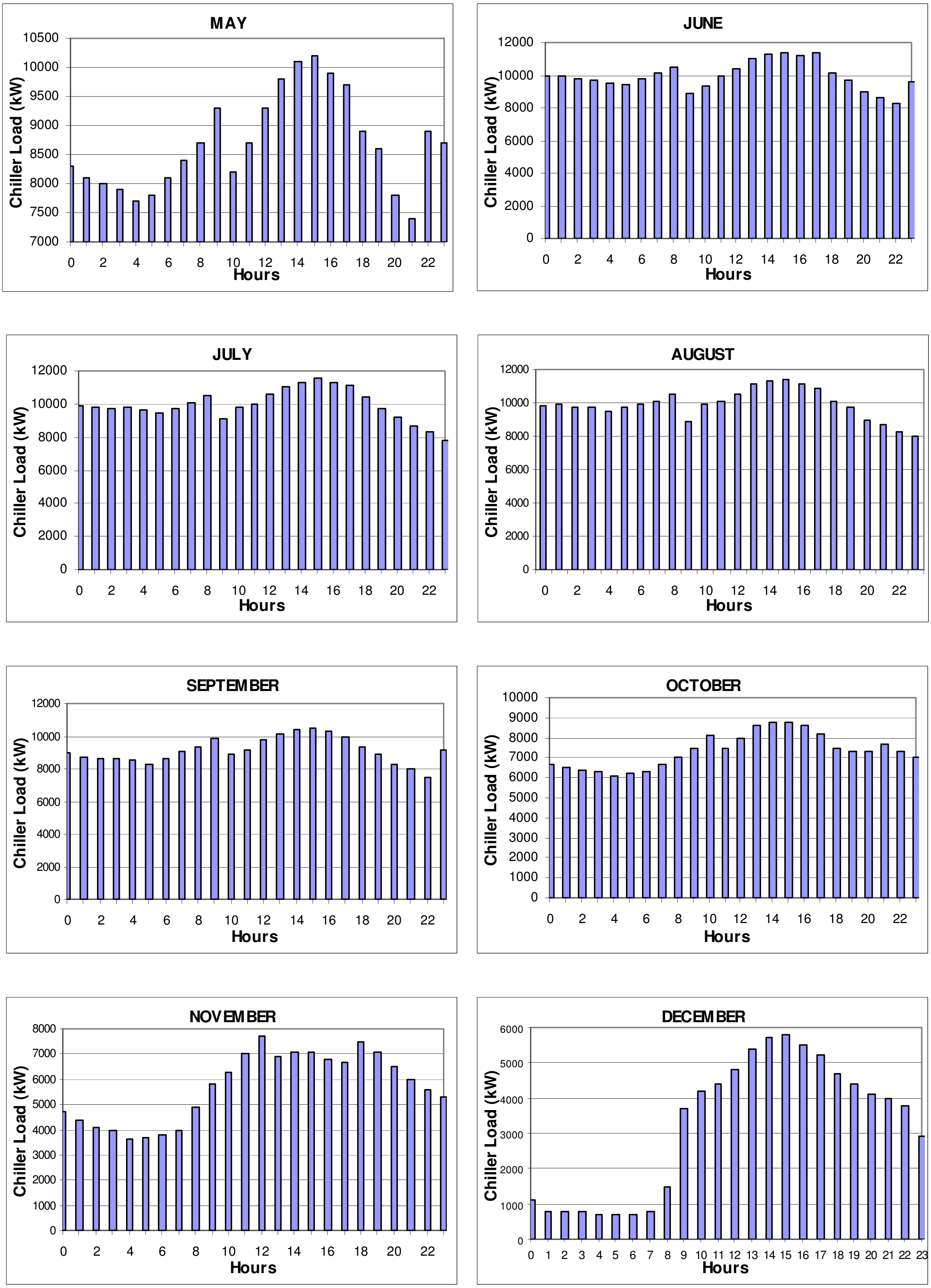


Fig.B.3: Daily profile of the cooling power, for all months of the year, (typical day)

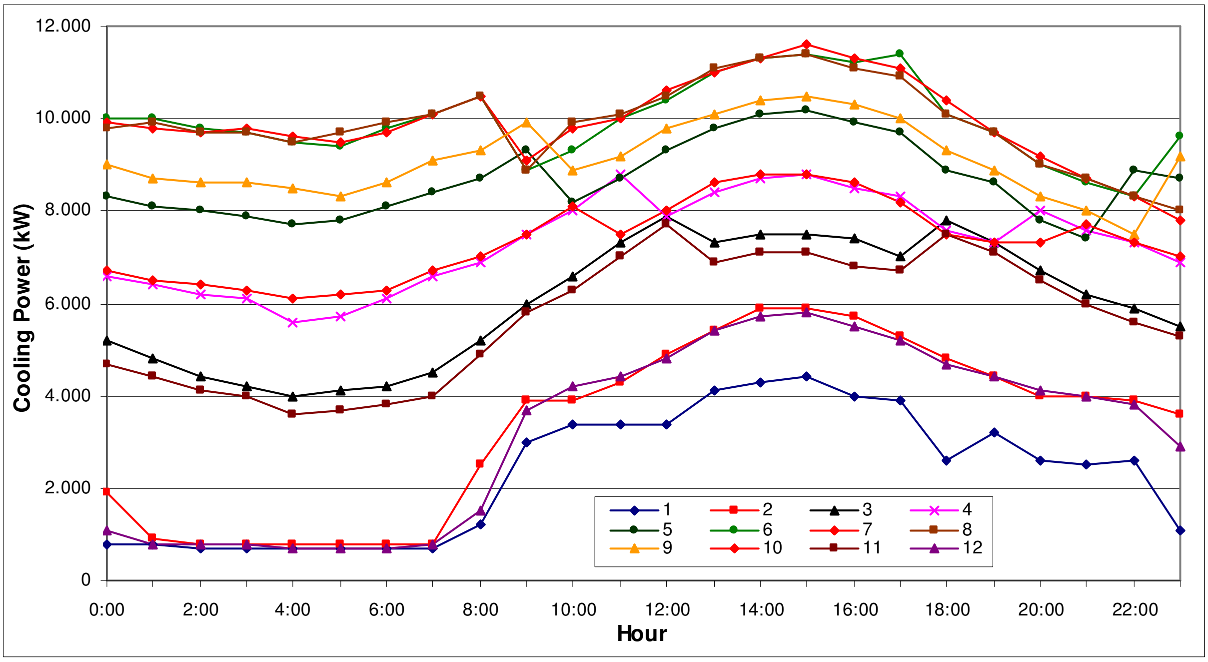


Fig.B.4: Comparative daily profiles of the cooling power, for all the months of the year, (typical day)

Heating

It is known that, the heating system has total thermal power **9MW** and is using **diesel** as fuel. Taking into account the data, concerning the hours which the heating system is used in a typical day (*Table 2.2*) for each month, the consumption of energy needed for the heating of the airport can be calculated.

Lighting-Motion

The data coming from Overhead Lighting and Electric Equipment have been used for the calculations for the worst case (night) with maximum population (*Table B.3*). The data shown in the 2nd and the 3rd line are independent from the month and are in operation during all day and night. The data shown in the 4^{rth} line of *Table B.3* is dependable on the sunshine hours and the climatic conditions (rain, snow, fog, heavy showers, hail) each month (*Table B.2*).

Table B.2: Cloudiness approximate factors

Days with	rain	snow	fog	heavy showers	hail
Factor	0.5	0.3	1	0.7	0.4

The estimation of the lighting-motion energy and power shown in *Table B.4* is obtained for the full operation of the airport.

Table B.3: Installed lighting-motion power (kW)

Overhead lighting	1,371
Electric equipment	782
Base power = Electric equipment +20% Overhead lighting	1,056.5
Variable power = 80% Overhead lighting	1,097.1

Table B.4: Presentation of the estimation procedure for the lighting-motion energy and power

	JAN	FEB	MAR	APR	MAY	JUN	JUL	AUG	SEP	OCT	NOV	DEC
Night hours	14	13	12	11	10	9	10	11	12	13	14	15
Night's kWhs	14792	13735	12679	11622	10566	9509	10566	11622	12678	13735	14792	15848
Rain days	10.2	10	12	11	10.5	7.3	5.7	4.6	5.9	8.7	11.3	11.7
Snow days	2.3	2.1	0.7	0	0	0	0	0.1	0	0	0.2	1.2
Fog days	4.4	2.8	2.3	0.9	0.3	0.1	0	0.1	0.2	1.4	3.8	3.6
Heavy showers days	0.3	0.7	0.9	2.2	5.6	6.6	5.5	4.5	2.8	1.9	1.8	0.8
Hail days	0	0	0	0	0.1	0.1	0	0	0.1	0	0.1	0
Icy days	2.8	1.4	0.3	0	0	0	0	0	0	0.1	0.9	2.9
Day hours	10	11	12	13	14	15	14	13	12	11	10	9
Rain	1.7	1.833	2.4	2.38	2.45	1.825	1.33	0.997	1.18	1.6	1.9	1.8
Snow	0.23	0.231	0.084	0	0	0	0	0.013	0	0	0.02	0.11
Fog	1.467	1.027	0.92	0.39	0.14	0.05	0	0.043	0.08	0.5	1.3	1.1
Heavy showers	0.07	0.1797	0.252	0.667	1.829	2.31	1.797	1.365	0.784	0.5	0.42	0.17
Hail	0	0	0	0	0.019	0.02	0	0	0.016	0	0.013	0
Icy	0	0	0	0	0	0	0	0	0	0	0	0
Dark hours during the day	3.467	3.271	3.656	3.441	4.438	4.205	3.127	2.418	2.06	2.596	3.6	3.1
kWh due to dark hours	3803	3588	4011	3775	4869	4613	3430	2653	2259	2847.9	3953	3412.9
Variable energy MWh	18.595	17.323	16.689	15.397	15.434	14.122	13.996	14.275	14.94	16.6	18.8	19.3
Base energy MWh	25.357	25.357	25.357	25.357	25.357	25.357	25.357	25.357	25.36	25.4	25.4	25.4
Light-motion energy,MWh (typical day)	43.951	42.680	42.046	40.753	40.791	39.479	39.352	39.631	40.295	41.9	44.1	44.7
Total Power (MW) (typical day)	1.831	1.778	1.752	1.698	1.700	1.645	1.640	1.651	1.679	1.747	1.8	1.9

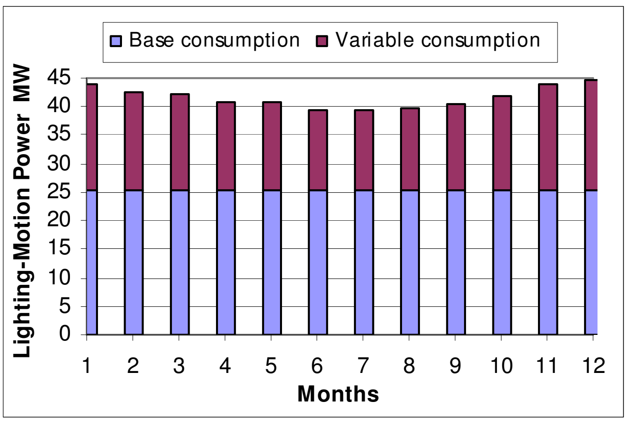


Fig.B.5: Distribution of base consumption and variable consumption, concerning the lighting and the motion of the airport

B.2 Lemnos processing procedure

The PPC gave us the statistical data, presented in *Table B.5,* concerning the economical year 2003. Having these data as a base, we modify them for the years 2001-2002, by taking into account the following factors:

1. Climatic data
2. Variation of the island population (permanent and tourists)
3. Economical growth per year

Table B.5: Total electric energy consumption per month (MWh), 2003

MONTHS	*JAN*	*FEB*	*MAR*	*APR*	*MAY*	*JUN*	*JUL*	*AUG*	*SEP*	*OCT*	*NOV*	*DEC*	**Total electric energy**	**Peak power MW**
MWH	4.365	4.370	4.495	4.016	3.693	4.318	5.489	5.902	4.237	3.898	4.009	4.673	53.466	11.000

Table B.6: Lemnos energy demand in MWh, 2001, (typical day)

YEAR: *2001*	**COOLING** MWh_c	**LIGHTING & OTHER** MWh_e	**HEATING** MWh_t		**TOTAL ENERGY**
			ELECTR.	BOILERS	
JAN	0	108.72	36.2	72.5	217.44
FEB	0	103.91	22.8	45.6	172.34
MAR	7.72	100.34	20.6	41.2	169.80
APR	13.3	103.71	16.0	31.9	164.87
MAY	29.31	81.56	16.6	33.1	160.57
JUN	35.88	99.36	2.8	5.5	143.52
JUL	49.93	124.82	3.6	7.1	185.45
AUG	57.67	128.8	5.8	11.5	203.77
SEP	35.94	94	8.3	16.6	154.83
OCT	19.7	83.72	19.7	39.4	162.52
NOV	0	97.92	24.5	49.0	171.36
DEC	0	106.74	37.5	75.0	219.24

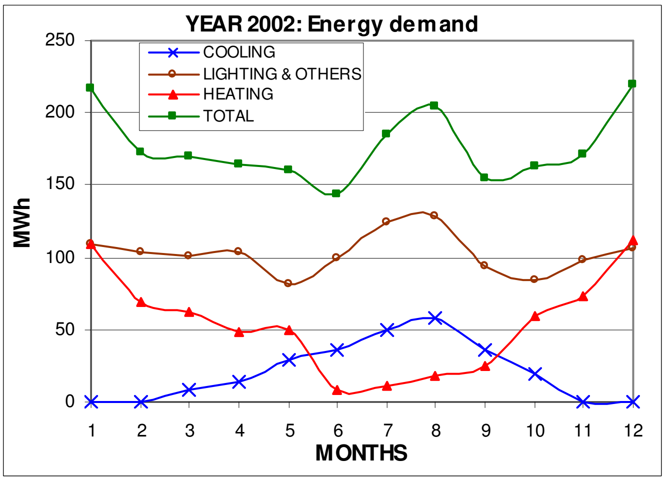


Fig.B.6: Lemnos energy demand in MWh, 2001, (typical day)

Table B.7: Lemnos power demand in MW, 2001, (typical day)

YEAR: *2001*	**COOLING** MW_c	**LIGHTING & OTHER** MW_e	**HEATING** MW_t		**TOTAL POWER**
			ELECT.	BOILER	
JAN	0	4.53	1.5	3.02	9.06
FEB	0	4.33	1.0	1.9	7.18
MAR	0.32	4.18	0.9	1.71	7.08
APR	0.55	4.32	0.7	1.32	6.87
MAY	1.22	3.4	0.7	1.38	6.69
JUN	1.5	4.14	0.1	0.24	5.98
JUL	2.08	5.2	0.1	0.3	7.73
AUG	2.4	5.37	0.2	0.48	8.49
SEP	1.5	3.92	0.3	0.7	6.45
OCT	0.82	3.49	0.8	1.64	6.77
NOV	0	4.08	1.0	2.04	7.14
DEC	0	4.45	1.6	3.02	9.14

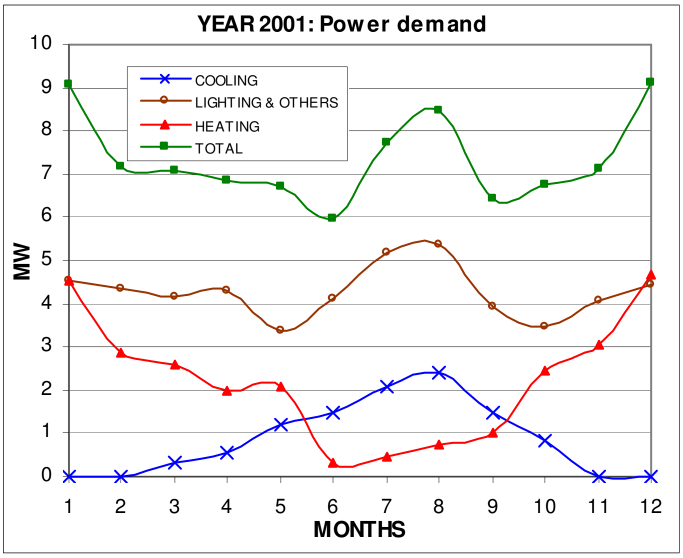


Fig.B.7: Lemnos power demand in MW, 2001, (typical day)

Table B.8: Lemnos energy demand in MWh, 2002, (typical day)

YEAR: 2002	COOLING MWh_c	LIGHTING & OTHER MWh_e	HEATING MWh_t		TOTAL ENERGY
			ELECTR.	BOILER	
JAN	0	120.4	51.6	103.2	275
FEB	0	106	41.3	83.9	231
MAR	3.5	131.2	40.2	91	266
APR	9.4	118.7	6.2	40.8	175
MAY	28	146.3	0.2	36.8	178
JUN	45.4	165.4	0.1	8.3	219
JUL	60	210.6	0.0	10.7	281
AUG	73.7	239.7	0.0	18.4	332
SEP	45.9	174.2	0.0	7.1	227
OCT	14.3	119.2	9.5	62	205
NOV	0	121.3	30.4	60.6	212
DEC	0	128.2	64.1	128.2	321

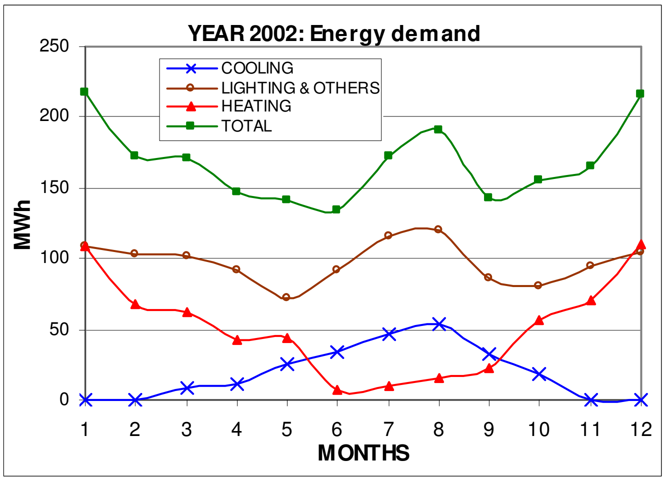


Fig.B.8: Lemnos energy demand in MWh, 2002, (typical day)

Table B.9: Lemnos power demand in MW, 2002, (typical day)

YEAR: 2002	COOLING MW_c	LIGHTING & OTHER MW_e	HEATING MW_t		TOTAL POWER
			ELECTR.	BOILER	
JAN	7.17	0	2.15	4.30	11.46
FEB	6.14	0	1.74	3.48	9.62
MAR	7.29	0.15	1.67	3.80	11.08
APR	5.60	0.39	0.27	1.69	7.30
MAY	6.15	1.17	0.04	0.11	7.42
JUN	7.01	1.89	0.00	0.35	9.13
JUL	8.92	2.50	0.00	0.45	11.72
AUG	10.24	3.07	0.00	0.77	13.83
SEP	7.35	1.91	0.00	0.29	9.46
OCT	5.96	0.60	0.39	2.59	8.54
NOV	6.32	0	1.26	2.53	8.85
DEC	8.01	0	2.67	5.34	13.36

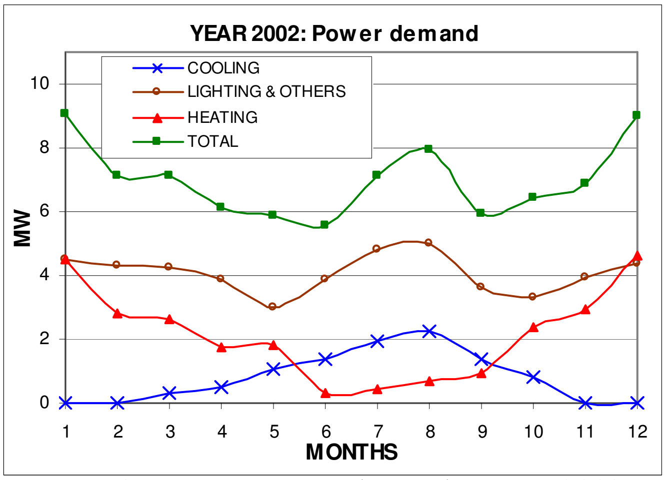


Fig.B.9: Lemnos power demand in MW, 2002, (typical day)

B.3 Rhodes processing procedure

Table B.10: Rhodes Island Population 1951 - 2001

Years	1951	1961	1971	1981	1991	2001	Increase % 91-01
Rhodes	59,087	63,954	66,609	87,833	98,181	117,007	19.17

*Table B.11: Existing stations in Rhodes**

Type of station	Number of Units	Type of fuel	Installed Power (MW)*	Net Power (MW)
Steam	1	Crude oil	15	14.2
	2	Crude oil	15	14.2
Piston Engines	1	Crude oil	12.28	11
	2	Crude oil	12.28	11
	3	Crude oil	23.41	22.8
	4	Crude oil	23.41	22.8
	5	Crude oil	23.41	22.8
Gas Turbines	1	Diesel	24	20
	2	Diesel	36	28
	3	Diesel	21.32	20
Total	**10**		**206.11**	**186.8**

* Up to 2005 is forecasted the manufacture of Wind parks of total force 5MW

<u>Thermoelectric Units of Rhodes</u>

The **Availability of Thermoelectric Units of** Rhodes in 2003 was 84.55% against 78.49% 2002.

The reduction is due to:

1. The reduction of the downtime for maintenance, reached 8.83% against 9.74% of year 2002.
2. The percentage because of the failure was decreased in the 6.25% against 10.33% of year 2002 and is emanated mainly from:
   - units piston engines and their equipment (12.07 %)
   - the conventional units of steam (3.79%) mainly from boiler's leakage
   - units Gas Turbines and their installation (0.38 %).
3. Others. The percentage of remaining causes minus damage reached in the 0.37%.

<u>Thermal efficiency</u>

The degree of output of Thermic Units of Rhodes in 2003 reached in the 35.10% against 2002 that were 33.26 %.

<u>Statistical data</u>

Based on the elements of PPC's data, the total installed power of autonomous stations of generation of electricity in the end of 1996 was 500MW and for this year, clean thermal production was 333.4GWh, while the consumption of electric energy were 321.7GWh. This energy is allocated in percentages as it appears in *Table B.12*:

Table B.12: Aborigines and foreigners tourists: arrivals and stayed overnight

YEARS	ABORIGINES	FOREIGNERS	TOTAL	% CHANGE	ABORIGINES	FOREIGNERS	TOTAL	% CHANGE
1970	90,374	170,705	261,079		615,892	1,513,787	2,129,68	
1971	71,755	282,131	353,886	35.40%	553,821	2,699,669	3,253,49	52.76%
1975	54,637	339,461	394,098		254,581	3,045,608	3,300,19	
1977	65,031	428,552	493,58		291,244	3,917,958	4,209,20	
1978	73,917	523,648	597,57	21.06	316,795	4,924,171	5,240,97	24.50%
1979	67,611	607,669	675,28	13%	280,015	5,956,521	6,236,54	18.99%
1980	74,014	643,555	717,57	6.26%	299,318	6,433,633	6,732,95	7.95%
1981	68,161	675,817	743,98	3.60%	288,689	6,819,253	7,107,94	-5.56%
1982	77,214	720,955	798,17	7.28%	324,044	7,220,579	7,544,62	6.14%
1983	94,886	643,433	738,32	-7.49%	411,441	6,277,630	6,689,07	-11.33%
1984	113,830	778,687	892,52	20.88%	565,516	7,307,866	7,873,38	17.70%
1985	111,521	898,246	1,009,77	13.13%	507,695	9,039,232	9,546,93	21.25%
1986	111,609	884,152	995,76	-1.38%	456,163	9,339,682	9,769,81	2.33%
1987	99,892	933,949	1,033,84	3.82%	393,260	9,426,352	9,819,61	0.50%
1988	113,331	892,193	1,005,52	-2.73%	485,819	9,165,687	9,651,51	-1.71%
1989	137,923	917,543	1,055,47	4.96%	616,931	9,449,060	10,066,0	4.29%
1990	141,659	992,009	1,133,67	7.40%	616,843	10,227,137	10,84340	7.72%
1991	135,331	929,889	1,065,22	-6.04 %	652,420	9,615,848	10,268,3	-5.31%
1992	170,662	1,149,965	1,320,63	23.97 %	738,687	12,008,514	12,747,2	24.14%
1993	160,113	1,114,527	1,277,64	-3.36 %	759,782	11,545,672	12,305,5	-3.59%
1994	167,250	1,306,704	1,473,95	15.37 %	679,934	13,509,263	14,207,2	15.46%
1995	194,594	1,247,426	1,442,02	-2.21%	825,268	12,366,359	13,191,6	-7.69%
1996	198,688	1,175,964	1,374,65	-4.28 %	874,966	11,360,689	12,235,7	-7.25%
1997	212,825	1,306,774	1,519,60	10.55 %	938,827	12,716,854	13,655,7	11.66%
1998	211,253	1,409,601	1,620,85	6.67 %	894,332	13,398,547	14,292,9	4.67%
1999	219,604	1,635,836	1,855,44	14.50 %	921,538	15,514,229	16,435,8	15 %
2000	240,567	1,664,975	1,905,54					
2001	212,000	1,623,302	1,835,30					
2002	247,323	1,662,819	1,910,14					

Table B.13: Percentage of different types of electrical consumption

Type of Use	Percentages % of electric energy
Domestic	38.9
Commercial	43.6
Industrial	4.4
Remaining	13.1

Table B.14: Rhodes energy demand in MWh, 2001, (typical day)

YEAR: *2001*	COOLING MWh_c	LIGHTING & OTHER MWh_e	HEATING MWh_t		TOTAL ENERGY
			ELECTR.	BOILER	
JAN	1,277	0	345	345	1,622
FEB	1,176	24	176	177	1,352
MAR	1,068	85	107	107	1,175
APR	1,210	339	72	73	1,283
MAY	1,558	499	93	94	1,651
JUN	1,899	684	38	38	1,937
JUL	2,388	955	48	48	2,436
AUG	2,541	1,093	76	76	2,617
SEP	2,054	760	82	82	2,136
OCT	1,646	477	99	99	1,745
NOV	1,143	34	126	125	1,269
DEC	1,521	0	426	426	1,947

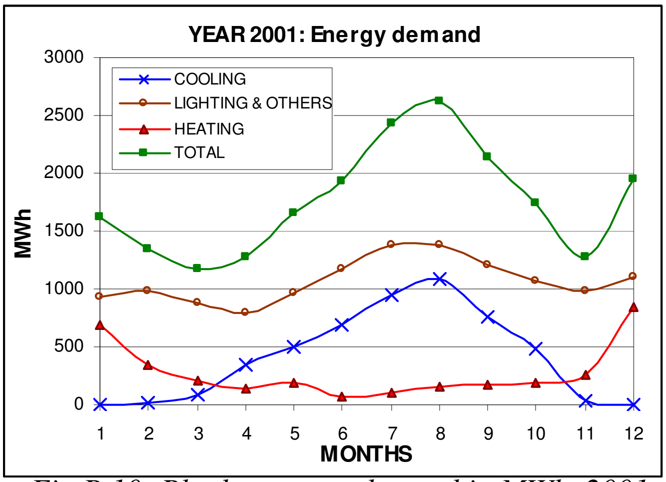


Fig.B.10: Rhodes energy demand in MWh, 2001, (typical day)

Table B.15: Rhodes power demand in MW, 2001, (typical day)

YEAR: *2001*	COOLING MW_c	LIGHTING & OTHER MW_e	HEATING MW_t		TOTAL POWER
			ELECTR.	BOILER	
JAN	53.22	0	14.370	14.37	67.59
FEB	48.99	0.98	7.350	7.35	56.34
MAR	44.50	3.56	4.450	4.45	48.95
APR	50.43	14.12	3.030	3.02	53.46
MAY	64.90	20.77	3.890	3.90	68.79
JUN	79.13	28.49	1.580	1.59	80.71
JUL	99.51	39.80	1.990	1.99	101.5
AUG	105.9	45.54	3.170	3.18	109.1
SEP	85.60	31.67	3.420	3.43	89.03
OCT	68.60	19.89	4.120	4.11	72.72
NOV	47.63	1.43	5.240	5.24	52.87
DEC	63.38	0	17.750	17.74	81.13

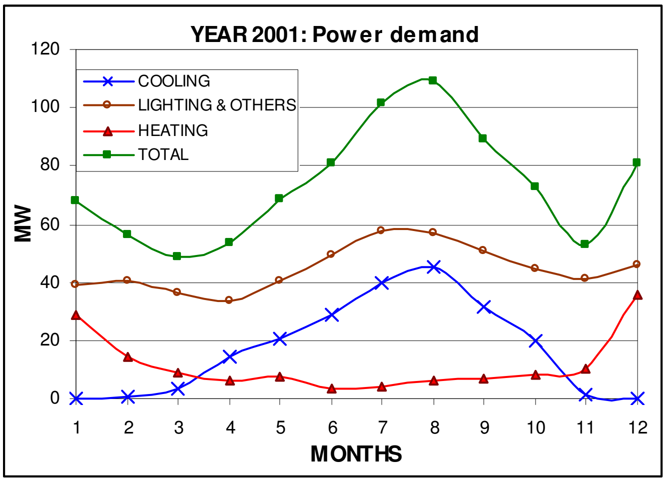


Fig.B.11: Rhodes power demand in MW, 2001, (typical day)

Table B.16: Rhodes energy demand in MWh, 2002, (typical day)

YEAR: *2002*	COOLING MWh_c	LIGHTING & OTHER MWh_e	HEATING MWh_t		TOTAL ENERGY
			ELECTR.	BOILER	
JAN	0	984	364	364	1,712
FEB	28	1,170	212	211	1,622
MAR	106	1,085	132	133	1,455
APR	442	1,042	95	94	1,674
MAY	521	1,009	97	98	1,725
JUN	696	1,199	39	38	1,973
JUL	992	1,439	50	49	2,531
AUG	1,123	1,410	78	79	2,689
SEP	776	1,238	84	84	2,182
OCT	485	1,086	100	101	1,771
NOV	35	1,000	128	128	1,291
DEC	0	1,094	425	426	1,944

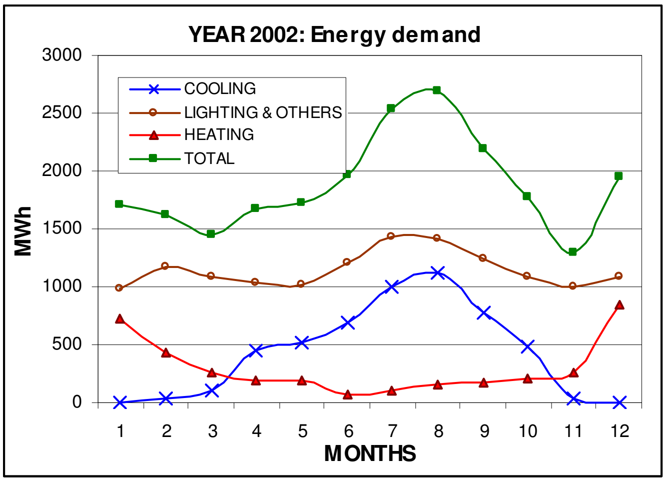


Fig.B.12 Rhodes energy demand in MWh, 2002, (typical day)

Table B.17: Rhodes power demand in MW ,
2002, (typical day)

YEAR: *2002*	COOLING MW_c	LIGHTING & OTHER MW_e	HEATING MW_t		TOTAL POWER
			ELECTR.	BOILER	
JAN	0	41	15.17	15.16	71.33
FEB	1.18	48.76	8.81	8.82	67.56
MAR	4.41	45.2	5.52	5.51	60.64
APR	18.42	43.42	3.95	3.95	69.74
MAY	21.69	42.03	4.07	4.07	71.86
JUN	29.01	49.96	1.61	1.61	82.20
JUL	41.35	59.96	2.09	2.05	105.4
AUG	46.78	58.75	3.27	3.26	112.1
SEP	32.34	51.58	3.50	3.49	90.91
OCT	20.19	45.26	4.18	4.18	73.80
NOV	1.45	41.67	5.34	5.32	53.79
DEC	0	45.57	17.72	17.72	81.01

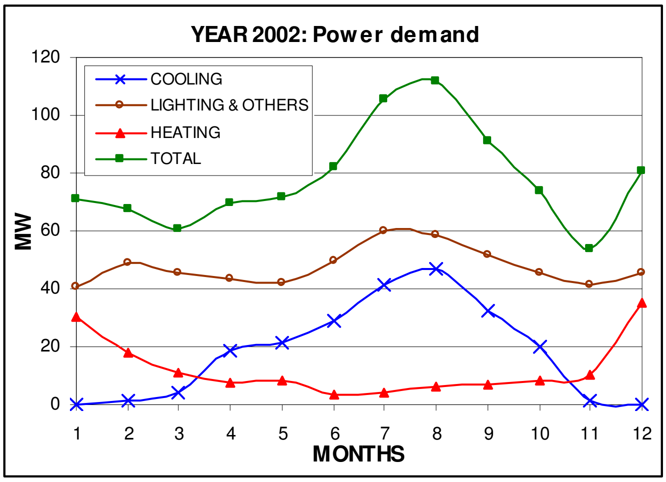


Fig.B.13: Rhodes power demand in MW,
2002, (typical day)

Table B.18: Total Electric Energy and Peak of Electric Power

Year	Total Electric Energy (GWh)	Peak of Electric Power (MW)
1988	249.5	56.5
1989	263	61.3
1990	276.8	66.0
1991	303.1	72.7
1992	345,9	81.1
1993	365.6	93.4
1994	396.4	95.4
1995	407	95.3
1996	415	99.3
1997	442	104.5
1998	472	120.0

B.4 Hotel in Rethimno-Crete processing procedure

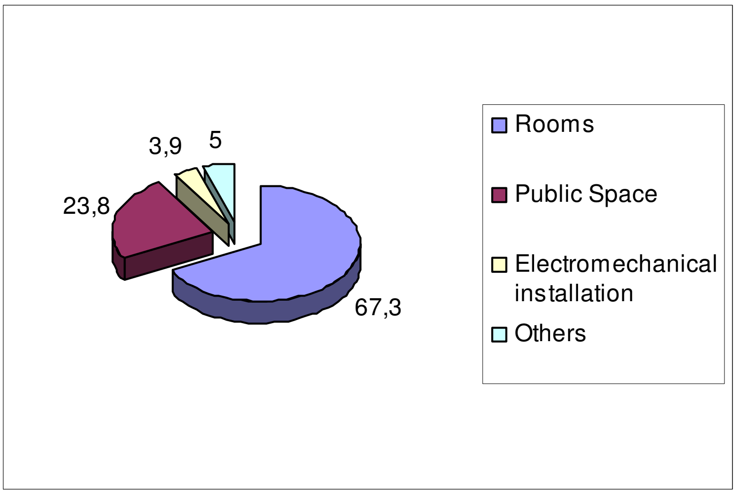


Fig.B.14: Space distribution per usage

<u>Electromechanical Installation</u>

Table B.19: Daily consumption of electricity per department in one typical day

DEPARTMENT	Consumption (kWh/ day)
Air conditioning	1,128
laundry	612
cuisine	361
pumps	143
restaurant	76
snack bar	71
mini market	12
remaining equipment	52

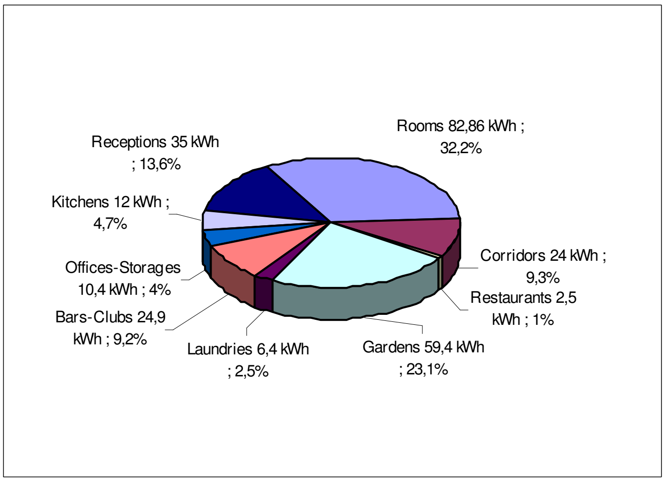


Fig.B.15: Electric energy distribution per space, in one typical day

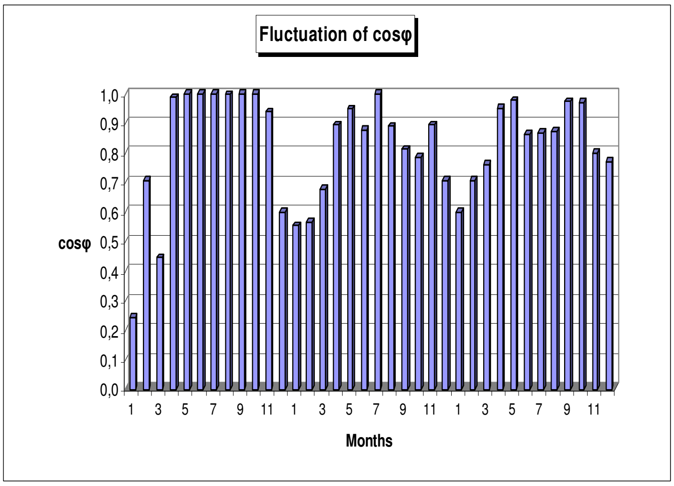


Fig.B.16: Fluctuation of cosφ, from 1/1/2000 to 1/12/2002

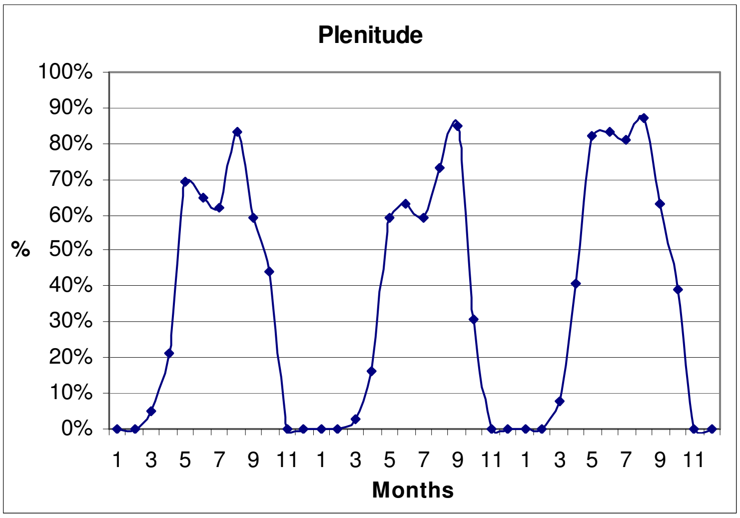


Fig.B.17: Hotel's plenitude from 1/1/2000 to 1/12/2002

Table B.20: Hotel in Rethimno energy demand in kWh, 2000, (typical day)

YEAR: *2000*	COOLING kWh_c	ELECTRICITY KWh_e	HEATING KWh_t	TOTAL ENERGY
JAN	0	931	0	931
FEB	0	3,375	0	3,375
MAR	149	1,713	16,708	18,570
APR	1,755	10,781	17,149	29,685
MAY	7,223	16,853	22,185	46,260
JUN	14,513	19,238	14,403	48,153
JUL	15,699	19,980	16,463	52,142
AUG	21,600	23,400	16,631	61,631
SEP	19,569	22,068	15,679	57,316
OCT	9,832	19,961	15,594	45,387
NOV	0	6,760	0	6,760
DEC	0	2,793	0	2,793

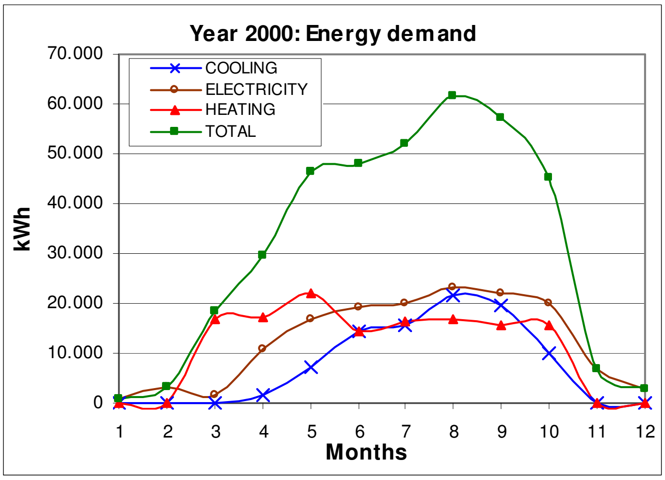


Fig.B.18: Hotel in Rethimno energy demand in kWh, 2000, (typical day)

Table B.21: Hotel in Rethimno power demand in kW, 2000, (typical day)

YEAR: 2000	COOLING kW_c	ELECTRICITY KW_e	HEATING KW_t	TOTAL POWER
JAN	0	1.29	0	1.29
FEB	0	4.69	0	4.69
MAR	0.21	2.38	23.21	25.79
APR	2.44	14.97	23.82	41.23
MAY	10.03	23.41	30.81	64.25
JUN	20.16	26.72	20.00	66.88
JUL	21.80	27.75	22.86	72.42
AUG	30.00	32.50	23.10	85.60
SEP	27.18	30.65	21.78	79.60
OCT	13.66	27.72	21.66	63.04
NOV	0	9.39	0	9.39
DEC	0	3.88	0	3.88

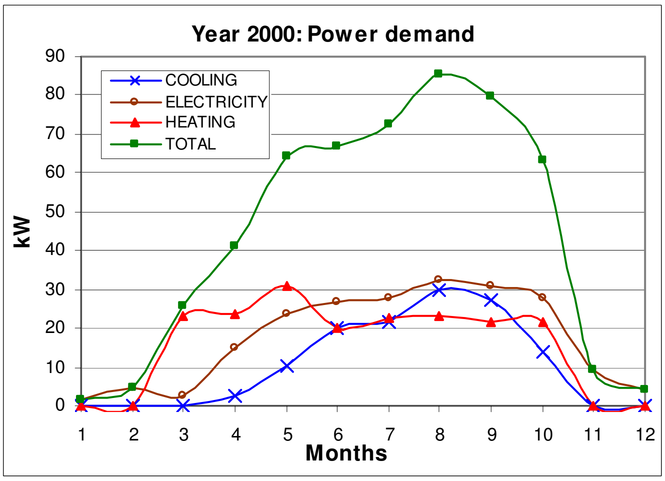


Fig.B.19: Hotel in Rethimno power demand in kW, 2000, (typical day)

Table B.22: Hotel in Rethimno energy demand in kWh, 2001, (typical day)

YEAR: ***2001***	**COOLING** kWh$_c$	**ELECTRICITY** KWh$_e$	**HEATING** KWh$_t$	*TOTAL ENERGY*
JAN	0	2.59	0	2.59
FEB	0	2.34	0	2.34
MAR	0.32	3.24	15.47	19.03
APR	1.55	11.38	27.29	40.22
MAY	5.91	18.70	30.94	55.55
JUN	10.61	21.54	26.91	59.06
JUL	15.26	25.99	27.41	68.66
AUG	20.66	27.39	24.62	72.67
SEP	20.25	27.96	26.91	75.13
OCT	7.12	21.36	30.94	59.42
NOV	0	14.52	0	14.52
DEC	0	1.75	0	1.75

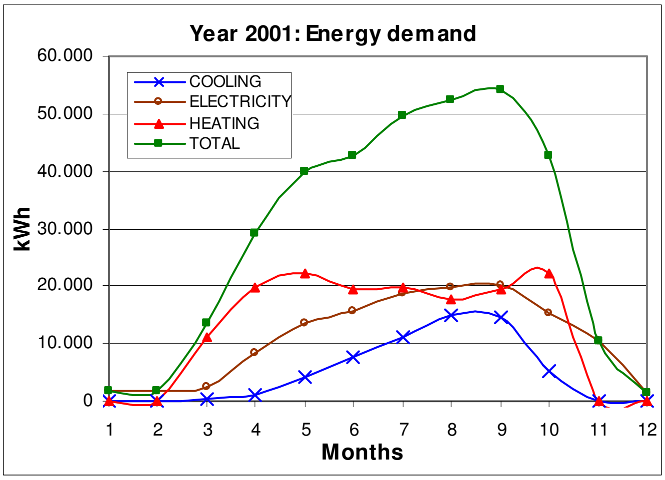


Fig.B.20: Hotel in Rethimno energy demand in kWh, 2001, (typical day)

Table B.23: Hotel in Rethimno power demand in kW, 2001, (typical day)

YEAR: *2001*	**COOLING** kW$_c$	**ELECTRICITY** KW$_e$	**HEATING** KW$_t$	*TOTAL POWER*
JAN	0	2.59	0	2.59
FEB	0	2.34	0	2.34
MAR	0.32	3.24	15.47	19.03
APR	1.55	11.38	27.29	40.22
MAY	5.91	18.70	30.94	55.55
JUN	10.61	21.54	26.91	59.06
JUL	15.26	25.99	27.41	68.66
AUG	20.66	27.39	24.62	72.67
SEP	20.25	27.96	26.91	75.13
OCT	7.12	21.36	30.94	59.42
NOV	0	14.52	0	14.52
DEC	0	1.75	0	1.75

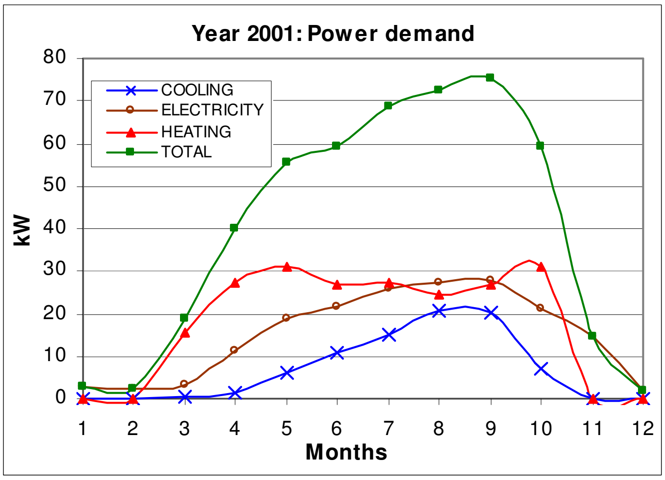


Fig.B.21: Hotel in Rethimno power demand in kW, 2001, (typical day)

B.5 Sani Beach Hotel Group processing procedure

1. Sani Beach Hotel

PPC's supply network: *two oil transformers, 1000 kVA*
Central cooling system:

- Total energy: 191,684Btu
- Months of operation: April-November

Central heating system:

- Total energy: 428,571kcal/h
- Number of boilers: 1
- Months of operation: 8
- Hours of operation: 24h
- Regulated temperature: 20°C

The *Sani Beach Hotel* energy results for the year 2000 are shown below (*Table B.24, B.25, Fig. B.20, B.21*):

Table B.24: Sani Beach Hotel energy demand in kWh, 2000, (typical day)

YEAR: 2000	COOLING kWh_e	LIGHTING & OTHER kWh_e	HEATING kWh_t		TOTAL ENERGY
			ELECT.	BOIL.	
JAN	118.4	815	0.6	11.8	945
FEB	191.2	1,062	0.8	11.5	1,265
MAR	233.8	1,046	0.2	12.1	1,292
APR	1,138.0	1,406	16.0	1,184.3	3,745
MAY	2,807.6	6,231	54.4	2,084.7	11,177
JUN	3,805.1	7,822	0.9	1,942.8	13,570
JUL	4,107.8	8,426	0.2	2,250.3	14,784
AUG	4,768.2	11,843	0.8	2,797.4	19,409
SEP	4,,043.0	8,301	52.0	1,869.0	14,265
OCT	2,987.7	7,848	29.3	1,591.2	12,456
NOV	1,429.5	7,389	8.5	1,120.5	9,948
DEC	524.3	2,454	0.7	11.7	2,990

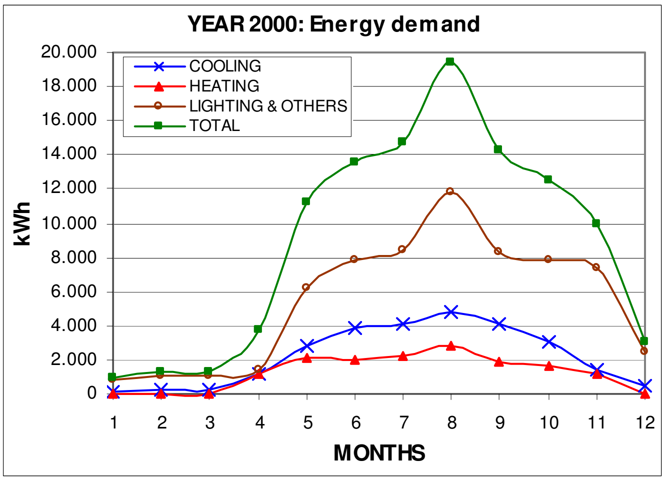


Fig.B.22: Sani Beach Hotel energy demand in kWh, 2000, (typical day)

Table B.25: Sani Beach Hotel power demand in kW, 2000, (typical day)

YEAR: *2000*	**COOLING** kW_e	**LIGHTING & OTHER** KW_E	**HEATING** KW_T		**TOTAL POWER**
			ELECTR.	BOILER	
JAN	9.9	41	0.0	1.4	52
FEB	15.9	52	0.0	1.4	69
MAR	19.5	50	0.0	1.4	71
APR	67.6	69	2.1	65.4	204
MAY	196.3	290	6.3	229.6	722
JUN	249.5	380	0.3	219.8	850
JUL	290.3	389	0.0	239.5	918
AUG	384.6	515	0.2	285.1	1,185
SEP	281.2	384	6.3	217	889
OCT	211.7	373	3.8	184.6	773
NOV	86.6	391	0.5	165.9	644
DEC	24.5	137	0.0	1.4	163

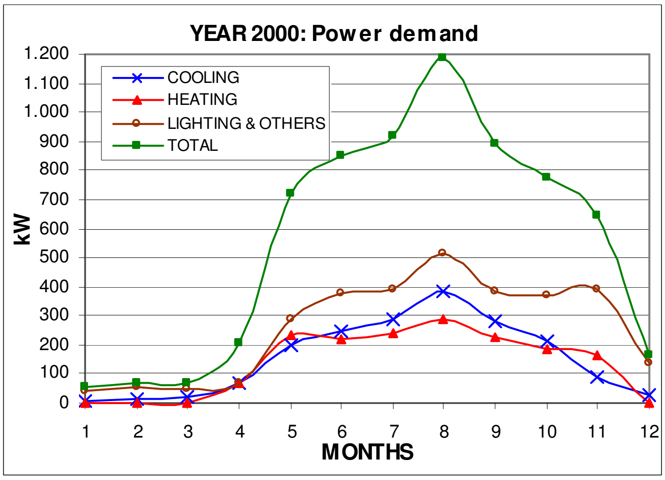


Fig.B.23: Sani Beach Hotel power demand in kW, 2000, (typical day)

The plenitude of *Sani Beach Hotel*, for the years 1998, 1999, 2000 and 2001 is shown below *(Table B.26, Fig. B.22)*:

Table B.26: Plenitude of Sani Beach Hotel, 1998-2001

YEAR	**1998**	**1999**	**2000**	**2001**
JAN	0%	0%	0%	0%
FEB	0%	0%	0%	0%
MAR	0%	0%	0%	0%
APR	10%	74%	23%	27%
MAY	76%	86%	69%	70%
JUN	86%	82%	88%	90%
JUL	91%	94%	95%	92%
AUG	96%	96%	97%	95%
SEP	79%	91%	92%	89%
OCT	48%	59%	72%	50%
NOV	0%	0%	17%	0%
DEC	0%	0%	0%	0%

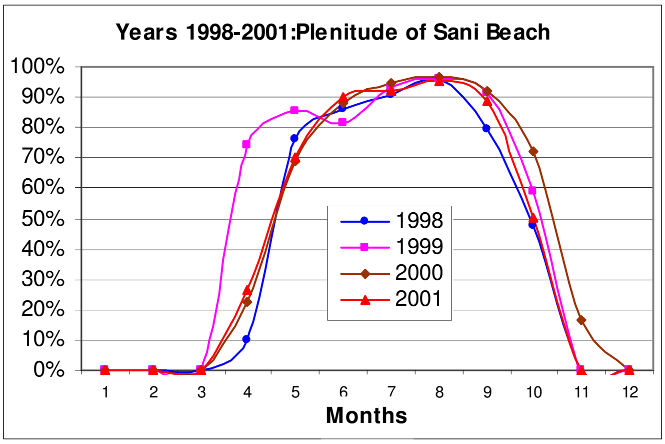


Fig. B.24: Plenitude of Sani Beach Hotel, 1998-2001

The Percentage of people stayed overnight at Sani Beach Hotel, for the years 1998, 1999, 2000 and 2001 is shown below (*Table B.27, Fig. B.23*):

Table B.27: Percentage of people stayed overnight, 1998-2001

YEAR	1998	1999	2000	2001
JAN	0%	0%	0%	0%
FEB	0%	0%	0%	0%
MAR	0%	0%	0%	0%
APR	2%	12%	4%	4%
MAY	14%	14%	12%	13%
JUN	17%	14%	16%	18%
JUL	20%	18%	19%	20%
AUG	21%	19%	20%	21%
SEP	16%	15%	15%	15%
OCT	9%	9%	11%	9%
NOV	0%	0%	3%	0%
DEC	0%	0%	0%	0%

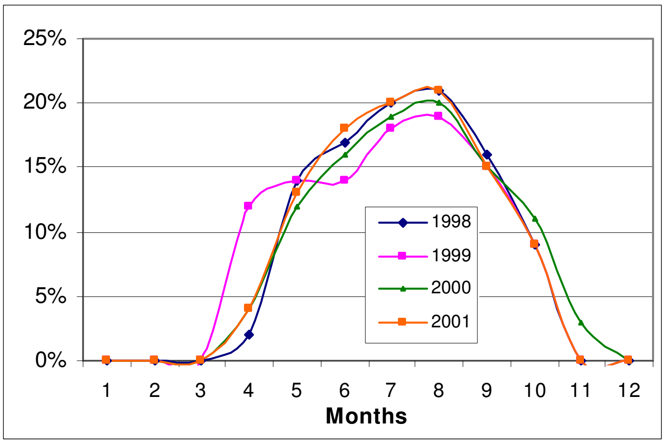


Fig. B.25: Percentage of people stayed overnight for the years 1998-2001

Electricity

Admissions

The constant needs in lighting of *Sani Beach hotel* were considered to follow the following percentages of total energy of each month.

JAN	FEB	MAR	APR	MAY	JUN	JUL	AUG	SEP	OCT	NOV	DEC
10%	10%	15%	20%	25%	30%	30%	30%	30%	25%	10%	10%

For the rest percentages of each month were taken into consideration the climatic conditions of the region (cloud, rainfall, etc) as well as the duration of the night in a typical day for each month.

Heating

Admissions

We consider the worst case, namely empty building, night, and no heat coming from the lights, etc. We assume that the heating energy is going both for central heating, and hot water. In the calculations we took in consider both the climatic conditions and the plenitude of the hotel.

The constant needs in heating of *Sani Beach hotel* were considered as the following percentages of total energy of each month.

JAN	FEB	MAR	APR	MAY	JUN	JUL	AUG	SEP	OCT	NOV	DEC
100%	100%	90%	60%	30%	0%	0%	0%	20%	70%	100%	100%

Cooling

Admissions

The constant needs in cooling of *Sani Beach hotel* were considered that they follow the percentages of total energy of each month, which are shown below.

JAN	FEB	MAR	APR	MAY	JUN	JUL	AUG	SEP	OCT	NOV	DEC
0%	0%	10%	40%	70%	100%	100%	100%	80%	30%	0%	0%

2. Porto Hotel

PPC's supply network: *dry transformer, 630kVA*
Central cooling system:

- Total energy: 190,000 Btu
- Months of operation: April-November

Central heating system:

- Total energy: 308,571kcal/h kcal/h
- Number of boilers: 4
- Months of operation: 8
- Hours of operation: 24h
- Regulated temperature: 20^oC

The *Porto Sani Hotel* energy results for the year 2000 are shown below (*Table B.28,B.29, Fig. B.24, B.25*):

Table B.28: Porto Sani Hotel energy demand in kWh, 2000, (typical day)

YEAR: *2000*	COOLING kWh_e	LIGHTING & OTHER kWh_e	HEATING kWh_t		TOTAL ENERGY
			ELECTR.	BOILER	
JAN	148.8	1,291	0.2	10.9	1,451
FEB	166.8	1,246	0.2	10.8	1,424
MAR	336.1	2,062	28.9	1,023.1	3,450
APR	1,241	3,095	37.0	1,533.0	5,906
MAY	2,265	3,038	30.0	2,018.0	7,351
JUN	2,737	3,024	0.0	1,924.0	7,684
JUL	3,088	3,392	0.0	2,111.0	8,591
AUG	3,666	5,108	0.0	2,511.0	11,283
SEP	2,442	4,301	30.0	1,939.0	8,711
OCT	1,300	4,212	8.0	849,0	6,369
NOV	153.7	2,006	0.3	10.6	2,171
DEC	211.3	935	0.7	10.5	1,158

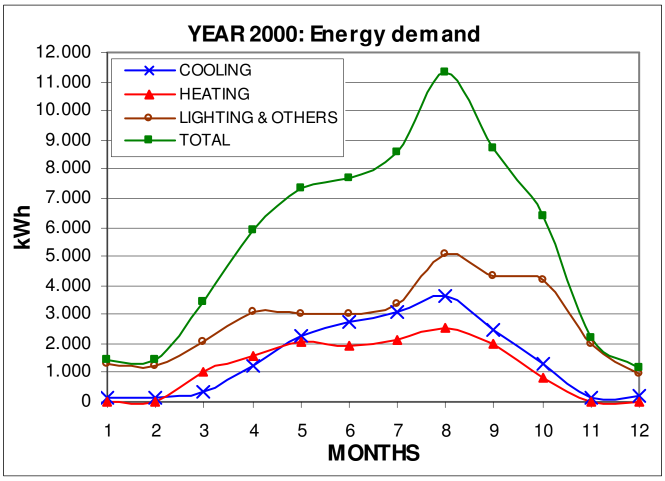


Fig.B.26: Porto Sani Hotel energy demand in kWh, 2000, (typical day)

Table B.29: Porto Sani Hotel power demand in kW, 2000, (typical day)

YEAR: *2000*	COOLING kW_e	LIGHTING & OTHER kW_e	HEATING kW_t		TOTAL POWER
			ELECT.	BOIL.	
JAN	12.4	65.6	0.0	0.6	78.6
FEB	13.9	62.6	0.1	0.5	77.1
MAR	28	100.4	3.0	52.2	183.7
APR	89	144	3.9	78.1	315.4
MAY	134.8	150.9	3.2	104.8	393.2
JUN	142	170	0.0	101	413.0
JUL	154.1	196.9	0.0	111	462
AUG	167.5	307.7	0.0	132	607
SEP	114.3	249.3	3.3	99.7	467
OCT	57.4	240.7	0.9	44.1	343.1
NOV	12.8	104.2	0.0	0.6	117.6
DEC	17.6	44.5	0.0	0.6	62.7

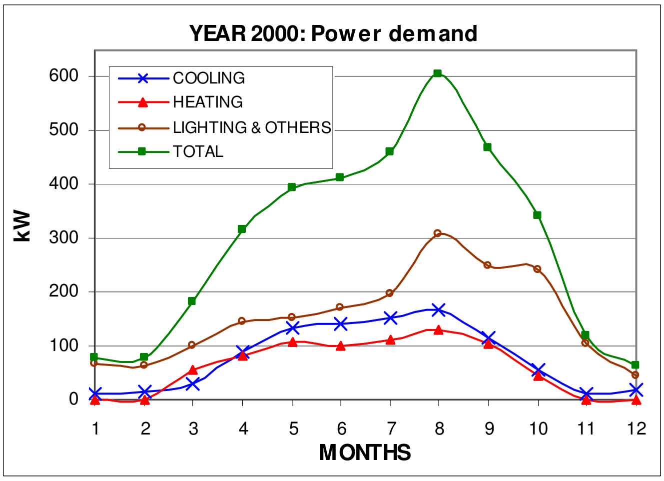


Fig.B.27: Porto Sani Hotel power demand in kW, 2000, (typical day)

The percentage of people stayed overnight at *Porto hotel*, for the years 1998, 1999, 2000 and 2001 is shown below (*Table B.30, Fig. B.26*).

Table B.30: Plenitude of Porto Hotel, 1998-2001

YEAR	1998	1999	2000	2001
JAN	0%	0%	0%	0%
FEB	0%	0%	0%	0%
MAR	0%	0%	0%	0%
APR	6%	1%	5%	5%
MAY	15%	14%	17%	17%
JUN	20%	17%	20%	20%
JUL	20%	18%	20%	20%
AUG	21%	19%	21%	21%
SEP	20%	19%	20%	20%
OCT	13%	9%	15%	16%
NOV	0%	0%	0%	0%
DEC	0%	0%	0%	0%

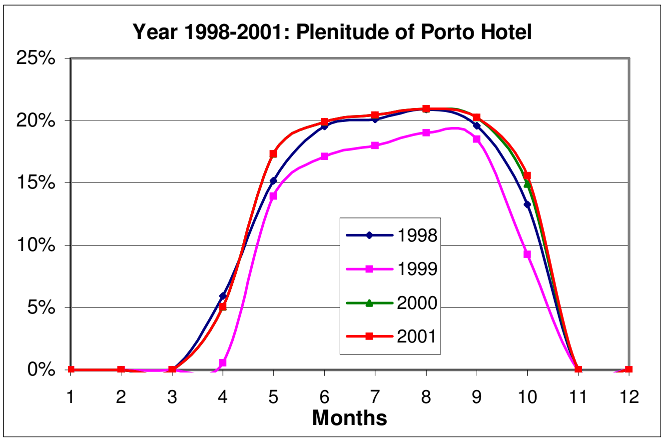


Fig.B.28: Plenitude of Sani Beach Hotel, 1998-2001

The Percentage of people stayed overnight at *Porto hotel*, for the years 1998, 1999, 2000 and 2001 is shown below (*Table B.31, Fig. B.27*)

Table B.31: Percentage of people stayed overnight, 1998-2001

YEAR	1998	1999	2000	2001
JAN	0%	0%	0%	0%
FEB	0%	0%	0%	0%
MAR	5%	3%	7%	4%
APR	12%	14%	14%	14%
MAY	16%	16%	16%	17%
JUN	19%	20%	19%	20%
JUL	21%	20%	19%	20%
AUG	16%	16%	15%	15%
SEP	11%	10%	10%	10%
OCT	5%	3%	7%	4%
NOV	0%	0%	0%	0%
DEC	0%	0%	0%	0%

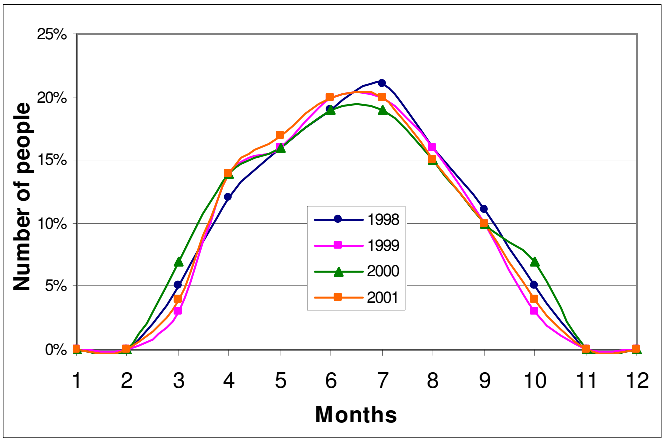


Fig.B.29: Percentage of people stayed overnight for the years 1998-2001

Electricity

Admissions

The constant needs in lighting of *Porto hotel* were considered to follow the following percentages of total energy of each month.

JAN	FEB	MAR	APR	MAY	JUN	JUL	AUG	SEP	OCT	NOV	DEC
10%	10%	15%	20%	25%	30%	30%	30%	30%	25%	10%	10%

For the rest percentages of each month were taken into consideration the climatic conditions of the region (cloud, rainfall, etc) as well as the duration of the night in a typical day for each month.

Heating

Admissions

We consider the worst case, namely empty building, night, and no heat coming from the lights, etc.

We assume that the heating energy is going both for central heating, and hot water. In the calculations we took in consider both the climatic conditions and the plenitude of the hotel.

The constant needs in heating of *Porto hotel* were considered as the following percentages of total energy of each month.

JAN	FEB	MAR	APR	MAY	JUN	JUL	AUG	SEP	OCT	NOV	DEC
100%	100%	90%	60%	30%	0%	0%	0%	20%	70%	100%	100%

Cooling

Admissions

The constant needs in cooling of *Porto hotel* were considered that they follow the percentages of total energy of each month, which are shown below.

JAN	FEB	MAR	APR	MAY	JUN	JUL	AUG	SEP	OCT	NOV	DEC
0%	0%	10%	40%	70%	100%	100%	100%	80%	30%	0%	0%

3. Club hotel

PPC's supply network:
dry transformer, 500kVA, and two oil transformers 500kVA, 250kVA each.
Central cooling system:

- Total energy: 191,684Btu
- Months of operation: April-November

Central heating system:

- Total energy: 300,000kcal/h
- Number of boilers: 1
- Months of operation: 8
- Hours of operation: 24h
- Regulated temperature: 20°C

The *Club Hotel* energy results for the year 2000 are shown below (*Table B.32, B.33, Fig. B.28, B.29*):

Table B.32: Club Hotel energy demand in kWh, 2000, (typical day)

YEAR: *2000*	COOLING kWh_e	LIGHTING & OTHER kWh_e	HEATING kWh_t		TOTAL ENERGY
			ELEC	BOIL.	
JAN	71	489	0.0	3.9	564
FEB	113.9	633	0.1	3.8	751
MAR	112	501	0.0	3.9	617
APR	91	442	0.0	4.0	537
MAY	812	1,421	7.0	1,720	3,961
JUN	2,028	2,399	0.0	1,687	6,114
JUL	2,215	3,199	0.0	1,754	7,167
AUG	2,773	4,347	0.0	1,896	9,016
SEP	1,866	3,515	6.0	1,692	7,079
OCT	809	3,775	3.0	1,676	6,263
NOV	480	2,826	1.0	1,258	4,565
DEC	112.7	527	0.3	3.6	644

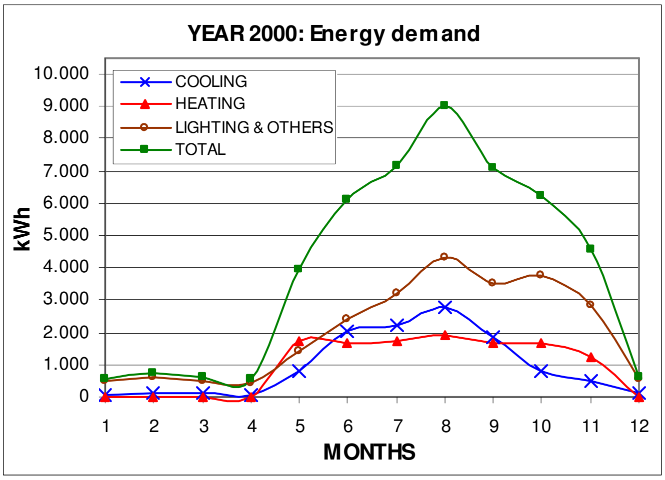


Fig.B.30: Club Hotel energy demand in kWh, 2000, (typical day)

Table B.33: Club Hotel power demand in kW, 2000, (typical day)

YEAR: *2000*	COOLING kW_e	LIGHTING & OTHER kW_e	HEATING kW_t		TOTAL POWER
			ELEC.	BOIL.	
JAN	5.9	24.4	0.0	0.4	30.7
FEB	9.5	31	0.0	0.4	40.8
MAR	9.3	23.9	0.0	0.4	33.6
APR	14.1	14.8	0.0	0.4	29.3
MAY	48.4	72.2	0.7	73.4	194.6
JUN	93.9	145.9	0.0	70	309.8
JUL	100.3	192.9	0.0	76.8	370
AUG	121.6	264.1	0.0	91.3	477
SEP	87.3	203.6	0.9	70.2	362
OCT	47.4	200.7	0.3	63.6	311.9
NOV	40	139	0.1	46.3	225.4
DEC	9.4	25.3	0.0	0.4	35.1

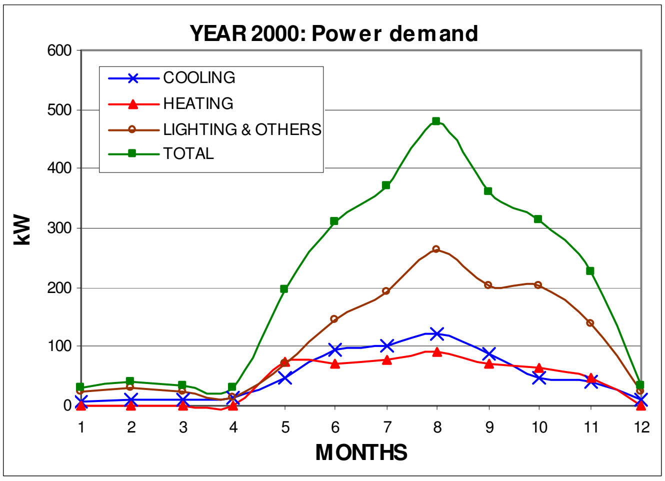


Fig.B.31: Club Hotel power demand in kW, 2000, (typical day)

The percentage of people stayed overnight at *Club hotel*, for the years 1998, 1999, 2000 and 2001 is shown below (*Table B.34, Fig. B.30*).

Table B.34: Plenitude of Club Hotel, 1998-2001

YEAR	1998	1999	2000	2001
JAN	0%	0%	0%	0%
FEB	0%	0%	0%	0%
MAR	0%	0%	0%	0%
APR	10%	74%	23%	27%
MAY	76%	86%	69%	70%
JUN	86%	82%	88%	90%
JUL	91%	94%	95%	92%
AUG	96%	96%	97%	95%
SEP	79%	91%	92%	89%
OCT	48%	59%	72%	50%
NOV	0%	0%	17%	0%
DEC	0%	0%	0%	0%

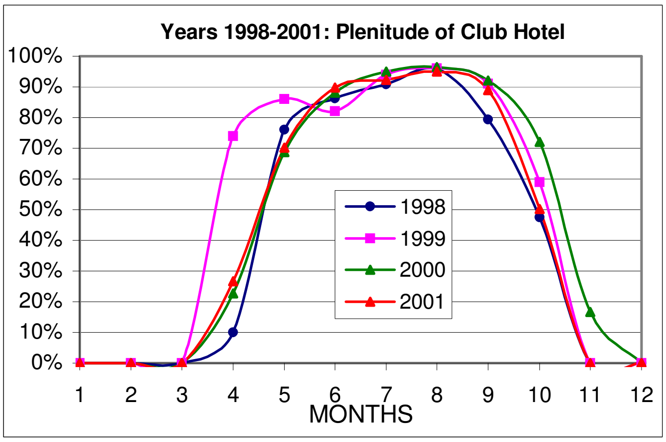


Fig.B.32: Plenitude of Club Hotel, 1998-2001

The percentage of people stayed overnight at *Club hotel*, for the years 1998, 1999, 2000 and 2001 is shown below (*Table B.35, Fig. B.31*)

Table B.35: Percentage of people stayed overnight, 1998-2001

YEAR	1998	1999	2000	2001
JAN	0%	0%	0%	0%
FEB	0%	0%	0%	0%
MAR	0%	0%	0%	0%
APR	0%	0%	0%	0%
MAY	9%	11%	10%	9%
JUN	20%	18%	19%	20%
JUL	22%	22%	21%	23%
AUG	22%	22%	21%	23%
SEP	18%	17%	17%	18%
OCT	9%	9%	12%	8%
NOV	0%	0%	8%	0%
DEC	0%	0%	0%	0%

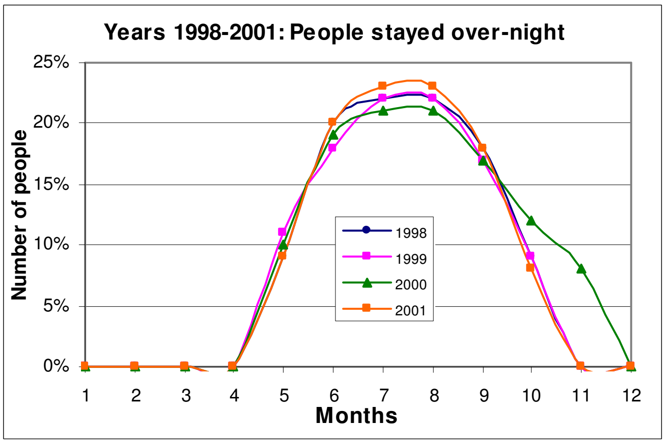


Fig.B.33: Percentage of people stayed overnight for the years 1998-2001

Electricity
Admissions
The constant needs in lighting of *Club hotel* were considered to follow the following percentages of total energy of each month.

JAN	FEB	MAR	APR	MAY	JUN	JUL	AUG	SEPT	OCT	NOV	DEC
10%	10%	15%	20%	25%	30%	30%	30%	30%	25%	10%	10%

For the rest percentages of each month were taken into consideration the climatic conditions of the region (cloud, rainfall, etc) as well as the duration of the night in a typical day for each month.

Heating
Admissions
We consider the worst case, namely empty building, night, and no heat coming from the lights, etc. We assume that the heating energy is going both for central heating, and hot water. In the calculations we took in consider both the climatic conditions and the plenitude of the hotel.

The constant needs in heating of *Club hotel* were considered as the following percentages of total energy of each month.

JAN	FEB	MAR	APR	MAY	JUN	JUL	AUG	SEP	OCT	NOV	DEC
100%	100%	90%	60%	30%	0%	0%	0%	20%	70%	100%	100%

Cooling
Admissions
The constant needs in cooling of *Club hotel* were considered that they follow the percentages of total energy of each month, which are shown below.

JAN	FEB	MAR	APR	MAY	JUN	JUL	AUG	SEP	OCT	NOV	DEC
0%	0%	10%	40%	70%	100%	100%	100%	80%	30%	0%	0%

APPENDIX C

C.1 Design point simulation program: Input file (4 engines)

```
Give Ambient pressure, Pa (90-110Pa)
101.3
Give Ambient temperature, Ta (233-323K)
288
Give Air mass flow, m
1
Give Compressor isentropic eff., nisc (0.77-0.91)
0.85
Give Heat exchanger cold eff., nche (0.7-0.90)
0.90
Give Combustion eff., ncc (0.985-0.995)
0.99
Give Compressor turbine isentropic eff., nist (0.83-0.97)
0.90
Give Compressor turbine isentropic eff., nisct (0.83-0.97)
0.89
Give Compressor turbine isentropic eff., nispt (0.83-0.97)
0.88
Give Heat exchanger hot eff., nhhe (0.7-0.90)
0.90
Give Intake pressure loss, DPinloss (0%-4%)
1
Give Heat exchanger cold pressure loss, DPcheloss(1%-4%)
1
Give Combustor pressure loss, DPccloss(3%-5%)
5
Give Heat exchanger hot pressure loss, DPhheloss (1%-4%)
1
Give Exhaust pressure loss, DPexhloss (0%-4%)
0
Give minimum Turbine Entry Temperature, TETmin (800- K)
900
Give maximum Turbine Entry Temperature, TETmax ( -2000K)
1500
Give minimum Compressor pressure ratio, Rcmin (5- )
5
Give maximum Compressor pressure ratio, Rcmax ( -35)
30
Give Inlet Mach number, Min
0
Give Exit Mach number, Mex
1
Give Compressor degradation, Pcde
0
Give Turbine degradation, Ptde
0
Give Exhaust degradation, Pexhde
0
Give FUEL (SynGas       (6.1)         :1,
           Biogas       (1)           :2,
           Biodiesel    (8)           :3,
           Residual Oil(40.3)         :4,
           Medium Heating Oil(41)     :5,
           Light Heating Oil(42.5)    :6,
           Kerozine(43.2)             :7,
           Russian Natural Gas(48.6) :8,
           Algerian Natural Gas(48.9):9)
8
Select Engine type (1ShaftGT    :1,
                    2ShaftGT    :2,
                    1ShaftGT HE :3,
                    2ShaftGT HE :4)
```

C.2 Design point simulation program: Output file (1-shaft)

One shaft simple cycle output file

```
ENGINE CONFIGURATION      FAR   MASS FLOW       P              T
    AMBIENT              0.00000   1.000   P1= 101.30   T1= 288.00
1-2 INTAKE               0.00000   1.000   P2= 100.29   T2= 288.00
2-3 COMPRESSOR           0.00000   1.000   P3=1002.87   T3= 603.34
3-4 PREMASS              0.00000   1.000   P4=1002.87   T4= 603.34
4-5 COMBUSTION CHAMBER   0.01128   1.011   P5= 952.73   T5=1000.00
5-6 MIXER                0.00000   1.011   P6= 952.73   T6=1000.00
6-7 COMPR & POWER TURB   0.00000   1.011   P7= 101.60   T7= 614.53
7-8 EXHAUST              0.00000   1.011   P8= 101.60   T8= 614.53
PERFORMANCE
UW=                      0.13
SW=                      0.1311
HI=                      0.55
mf=                      0.0113
sfc=                     0.0861
nth=                     0.2391
Rc=                      10.00
```

```
ENGINE CONFIGURATION      FAR   MASS FLOW       P              T
    AMBIENT              0.00000   1.000   P1= 101.30   T1= 288.00
1-2 INTAKE               0.00000   1.000   P2= 100.29   T2= 288.00
2-3 COMPRESSOR           0.00000   1.000   P3=1504.31   T3= 683.69
3-4 PREMASS              0.00000   1.000   P4=1504.31   T4= 683.69
4-5 COMBUSTION CHAMBER   0.00960   1.010   P5=1429.09   T5=1000.00
5-6 MIXER                0.00000   1.010   P6=1429.09   T6=1000.00
6-7 COMPR & POWER TURB   0.00000   1.010   P7= 101.60   T7= 564.97
7-8 EXHAUST              0.00000   1.010   P8= 101.60   T8= 564.97
PERFORMANCE
UW=                      0.11
SW=                      0.1071
HI=                      0.47
mf=                      0.0096
sfc=                     0.0897
nth=                     0.2295
Rc=                      15.00
```

```
ENGINE CONFIGURATION      FAR   MASS FLOW       P              T
    AMBIENT              0.00000   1.000   P1= 101.30   T1= 288.00
1-2 INTAKE               0.00000   1.000   P2= 100.29   T2= 288.00
2-3 COMPRESSOR           0.00000   1.000   P3=2005.74   T3= 746.61
3-4 PREMASS              0.00000   1.000   P4=2005.74   T4= 746.61
4-5 COMBUSTION CHAMBER   0.00829   1.008   P5=1905.45   T5=1000.00
5-6 MIXER                0.00000   1.008   P6=1905.45   T6=1000.00
6-7 COMPR & POWER TURB   0.00000   1.008   P7= 101.60   T7= 532.72
7-8 EXHAUST              0.00000   1.008   P8= 101.60   T8= 532.72
PERFORMANCE
UW=                      0.08
SW=                      0.0806
HI=                      0.40
mf=                      0.0083
sfc=                     0.1028
nth=                     0.2001
Rc=                      20.00
```

```
ENGINE CONFIGURATION      FAR   MASS FLOW       P              T
    AMBIENT              0.00000   1.000   P1= 101.30   T1= 288.00
1-2 INTAKE               0.00000   1.000   P2= 100.29   T2= 288.00
2-3 COMPRESSOR           0.00000   1.000   P3=1002.87   T3= 603.34
3-4 PREMASS              0.00000   1.000   P4=1002.87   T4= 603.34
4-5 COMBUSTION CHAMBER   0.01367   1.014   P5= 952.73   T5=1100.00
5-6 MIXER                0.00000   1.014   P6= 952.73   T6=1100.00
6-7 COMPR & POWER TURB   0.00000   1.014   P7= 101.60   T7= 675.98
7-8 EXHAUST              0.00000   1.014   P8= 101.60   T8= 675.98
PERFORMANCE
UW=                      0.18
SW=                      0.1770
HI=                      0.66
mf=                      0.0137
sfc=                     0.0772
nth=                     0.2665
Rc=                      10.00
```

```
ENGINE CONFIGURATION      FAR   MASS FLOW       P              T
    AMBIENT              0.00000   1.000   P1= 101.30   T1= 288.00
1-2 INTAKE               0.00000   1.000   P2= 100.29   T2= 288.00
2-3 COMPRESSOR           0.00000   1.000   P3=1504.31   T3= 683.69
3-4 PREMASS              0.00000   1.000   P4=1504.31   T4= 683.69
4-5 COMBUSTION CHAMBER   0.01199   1.012   P5=1429.09   T5=1100.00
```

```
5-6 MIXER              0.00000   1.012   P6=1429.09   T6=1100.00
6-7 COMPR & POWER TURB 0.00000   1.012   P7= 101.60   T7= 621.46
7-8 EXHAUST            0.00000   1.012   P8= 101.60   T8= 621.46
PERFORMANCE
UW=                    0.16
SW=                    0.1589
HI=                    0.58
mf=                    0.0120
sfc=                   0.0755
nth=                   0.2726
Rc=                    15.00
------------------------------------------------------------------------
ENGINE CONFIGURATION     FAR   MASS FLOW       P             T
------------------------------------------------------------------------
    AMBIENT            0.00000   1.000   P1= 101.30   T1= 288.00
1-2 INTAKE             0.00000   1.000   P2= 100.29   T2= 288.00
2-3 COMPRESSOR         0.00000   1.000   P3=2005.74   T3= 746.61
3-4 PREMASS            0.00000   1.000   P4=2005.74   T4= 746.61
4-5 COMBUSTION CHAMBER 0.01068   1.011   P5=1905.45   T5=1100.00
5-6 MIXER              0.00000   1.011   P6=1905.45   T6=1100.00
6-7 COMPR & POWER TURB 0.00000   1.011   P7= 101.60   T7= 585.99
7-8 EXHAUST            0.00000   1.011   P8= 101.60   T8= 585.99
PERFORMANCE
UW=                    0.14
SW=                    0.1361
HI=                    0.52
mf=                    0.0107
sfc=                   0.0784
nth=                   0.2623
Rc=                    20.00
------------------------------------------------------------------------
ENGINE CONFIGURATION     FAR   MASS FLOW       P             T
------------------------------------------------------------------------
    AMBIENT            0.00000   1.000   P1= 101.30   T1= 288.00
1-2 INTAKE             0.00000   1.000   P2= 100.29   T2= 288.00
2-3 COMPRESSOR         0.00000   1.000   P3=1002.87   T3= 603.34
3-4 PREMASS            0.00000   1.000   P4=1002.87   T4= 603.34
4-5 COMBUSTION CHAMBER 0.01606   1.016   P5= 952.73   T5=1200.00
5-6 MIXER              0.00000   1.016   P6= 952.73   T6=1200.00
6-7 COMPR & POWER TURB 0.00000   1.016   P7= 101.60   T7= 737.44
7-8 EXHAUST            0.00000   1.016   P8= 101.60   T8= 737.44
PERFORMANCE
UW=                    0.22
SW=                    0.2232
HI=                    0.78
mf=                    0.0161
sfc=                   0.0719
nth=                   0.2860
Rc=                    10.00
------------------------------------------------------------------------
ENGINE CONFIGURATION     FAR   MASS FLOW       P             T
------------------------------------------------------------------------
    AMBIENT            0.00000   1.000   P1= 101.30   T1= 288.00
1-2 INTAKE             0.00000   1.000   P2= 100.29   T2= 288.00
2-3 COMPRESSOR         0.00000   1.000   P3=1504.31   T3= 683.69
3-4 PREMASS            0.00000   1.000   P4=1504.31   T4= 683.69
4-5 COMBUSTION CHAMBER 0.01438   1.014   P5=1429.09   T5=1200.00
5-6 MIXER              0.00000   1.014   P6=1429.09   T6=1200.00
6-7 COMPR & POWER TURB 0.00000   1.014   P7= 101.60   T7= 677.96
7-8 EXHAUST            0.00000   1.014   P8= 101.60   T8= 677.96
PERFORMANCE
UW=                    0.21
SW=                    0.2109
HI=                    0.70
mf=                    0.0144
sfc=                   0.0682
nth=                   0.3018
Rc=                    15.00
------------------------------------------------------------------------
ENGINE CONFIGURATION     FAR   MASS FLOW       P             T
------------------------------------------------------------------------
    AMBIENT            0.00000   1.000   P1= 101.30   T1= 288.00
1-2 INTAKE             0.00000   1.000   P2= 100.29   T2= 288.00
2-3 COMPRESSOR         0.00000   1.000   P3=2005.74   T3= 746.61
3-4 PREMASS            0.00000   1.000   P4=2005.74   T4= 746.61
4-5 COMBUSTION CHAMBER 0.01307   1.013   P5=1905.45   T5=1200.00
5-6 MIXER              0.00000   1.013   P6=1905.45   T6=1200.00
6-7 COMPR & POWER TURB 0.00000   1.013   P7= 101.60   T7= 639.27
7-8 EXHAUST            0.00000   1.013   P8= 101.60   T8= 639.27
PERFORMANCE
UW=                    0.19
SW=                    0.1919
HI=                    0.64
mf=                    0.0131
sfc=                   0.0681
nth=                   0.3023
Rc=                    20.00
```

ENGINE CONFIGURATION	FAR	MASS FLOW	P	T
AMBIENT	0.00000	1.000	P1= 101.30	T1= 288.00
1-2 INTAKE	0.00000	1.000	P2= 100.29	T2= 288.00
2-3 COMPRESSOR	0.00000	1.000	P3=1002.87	T3= 603.34
3-4 PREMASS	0.00000	1.000	P4=1002.87	T4= 603.34
4-5 COMBUSTION CHAMBER	0.01844	1.018	P5= 952.73	T5=1300.00
5-6 MIXER	0.00000	1.018	P6= 952.73	T6=1300.00
6-7 COMPR & POWER TURB	0.00000	1.018	P7= 101.60	T7= 798.89
7-8 EXHAUST	0.00000	1.018	P8= 101.60	T8= 798.89
PERFORMANCE				
UW=	0.27			
SW=	0.2696			
HI=	0.90			
mf=	0.0184			
sfc=	0.0684			
nth=	0.3007			
Rc=	10.00			

ENGINE CONFIGURATION	FAR	MASS FLOW	P	T
AMBIENT	0.00000	1.000	P1= 101.30	T1= 288.00
1-2 INTAKE	0.00000	1.000	P2= 100.29	T2= 288.00
2-3 COMPRESSOR	0.00000	1.000	P3=1504.31	T3= 683.69
3-4 PREMASS	0.00000	1.000	P4=1504.31	T4= 683.69
4-5 COMBUSTION CHAMBER	0.01677	1.017	P5=1429.09	T5=1300.00
5-6 MIXER	0.00000	1.017	P6=1429.09	T6=1300.00
6-7 COMPR & POWER TURB	0.00000	1.017	P7= 101.60	T7= 734.45
7-8 EXHAUST	0.00000	1.017	P8= 101.60	T8= 734.45
PERFORMANCE				
UW=	0.26			
SW=	0.2632			
HI=	0.81			
mf=	0.0168			
sfc=	0.0637			
nth=	0.3229			
Rc=	15.00			

ENGINE CONFIGURATION	FAR	MASS FLOW	P	T
AMBIENT	0.00000	1.000	P1= 101.30	T1= 288.00
1-2 INTAKE	0.00000	1.000	P2= 100.29	T2= 288.00
2-3 COMPRESSOR	0.00000	1.000	P3=2005.74	T3= 746.61
3-4 PREMASS	0.00000	1.000	P4=2005.74	T4= 746.61
4-5 COMBUSTION CHAMBER	0.01545	1.015	P5=1905.45	T5=1300.00
5-6 MIXER	0.00000	1.015	P6=1905.45	T6=1300.00
6-7 COMPR & POWER TURB	0.00000	1.015	P7= 101.60	T7= 692.54
7-8 EXHAUST	0.00000	1.015	P8= 101.60	T8= 692.54
PERFORMANCE				
UW=	0.25			
SW=	0.2480			
HI=	0.75			
mf=	0.0155			
sfc=	0.0623			
nth=	0.3302			
Rc=	20.00			

C.3 Design point simulation program: Output file (2-shaft)

```
ENGINE CONFIGURATION      FAR   MASS FLOW      P              T
    AMBIENT             0.00000   1.000   P1= 101.30   T1= 288.00
1-2 INTAKE              0.00000   1.000   P2= 100.29   T2= 288.00
2-3 COMPRESSOR          0.00000   1.000   P3=1002.87   T3= 603.34
3-4 PREMASS             0.00000   1.000   P4=1002.87   T4= 603.34
4-5 COMBUSTION CHAMBER  0.01128   1.011   P5= 952.73   T5=1000.00
5-6 MIXER               0.00000   1.011   P6= 952.73   T6=1000.00
6-7 COMPR & POWER TURB  0.00000   1.011   P7= 220.33   T7= 727.36
7-8 POWER TURBINE       0.00000   1.011   P8= 101.60   T8= 614.82
8-9 EXHAUST             0.00000   1.011   P9= 101.60   T9= 614.82
PERFORMANCE
UW=                     0.13
SW=                     0.1308
HI=                     0.55
mf=                     0.0113
sfc=                    0.0863
nth=                    0.2385
Rc=                     10.00

ENGINE CONFIGURATION      FAR   MASS FLOW      P              T
    AMBIENT             0.00000   1.000   P1= 101.30   T1= 288.00
1-2 INTAKE              0.00000   1.000   P2= 100.29   T2= 288.00
2-3 COMPRESSOR          0.00000   1.000   P3=1504.31   T3= 683.69
3-4 PREMASS             0.00000   1.000   P4=1504.31   T4= 683.69
4-5 COMBUSTION CHAMBER  0.00960   1.010   P5=1429.09   T5=1000.00
5-6 MIXER               0.00000   1.010   P6=1429.09   T6=1000.00
6-7 COMPR & POWER TURB  0.00000   1.010   P7= 204.10   T7= 657.32
7-8 POWER TURBINE       0.00000   1.010   P8= 101.60   T8= 564.82
8-9 EXHAUST             0.00000   1.010   P9= 101.60   T9= 564.82
PERFORMANCE
UW=                     0.11
SW=                     0.1073
HI=                     0.47
mf=                     0.0096
sfc=                    0.0895
nth=                    0.2299
Rc=                     15.00

ENGINE CONFIGURATION      FAR   MASS FLOW      P              T
    AMBIENT             0.00000   1.000   P1= 101.30   T1= 288.00
1-2 INTAKE              0.00000   1.000   P2= 100.29   T2= 288.00
2-3 COMPRESSOR          0.00000   1.000   P3=2005.74   T3= 746.61
3-4 PREMASS             0.00000   1.000   P4=2005.74   T4= 746.61
4-5 COMBUSTION CHAMBER  0.00829   1.008   P5=1905.45   T5=1000.00
5-6 MIXER               0.00000   1.008   P6=1905.45   T6=1000.00
6-7 COMPR & POWER TURB  0.00000   1.008   P7= 178.08   T7= 602.31
7-8 POWER TURBINE       0.00000   1.008   P8= 101.60   T8= 532.98
8-9 EXHAUST             0.00000   1.008   P9= 101.60   T9= 532.98
PERFORMANCE
UW=                     0.08
SW=                     0.0803
HI=                     0.40
mf=                     0.0083
sfc=                    0.1032
nth=                    0.1993
Rc=                     20.00

ENGINE CONFIGURATION      FAR   MASS FLOW      P              T
    AMBIENT             0.00000   1.000   P1= 101.30   T1= 288.00
1-2 INTAKE              0.00000   1.000   P2= 100.29   T2= 288.00
2-3 COMPRESSOR          0.00000   1.000   P3=1002.87   T3= 603.34
3-4 PREMASS             0.00000   1.000   P4=1002.87   T4= 603.34
4-5 COMBUSTION CHAMBER  0.01367   1.014   P5= 952.73   T5=1100.00
5-6 MIXER               0.00000   1.014   P6= 952.73   T6=1100.00
6-7 COMPR & POWER TURB  0.00000   1.014   P7= 258.88   T7= 828.00
7-8 POWER TURBINE       0.00000   1.014   P8= 101.60   T8= 676.19
8-9 EXHAUST             0.00000   1.014   P9= 101.60   T9= 676.19
PERFORMANCE
UW=                     0.18
SW=                     0.1768
HI=                     0.66
mf=                     0.0137
sfc=                    0.0773
nth=                    0.2661
Rc=                     10.00

ENGINE CONFIGURATION      FAR   MASS FLOW      P              T
    AMBIENT             0.00000   1.000   P1= 101.30   T1= 288.00
1-2 INTAKE              0.00000   1.000   P2= 100.29   T2= 288.00
2-3 COMPRESSOR          0.00000   1.000   P3=1504.31   T3= 683.69
```

```
3-4 PREMASS              0.00000    1.000    P4=1504.31    T4= 683.69
4-5 COMBUSTION CHAMBER 0.01199    1.012    P5=1429.09    T5=1100.00
5-6 MIXER                0.00000    1.012    P6=1429.09    T6=1100.00
6-7 COMPR & POWER TURB 0.00000    1.012    P7= 256.02    T7= 758.13
7-8 POWER TURBINE        0.00000    1.012    P8= 101.60    T8= 620.59
8-9 EXHAUST              0.00000    1.012    P9= 101.60    T9= 620.59
PERFORMANCE
UW=                      0.16
SW=                      0.1599
HI=                      0.58
mf=                      0.0120
sfc=                     0.0750
nth=                     0.2744
Rc=                      15.00
-----------------------------------------------------------------------------
ENGINE CONFIGURATION       FAR    MASS FLOW       P              T
-----------------------------------------------------------------------------
    AMBIENT              0.00000    1.000    P1= 101.30    T1= 288.00
1-2 INTAKE               0.00000    1.000    P2= 100.29    T2= 288.00
2-3 COMPRESSOR           0.00000    1.000    P3=2005.74    T3= 746.61
3-4 PREMASS              0.00000    1.000    P4=2005.74    T4= 746.61
4-5 COMBUSTION CHAMBER 0.01068    1.011    P5=1905.45    T5=1100.00
5-6 MIXER                0.00000    1.011    P6=1905.45    T6=1100.00
6-7 COMPR & POWER TURB 0.00000    1.011    P7= 238.03    T7= 703.25
7-8 POWER TURBINE        0.00000    1.011    P8= 101.60    T8= 584.69
8-9 EXHAUST              0.00000    1.011    P9= 101.60    T9= 584.69
PERFORMANCE
UW=                      0.14
SW=                      0.1377
HI=                      0.52
mf=                      0.0107
sfc=                     0.0776
nth=                     0.2653
Rc=                      20.00
-----------------------------------------------------------------------------
ENGINE CONFIGURATION       FAR    MASS FLOW       P              T
-----------------------------------------------------------------------------
    AMBIENT              0.00000    1.000    P1= 101.30    T1= 288.00
1-2 INTAKE               0.00000    1.000    P2= 100.29    T2= 288.00
2-3 COMPRESSOR           0.00000    1.000    P3=1002.87    T3= 603.34
3-4 PREMASS              0.00000    1.000    P4=1002.87    T4= 603.34
4-5 COMBUSTION CHAMBER 0.01606    1.016    P5= 952.73    T5=1200.00
5-6 MIXER                0.00000    1.016    P6= 952.73    T6=1200.00
6-7 COMPR & POWER TURB 0.00000    1.016    P7= 294.68    T7= 928.64
7-8 POWER TURBINE        0.00000    1.016    P8= 101.60    T8= 737.77
8-9 EXHAUST              0.00000    1.016    P9= 101.60    T9= 737.77
PERFORMANCE
UW=                      0.22
SW=                      0.2228
HI=                      0.78
mf=                      0.0161
sfc=                     0.0721
nth=                     0.2855
Rc=                      10.00
-----------------------------------------------------------------------------
ENGINE CONFIGURATION       FAR    MASS FLOW       P              T
-----------------------------------------------------------------------------
    AMBIENT              0.00000    1.000    P1= 101.30    T1= 288.00
1-2 INTAKE               0.00000    1.000    P2= 100.29    T2= 288.00
2-3 COMPRESSOR           0.00000    1.000    P3=1504.31    T3= 683.69
3-4 PREMASS              0.00000    1.000    P4=1504.31    T4= 683.69
4-5 COMBUSTION CHAMBER 0.01438    1.014    P5=1429.09    T5=1200.00
5-6 MIXER                0.00000    1.014    P6=1429.09    T6=1200.00
6-7 COMPR & POWER TURB 0.00000    1.014    P7= 306.38    T7= 858.94
7-8 POWER TURBINE        0.00000    1.014    P8= 101.60    T8= 676.79
8-9 EXHAUST              0.00000    1.014    P9= 101.60    T9= 676.79
PERFORMANCE
UW=                      0.21
SW=                      0.2123
HI=                      0.70
mf=                      0.0144
sfc=                     0.0677
nth=                     0.3037
Rc=                      15.00
-----------------------------------------------------------------------------
ENGINE CONFIGURATION       FAR    MASS FLOW       P              T
-----------------------------------------------------------------------------
    AMBIENT              0.00000    1.000    P1= 101.30    T1= 288.00
1-2 INTAKE               0.00000    1.000    P2= 100.29    T2= 288.00
2-3 COMPRESSOR           0.00000    1.000    P3=2005.74    T3= 746.61
3-4 PREMASS              0.00000    1.000    P4=2005.74    T4= 746.61
4-5 COMBUSTION CHAMBER 0.01307    1.013    P5=1905.45    T5=1200.00
5-6 MIXER                0.00000    1.013    P6=1905.45    T6=1200.00
6-7 COMPR & POWER TURB 0.00000    1.013    P7= 298.58    T7= 804.19
7-8 POWER TURBINE        0.00000    1.013    P8= 101.60    T8= 637.12
8-9 EXHAUST              0.00000    1.013    P9= 101.60    T9= 637.12
PERFORMANCE
UW=                      0.19
```

```
SW=                       0.1944
HI=                       0.64
mf=                       0.0131
sfc=                      0.0672
nth=                      0.3062
Rc=                       20.00
--------------------------------------------------------------------------------------
ENGINE CONFIGURATION       FAR   MASS FLOW       P            T
--------------------------------------------------------------------------------------
    AMBIENT               0.00000   1.000   P1= 101.30   T1= 288.00
1-2 INTAKE                0.00000   1.000   P2= 100.29   T2= 288.00
2-3 COMPRESSOR            0.00000   1.000   P3=1002.87   T3= 603.34
3-4 PREMASS               0.00000   1.000   P4=1002.87   T4= 603.34
4-5 COMBUSTION CHAMBER    0.01844   1.018   P5= 952.73   T5=1300.00
5-6 MIXER                 0.00000   1.018   P6= 952.73   T6=1300.00
6-7 COMPR & POWER TURB    0.00000   1.018   P7= 327.76   T7=1029.28
7-8 POWER TURBINE         0.00000   1.018   P8= 101.60   T8= 799.51
8-9 EXHAUST               0.00000   1.018   P9= 101.60   T9= 799.51
PERFORMANCE
UW=                       0.27
SW=                       0.2688
HI=                       0.90
mf=                       0.0184
sfc=                      0.0686
nth=                      0.2999
Rc=                       10.00
--------------------------------------------------------------------------------------
ENGINE CONFIGURATION       FAR   MASS FLOW       P            T
--------------------------------------------------------------------------------------
    AMBIENT               0.00000   1.000   P1= 101.30   T1= 288.00
1-2 INTAKE                0.00000   1.000   P2= 100.29   T2= 288.00
2-3 COMPRESSOR            0.00000   1.000   P3=1504.31   T3= 683.69
3-4 PREMASS               0.00000   1.000   P4=1504.31   T4= 683.69
4-5 COMBUSTION CHAMBER    0.01677   1.017   P5=1429.09   T5=1300.00
5-6 MIXER                 0.00000   1.017   P6=1429.09   T6=1300.00
6-7 COMPR & POWER TURB    0.00000   1.017   P7= 354.49   T7= 959.74
7-8 POWER TURBINE         0.00000   1.017   P8= 101.60   T8= 733.28
8-9 EXHAUST               0.00000   1.017   P9= 101.60   T9= 733.28
PERFORMANCE
UW=                       0.26
SW=                       0.2645
HI=                       0.81
mf=                       0.0168
sfc=                      0.0634
nth=                      0.3246
Rc=                       15.00
--------------------------------------------------------------------------------------
ENGINE CONFIGURATION       FAR   MASS FLOW       P            T
--------------------------------------------------------------------------------------
    AMBIENT               0.00000   1.000   P1= 101.30   T1= 288.00
1-2 INTAKE                0.00000   1.000   P2= 100.29   T2= 288.00
2-3 COMPRESSOR            0.00000   1.000   P3=2005.74   T3= 746.61
3-4 PREMASS               0.00000   1.000   P4=2005.74   T4= 746.61
4-5 COMBUSTION CHAMBER    0.01545   1.015   P5=1905.45   T5=1300.00
5-6 MIXER                 0.00000   1.015   P6=1905.45   T6=1300.00
6-7 COMPR & POWER TURB    0.00000   1.015   P7= 358.27   T7= 905.12
7-8 POWER TURBINE         0.00000   1.015   P8= 101.60   T8= 690.00
8-9 EXHAUST               0.00000   1.015   P9= 101.60   T9= 690.00
PERFORMANCE
UW=                       0.25
SW=                       0.2510
HI=                       0.75
mf=                       0.0155
sfc=                      0.0616
nth=                      0.3341
Rc=                       20.00
```

C.4 Design point simulation program: Output file (1-shaft HE)

```
--------------------------------------------------------------------------------
ENGINE CONFIGURATION      FAR   MASS FLOW      P             T
--------------------------------------------------------------------------------
    AMBIENT              0.00000   1.000   P1= 101.30   T1= 288.00
1-2 INTAKE               0.00000   1.000   P2= 100.29   T2= 288.00
2-3 COMPRESSOR           0.00000   1.000   P3=1002.87   T3= 603.34
3-4 PREMASS              0.00000   1.000   P4=1002.87   T4= 603.34
4-5 HEAT EXCHANGER COLD  0.00000   1.000   P5= 992.84   T5= 615.73
5-6 BURNER               0.01102   1.011   P6= 943.20   T6=1000.00
6-7 MIXER                0.00000   1.011   P7= 943.20   T7=1000.00
7-8 COMPR & POWER TURB   0.00000   1.011   P8= 102.62   T8= 617.11
8-9 HEAT EXCHANGER HOT   0.00000   1.011   P9= 101.60   T9= 604.72
9-10 EXHAUST             0.00000   1.011  P10= 101.60  T10= 604.72
PERFORMANCE
UW=                     0.13
SW=                     0.1280
HI=                     0.54
mf=                     0.0110
sfc=                    0.0861
nth=                    0.2389
Rc=                     10.00
--------------------------------------------------------------------------------
ENGINE CONFIGURATION      FAR   MASS FLOW      P             T
--------------------------------------------------------------------------------
    AMBIENT              0.00000   1.000   P1= 101.30   T1= 288.00
1-2 INTAKE               0.00000   1.000   P2= 100.29   T2= 288.00
2-3 COMPRESSOR           0.00000   1.000   P3=1504.31   T3= 683.69
3-4 PREMASS              0.00000   1.000   P4=1504.31   T4= 683.69
4-5 HEAT EXCHANGER COLD  0.00000   1.000   P5=1489.26   T5= 578.93
5-6 BURNER               0.01179   1.012   P6=1414.80   T6=1000.00
6-7 MIXER                0.00000   1.012   P7=1414.80   T7=1000.00
7-8 COMPR & POWER TURB   0.00000   1.012   P8= 102.62   T8= 567.29
8-9 HEAT EXCHANGER HOT   0.00000   1.012   P9= 101.60   T9= 672.05
9-10 EXHAUST             0.00000   1.012  P10= 101.60  T10= 672.05
PERFORMANCE
UW=                     0.00
SW=                     0.0000
HI=                     0.00
mf=                     0.0118
sfc=                    0.0000
nth=                    0.0000
Rc=                     15.00
--------------------------------------------------------------------------------
ENGINE CONFIGURATION      FAR   MASS FLOW      P             T
--------------------------------------------------------------------------------
    AMBIENT              0.00000   1.000   P1= 101.30   T1= 288.00
1-2 INTAKE               0.00000   1.000   P2= 100.29   T2= 288.00
2-3 COMPRESSOR           0.00000   1.000   P3=2005.74   T3= 746.61
3-4 PREMASS              0.00000   1.000   P4=2005.74   T4= 746.61
4-5 HEAT EXCHANGER COLD  0.00000   1.000   P5=1985.68   T5= 556.06
5-6 BURNER               0.01227   1.012   P6=1886.40   T6=1000.00
6-7 MIXER                0.00000   1.012   P7=1886.40   T7=1000.00
7-8 COMPR & POWER TURB   0.00000   1.012   P8= 102.62   T8= 534.89
8-9 HEAT EXCHANGER HOT   0.00000   1.012   P9= 101.60   T9= 725.44
9-10 EXHAUST             0.00000   1.012  P10= 101.60  T10= 725.44
PERFORMANCE
UW=                     0.00
SW=                     0.0000
HI=                     0.00
mf=                     0.0123
sfc=                    0.0000
nth=                    0.0000
Rc=                     20.00
--------------------------------------------------------------------------------
ENGINE CONFIGURATION      FAR   MASS FLOW      P             T
--------------------------------------------------------------------------------
    AMBIENT              0.00000   1.000   P1= 101.30   T1= 288.00
1-2 INTAKE               0.00000   1.000   P2= 100.29   T2= 288.00
2-3 COMPRESSOR           0.00000   1.000   P3=1002.87   T3= 603.34
3-4 PREMASS              0.00000   1.000   P4=1002.87   T4= 603.34
4-5 HEAT EXCHANGER COLD  0.00000   1.000   P5= 992.84   T5= 671.27
5-6 BURNER               0.01225   1.012   P6= 943.20   T6=1100.00
6-7 MIXER                0.00000   1.012   P7= 943.20   T7=1100.00
7-8 COMPR & POWER TURB   0.00000   1.012   P8= 102.62   T8= 678.82
8-9 HEAT EXCHANGER HOT   0.00000   1.012   P9= 101.60   T9= 610.89
9-10 EXHAUST             0.00000   1.012  P10= 101.60  T10= 610.89
PERFORMANCE
UW=                     0.17
SW=                     0.1730
HI=                     0.60
mf=                     0.0123
sfc=                    0.0708
nth=                    0.2906
Rc=                     10.00
```

```
ENGINE CONFIGURATION      FAR    MASS FLOW      P              T
    AMBIENT              0.00000    1.000    P1= 101.30    T1= 288.00
1-2 INTAKE               0.00000    1.000    P2= 100.29    T2= 288.00
2-3 COMPRESSOR           0.00000    1.000    P3=1504.31    T3= 683.69
3-4 PREMASS              0.00000    1.000    P4=1504.31    T4= 683.69
4-5 HEAT EXCHANGER COLD  0.00000    1.000    P5=1489.26    T5= 629.99
5-6 BURNER               0.01311    1.013    P6=1414.80    T6=1100.00
6-7 MIXER                0.00000    1.013    P7=1414.80    T7=1100.00
7-8 COMPR & POWER TURB   0.00000    1.013    P8= 102.62    T8= 624.02
8-9 HEAT EXCHANGER HOT   0.00000    1.013    P9= 101.60    T9= 677.72
9-10 EXHAUST             0.00000    1.013   P10= 101.60   T10= 677.72
PERFORMANCE
UW=                     0.00
SW=                     0.0000
HI=                     0.00
mf=                     0.0131
sfc=                    0.0000
nth=                    0.0000
Rc=                     15.00
```

```
ENGINE CONFIGURATION      FAR    MASS FLOW      P              T
    AMBIENT              0.00000    1.000    P1= 101.30    T1= 288.00
1-2 INTAKE               0.00000    1.000    P2= 100.29    T2= 288.00
2-3 COMPRESSOR           0.00000    1.000    P3=2005.74    T3= 746.61
3-4 PREMASS              0.00000    1.000    P4=2005.74    T4= 746.61
4-5 HEAT EXCHANGER COLD  0.00000    1.000    P5=1985.68    T5= 604.20
5-6 BURNER               0.01365    1.014    P6=1886.40    T6=1100.00
6-7 MIXER                0.00000    1.014    P7=1886.40    T7=1100.00
7-8 COMPR & POWER TURB   0.00000    1.014    P8= 102.62    T8= 588.38
8-9 HEAT EXCHANGER HOT   0.00000    1.014    P9= 101.60    T9= 730.79
9-10 EXHAUST             0.00000    1.014   P10= 101.60   T10= 730.79
PERFORMANCE
UW=                     0.00
SW=                     0.0000
HI=                     0.00
mf=                     0.0137
sfc=                    0.0000
nth=                    0.0000
Rc=                     20.00
```

```
ENGINE CONFIGURATION      FAR    MASS FLOW      P              T
    AMBIENT              0.00000    1.000    P1= 101.30    T1= 288.00
1-2 INTAKE               0.00000    1.000    P2= 100.29    T2= 288.00
2-3 COMPRESSOR           0.00000    1.000    P3=1002.87    T3= 603.34
3-4 PREMASS              0.00000    1.000    P4=1002.87    T4= 603.34
4-5 HEAT EXCHANGER COLD  0.00000    1.000    P5= 992.84    T5= 726.81
5-6 BURNER               0.01348    1.013    P6= 943.20    T6=1200.00
6-7 MIXER                0.00000    1.013    P7= 943.20    T7=1200.00
7-8 COMPR & POWER TURB   0.00000    1.013    P8= 102.62    T8= 740.53
8-9 HEAT EXCHANGER HOT   0.00000    1.013    P9= 101.60    T9= 617.06
9-10 EXHAUST             0.00000    1.013   P10= 101.60   T10= 617.06
PERFORMANCE
UW=                     0.22
SW=                     0.2182
HI=                     0.66
mf=                     0.0135
sfc=                    0.0618
nth=                    0.3331
Rc=                     10.00
```

```
ENGINE CONFIGURATION      FAR    MASS FLOW      P              T
    AMBIENT              0.00000    1.000    P1= 101.30    T1= 288.00
1-2 INTAKE               0.00000    1.000    P2= 100.29    T2= 288.00
2-3 COMPRESSOR           0.00000    1.000    P3=1504.31    T3= 683.69
3-4 PREMASS              0.00000    1.000    P4=1504.31    T4= 683.69
4-5 HEAT EXCHANGER COLD  0.00000    1.000    P5=1489.26    T5= 681.05
5-6 BURNER               0.01443    1.014    P6=1414.80    T6=1200.00
6-7 MIXER                0.00000    1.014    P7=1414.80    T7=1200.00
7-8 COMPR & POWER TURB   0.00000    1.014    P8= 102.62    T8= 680.75
8-9 HEAT EXCHANGER HOT   0.00000    1.014    P9= 101.60    T9= 683.40
9-10 EXHAUST             0.00000    1.014   P10= 101.60   T10= 683.40
PERFORMANCE
UW=                     0.00
SW=                     0.0000
HI=                     0.00
mf=                     0.0144
sfc=                    0.0000
nth=                    0.0000
Rc=                     15.00
```

```
ENGINE CONFIGURATION      FAR    MASS FLOW      P              T
    AMBIENT              0.00000    1.000    P1= 101.30    T1= 288.00
1-2 INTAKE               0.00000    1.000    P2= 100.29    T2= 288.00
2-3 COMPRESSOR           0.00000    1.000    P3=2005.74    T3= 746.61
```

```
3-4 PREMASS               0.00000    1.000    P4=2005.74    T4= 746.61
4-5 HEAT EXCHANGER COLD 0.00000    1.000    P5=1985.68    T5= 652.34
5-6 BURNER                0.01503    1.015    P6=1886.40    T6=1200.00
6-7 MIXER                 0.00000    1.015    P7=1886.40    T7=1200.00
7-8 COMPR & POWER TURB    0.00000    1.015    P8= 102.62    T8= 641.87
8-9 HEAT EXCHANGER HOT    0.00000    1.015    P9= 101.60    T9= 736.14
9-10 EXHAUST              0.00000    1.015   P10= 101.60   T10= 736.14
PERFORMANCE
UW=                     0.00
SW=                     0.0000
HI=                     0.00
mf=                     0.0150
sfc=                    0.0000
nth=                    0.0000
Rc=                     20.00
-----------------------------------------------------------------------------------
ENGINE CONFIGURATION      FAR    MASS FLOW       P              T
-----------------------------------------------------------------------------------
    AMBIENT               0.00000    1.000    P1= 101.30    T1= 288.00
1-2 INTAKE                0.00000    1.000    P2= 100.29    T2= 288.00
2-3 COMPRESSOR            0.00000    1.000    P3=1002.87    T3= 603.34
3-4 PREMASS               0.00000    1.000    P4=1002.87    T4= 603.34
4-5 HEAT EXCHANGER COLD 0.00000    1.000    P5= 992.84    T5= 782.35
5-6 BURNER                0.01471    1.015    P6= 943.20    T6=1300.00
6-7 MIXER                 0.00000    1.015    P7= 943.20    T7=1300.00
7-8 COMPR & POWER TURB    0.00000    1.015    P8= 102.62    T8= 802.24
8-9 HEAT EXCHANGER HOT    0.00000    1.015    P9= 101.60    T9= 623.23
9-10 EXHAUST              0.00000    1.015   P10= 101.60   T10= 623.23
PERFORMANCE
UW=                     0.26
SW=                     0.2635
HI=                     0.71
mf=                     0.0147
sfc=                    0.0558
nth=                    0.3686
Rc=                     10.00
-----------------------------------------------------------------------------------
ENGINE CONFIGURATION      FAR    MASS FLOW       P              T
-----------------------------------------------------------------------------------
    AMBIENT               0.00000    1.000    P1= 101.30    T1= 288.00
1-2 INTAKE                0.00000    1.000    P2= 100.29    T2= 288.00
2-3 COMPRESSOR            0.00000    1.000    P3=1504.31    T3= 683.69
3-4 PREMASS               0.00000    1.000    P4=1504.31    T4= 683.69
4-5 HEAT EXCHANGER COLD 0.00000    1.000    P5=1489.26    T5= 732.10
5-6 BURNER                0.01576    1.016    P6=1414.80    T6=1300.00
6-7 MIXER                 0.00000    1.016    P7=1414.80    T7=1300.00
7-8 COMPR & POWER TURB    0.00000    1.016    P8= 102.62    T8= 737.48
8-9 HEAT EXCHANGER HOT    0.00000    1.016    P9= 101.60    T9= 689.07
9-10 EXHAUST              0.00000    1.016   P10= 101.60   T10= 689.07
PERFORMANCE
UW=                     0.26
SW=                     0.2590
HI=                     0.77
mf=                     0.0158
sfc=                    0.0608
nth=                    0.3382
Rc=                     15.00
-----------------------------------------------------------------------------------
ENGINE CONFIGURATION      FAR    MASS FLOW       P              T
-----------------------------------------------------------------------------------
    AMBIENT               0.00000    1.000    P1= 101.30    T1= 288.00
1-2 INTAKE                0.00000    1.000    P2= 100.29    T2= 288.00
2-3 COMPRESSOR            0.00000    1.000    P3=2005.74    T3= 746.61
3-4 PREMASS               0.00000    1.000    P4=2005.74    T4= 746.61
4-5 HEAT EXCHANGER COLD 0.00000    1.000    P5=1985.68    T5= 700.48
5-6 BURNER                0.01642    1.016    P6=1886.40    T6=1300.00
6-7 MIXER                 0.00000    1.016    P7=1886.40    T7=1300.00
7-8 COMPR & POWER TURB    0.00000    1.016    P8= 102.62    T8= 695.36
8-9 HEAT EXCHANGER HOT    0.00000    1.016    P9= 101.60    T9= 741.49
9-10 EXHAUST              0.00000    1.016   P10= 101.60   T10= 741.49
PERFORMANCE
UW=                     0.00
SW=                     0.0000
HI=                     0.00
mf=                     0.0164
sfc=                    0.0000
nth=                    0.0000
Rc=                     20.00
```

C.5 Design point simulation program: Output file (2-shaft HE)

```
ENGINE CONFIGURATION      FAR    MASS FLOW      P              T
    AMBIENT              0.00000   1.000    P1= 101.30    T1= 288.00
1-2 INTAKE               0.00000   1.000    P2= 100.29    T2= 288.00
2-3 COMPRESSOR           0.00000   1.000    P3=1002.87    T3= 603.34
3-4 PREMASS              0.00000   1.000    P4=1002.87    T4= 603.34
4-5 HEAT EXCHANGER COLD  0.00000   1.000    P5= 992.84    T5= 543.92
5-6 BURNER               0.01252   1.013    P6= 943.20    T6=1000.00
6-7 MIXER                0.00000   1.013    P7= 943.20    T7=1000.00
7-8 COMPRESSOR TURBINE   0.00000   1.013    P8= 421.28    T8= 727.69
8-9 POWER TURBINE        0.00000   1.013    P9= 102.62    T9= 537.32
9-10 HEAT EXCHANGER HOT  0.00000   1.013   P10= 101.60   T10= 596.74
10-11 EXHAUST            0.00000   1.013   P11= 101.60   T11= 596.74
PERFORMANCE
UW=                     0.22
SW=                     0.2214
HI=                     0.61
mf=                     0.0125
sfc=                    0.0565
nth=                    0.3639
Rc=                     10.00
```

```
ENGINE CONFIGURATION      FAR    MASS FLOW      P              T
    AMBIENT              0.00000   1.000    P1= 101.30    T1= 288.00
1-2 INTAKE               0.00000   1.000    P2= 100.29    T2= 288.00
2-3 COMPRESSOR           0.00000   1.000    P3=1504.31    T3= 683.69
3-4 PREMASS              0.00000   1.000    P4=1504.31    T4= 683.69
4-5 HEAT EXCHANGER COLD  0.00000   1.000    P5=1489.26    T5= 505.48
5-6 BURNER               0.01332   1.013    P6=1414.80    T6=1000.00
6-7 MIXER                0.00000   1.013    P7=1414.80    T7=1000.00
7-8 COMPRESSOR TURBINE   0.00000   1.013    P8= 423.81    T8= 658.58
8-9 POWER TURBINE        0.00000   1.013    P9= 102.62    T9= 485.68
9-10 HEAT EXCHANGER HOT  0.00000   1.013   P10= 101.60   T10= 663.89
10-11 EXHAUST            0.00000   1.013   P11= 101.60   T11= 663.89
PERFORMANCE
UW=                     0.00
SW=                     0.0000
HI=                     0.00
mf=                     0.0133
sfc=                    0.0000
nth=                    0.0000
Rc=                     15.00
```

```
ENGINE CONFIGURATION      FAR    MASS FLOW      P              T
    AMBIENT              0.00000   1.000    P1= 101.30    T1= 288.00
1-2 INTAKE               0.00000   1.000    P2= 100.29    T2= 288.00
2-3 COMPRESSOR           0.00000   1.000    P3=2005.74    T3= 746.61
3-4 PREMASS              0.00000   1.000    P4=2005.74    T4= 746.61
4-5 HEAT EXCHANGER COLD  0.00000   1.000    P5=1985.68    T5= 480.55
5-6 BURNER               0.01385   1.014    P6=1886.40    T6=1000.00
6-7 MIXER                0.00000   1.014    P7=1886.40    T7=1000.00
7-8 COMPRESSOR TURBINE   0.00000   1.014    P8= 400.98    T8= 604.49
8-9 POWER TURBINE        0.00000   1.014    P9= 102.62    T9= 450.99
9-10 HEAT EXCHANGER HOT  0.00000   1.014   P10= 101.60   T10= 717.05
10-11 EXHAUST            0.00000   1.014   P11= 101.60   T11= 717.05
PERFORMANCE
UW=                     0.00
SW=                     0.0000
HI=                     0.00
mf=                     0.0138
sfc=                    0.0000
nth=                    0.0000
Rc=                     20.00
```

```
ENGINE CONFIGURATION      FAR    MASS FLOW      P              T
    AMBIENT              0.00000   1.000    P1= 101.30    T1= 288.00
1-2 INTAKE               0.00000   1.000    P2= 100.29    T2= 288.00
2-3 COMPRESSOR           0.00000   1.000    P3=1002.87    T3= 603.34
3-4 PREMASS              0.00000   1.000    P4=1002.87    T4= 603.34
4-5 HEAT EXCHANGER COLD  0.00000   1.000    P5= 992.84    T5= 595.26
5-6 BURNER               0.01384   1.014    P6= 943.20    T6=1100.00
6-7 MIXER                0.00000   1.014    P7= 943.20    T7=1100.00
7-8 COMPRESSOR TURBINE   0.00000   1.014    P8= 482.47    T8= 828.05
8-9 POWER TURBINE        0.00000   1.014    P9= 102.62    T9= 594.36
9-10 HEAT EXCHANGER HOT  0.00000   1.014   P10= 101.60   T10= 602.44
10-11 EXHAUST            0.00000   1.014   P11= 101.60   T11= 602.44
PERFORMANCE
UW=                     0.27
SW=                     0.2722
HI=                     0.67
mf=                     0.0138
sfc=                    0.0508
nth=                    0.4047
Rc=                     10.00
```

ENGINE CONFIGURATION	FAR	MASS FLOW	P	T
AMBIENT	0.00000	1.000	P1= 101.30	T1= 288.00
1-2 INTAKE	0.00000	1.000	P2= 100.29	T2= 288.00
2-3 COMPRESSOR	0.00000	1.000	P3=1504.31	T3= 683.69
3-4 PREMASS	0.00000	1.000	P4=1504.31	T4= 683.69
4-5 HEAT EXCHANGER COLD	0.00000	1.000	P5=1489.26	T5= 552.93
5-6 BURNER	0.01472	1.015	P6=1414.80	T6=1100.00
6-7 MIXER	0.00000	1.015	P7=1414.80	T7=1100.00
7-8 COMPRESSOR TURBINE	0.00000	1.015	P8= 510.87	T8= 759.05
8-9 POWER TURBINE	0.00000	1.015	P9= 102.62	T9= 538.40
9-10 HEAT EXCHANGER HOT	0.00000	1.015	P10= 101.60	T10= 669.16
10-11 EXHAUST	0.00000	1.015	P11= 101.60	T11= 669.16

PERFORMANCE

UW=	0.26
SW=	0.2572
HI=	0.72
mf=	0.0147
sfc=	0.0572
nth=	0.3595
Rc=	15.00

ENGINE CONFIGURATION	FAR	MASS FLOW	P	T
AMBIENT	0.00000	1.000	P1= 101.30	T1= 288.00
1-2 INTAKE	0.00000	1.000	P2= 100.29	T2= 288.00
2-3 COMPRESSOR	0.00000	1.000	P3=2005.74	T3= 746.61
3-4 PREMASS	0.00000	1.000	P4=2005.74	T4= 746.61
4-5 HEAT EXCHANGER COLD	0.00000	1.000	P5=1985.68	T5= 525.47
5-6 BURNER	0.01530	1.015	P6=1886.40	T6=1100.00
6-7 MIXER	0.00000	1.015	P7=1886.40	T7=1100.00
7-8 COMPRESSOR TURBINE	0.00000	1.015	P8= 506.95	T8= 705.05
8-9 POWER TURBINE	0.00000	1.015	P9= 102.62	T9= 500.90
9-10 HEAT EXCHANGER HOT	0.00000	1.015	P10= 101.60	T10= 722.04
10-11 EXHAUST	0.00000	1.015	P11= 101.60	T11= 722.04

PERFORMANCE

UW=	0.00
SW=	0.0000
HI=	0.00
mf=	0.0153
sfc=	0.0000
nth=	0.0000
Rc=	20.00

ENGINE CONFIGURATION	FAR	MASS FLOW	P	T
AMBIENT	0.00000	1.000	P1= 101.30	T1= 288.00
1-2 INTAKE	0.00000	1.000	P2= 100.29	T2= 288.00
2-3 COMPRESSOR	0.00000	1.000	P3=1002.87	T3= 603.34
3-4 PREMASS	0.00000	1.000	P4=1002.87	T4= 603.34
4-5 HEAT EXCHANGER COLD	0.00000	1.000	P5= 992.84	T5= 646.60
5-6 BURNER	0.01515	1.015	P6= 943.20	T6=1200.00
6-7 MIXER	0.00000	1.015	P7= 943.20	T7=1200.00
7-8 COMPRESSOR TURBINE	0.00000	1.015	P8= 538.36	T8= 928.40
8-9 POWER TURBINE	0.00000	1.015	P9= 102.62	T9= 651.41
9-10 HEAT EXCHANGER HOT	0.00000	1.015	P10= 101.60	T10= 608.15
10-11 EXHAUST	0.00000	1.015	P11= 101.60	T11= 608.15

PERFORMANCE

UW=	0.32
SW=	0.3230
HI=	0.74
mf=	0.0152
sfc=	0.0469
nth=	0.4386
Rc=	10.00

ENGINE CONFIGURATION	FAR	MASS FLOW	P	T
AMBIENT	0.00000	1.000	P1= 101.30	T1= 288.00
1-2 INTAKE	0.00000	1.000	P2= 100.29	T2= 288.00
2-3 COMPRESSOR	0.00000	1.000	P3=1504.31	T3= 683.69
3-4 PREMASS	0.00000	1.000	P4=1504.31	T4= 683.69
4-5 HEAT EXCHANGER COLD	0.00000	1.000	P5=1489.26	T5= 600.38
5-6 BURNER	0.01612	1.016	P6=1414.80	T6=1200.00
6-7 MIXER	0.00000	1.016	P7=1414.80	T7=1200.00
7-8 COMPRESSOR TURBINE	0.00000	1.016	P8= 593.13	T8= 859.52
8-9 POWER TURBINE	0.00000	1.016	P9= 102.62	T9= 591.12
9-10 HEAT EXCHANGER HOT	0.00000	1.016	P10= 101.60	T10= 674.43
10-11 EXHAUST	0.00000	1.016	P11= 101.60	T11= 674.43

PERFORMANCE

UW=	0.31
SW=	0.3133
HI=	0.78
mf=	0.0161
sfc=	0.0514
nth=	0.4000
Rc=	15.00

```
ENGINE CONFIGURATION      FAR    MASS FLOW      P             T
    AMBIENT                0.00000   1.000   P1= 101.30   T1= 288.00
1-2 INTAKE                 0.00000   1.000   P2= 100.29   T2= 288.00
2-3 COMPRESSOR             0.00000   1.000   P3=2005.74   T3= 746.61
3-4 PREMASS                0.00000   1.000   P4=2005.74   T4= 746.61
4-5 HEAT EXCHANGER COLD    0.00000   1.000   P5=1985.68   T5= 570.39
5-6 BURNER                 0.01675   1.017   P6=1886.40   T6=1200.00
6-7 MIXER                  0.00000   1.017   P7=1886.40   T7=1200.00
7-8 COMPRESSOR TURBINE     0.00000   1.017   P8= 610.23   T8= 805.62
8-9 POWER TURBINE          0.00000   1.017   P9= 102.62   T9= 550.82
9-10 HEAT EXCHANGER HOT    0.00000   1.017  P10= 101.60  T10= 727.03
10-11 EXHAUST              0.00000   1.017  P11= 101.60  T11= 727.03
PERFORMANCE
UW=                       0.30
SW=                       0.2976
HI=                       0.81
mf=                       0.0167
sfc=                      0.0563
nth=                      0.3657
Rc=                       20.00
```

```
ENGINE CONFIGURATION      FAR    MASS FLOW      P             T
    AMBIENT                0.00000   1.000   P1= 101.30   T1= 288.00
1-2 INTAKE                 0.00000   1.000   P2= 100.29   T2= 288.00
2-3 COMPRESSOR             0.00000   1.000   P3=1002.87   T3= 603.34
3-4 PREMASS                0.00000   1.000   P4=1002.87   T4= 603.34
4-5 HEAT EXCHANGER COLD    0.00000   1.000   P5= 992.84   T5= 697.94
5-6 BURNER                 0.01647   1.016   P6= 943.20   T6=1300.00
6-7 MIXER                  0.00000   1.016   P7= 943.20   T7=1300.00
7-8 COMPRESSOR TURBINE     0.00000   1.016   P8= 589.33   T8=1028.75
8-9 POWER TURBINE          0.00000   1.016   P9= 102.62   T9= 708.45
9-10 HEAT EXCHANGER HOT    0.00000   1.016  P10= 101.60  T10= 613.85
10-11 EXHAUST              0.00000   1.016  P11= 101.60  T11= 613.85
PERFORMANCE
UW=                       0.37
SW=                       0.3740
HI=                       0.80
mf=                       0.0165
sfc=                      0.0440
nth=                      0.4673
Rc=                       10.00
```

```
ENGINE CONFIGURATION      FAR    MASS FLOW      P             T
    AMBIENT                0.00000   1.000   P1= 101.30   T1= 288.00
1-2 INTAKE                 0.00000   1.000   P2= 100.29   T2= 288.00
2-3 COMPRESSOR             0.00000   1.000   P3=1504.31   T3= 683.69
3-4 PREMASS                0.00000   1.000   P4=1504.31   T4= 683.69
4-5 HEAT EXCHANGER COLD    0.00000   1.000   P5=1489.26   T5= 647.83
5-6 BURNER                 0.01752   1.018   P6=1414.80   T6=1300.00
6-7 MIXER                  0.00000   1.018   P7=1414.80   T7=1300.00
7-8 COMPRESSOR TURBINE     0.00000   1.018   P8= 670.16   T8= 959.99
8-9 POWER TURBINE          0.00000   1.018   P9= 102.62   T9= 643.84
9-10 HEAT EXCHANGER HOT    0.00000   1.018  P10= 101.60  T10= 679.71
10-11 EXHAUST              0.00000   1.018  P11= 101.60  T11= 679.71
PERFORMANCE
UW=                       0.37
SW=                       0.3696
HI=                       0.85
mf=                       0.0175
sfc=                      0.0474
nth=                      0.4341
Rc=                       15.00
```

```
ENGINE CONFIGURATION      FAR    MASS FLOW      P             T
    AMBIENT                0.00000   1.000   P1= 101.30   T1= 288.00
1-2 INTAKE                 0.00000   1.000   P2= 100.29   T2= 288.00
2-3 COMPRESSOR             0.00000   1.000   P3=2005.74   T3= 746.61
3-4 PREMASS                0.00000   1.000   P4=2005.74   T4= 746.61
4-5 HEAT EXCHANGER COLD    0.00000   1.000   P5=1985.68   T5= 615.32
5-6 BURNER                 0.01819   1.018   P6=1886.40   T6=1300.00
6-7 MIXER                  0.00000   1.018   P7=1886.40   T7=1300.00
7-8 COMPRESSOR TURBINE     0.00000   1.018   P8= 709.31   T8= 906.18
8-9 POWER TURBINE          0.00000   1.018   P9= 102.62   T9= 600.73
9-10 HEAT EXCHANGER HOT    0.00000   1.018  P10= 101.60  T10= 732.02
10-11 EXHAUST              0.00000   1.018  P11= 101.60  T11= 732.02
PERFORMANCE
UW=                       0.36
SW=                       0.3573
HI=                       0.88
mf=                       0.0182
sfc=                      0.0509
nth=                      0.4041
Rc=                       20.00
```

C.6 Off Design simulation program: Input file (1-shaft)

```
********************************************************
INPUT FILE: 1 SHAFT GT OFF DESIGN POINT PERFORMANCE
 ********************************************************
Give Ambient pressure, Pa (90-110Pa)
101.3
Give Ambient temperature, Ta (233-323K)
288
Give Air mass flow, m
1
Give Compressor isentropic eff., nisc (0.85-0.95)
0.85
Give Combustion eff., ncc (0.80-0.995)
0.99
Give Compressor turbine isentropic eff., nist (0.80-0.97)
0.90
Give Intake pressure loss, DPinloss (0%-4%)
1
Give Combustor pressure loss, Dpccloss (3%-5%)
5
Give Exhaust pressure loss, DPexhloss (0%-4%)
0
Give Turbine Entry Temperature, TET (K)
1300
Give minimum Compressor pressure ratio, Rc
15
Give Inlet Mach number, Min
0
Give Exit Mach number, Mex
1
Give Compressor degradation, Pcde
0
Give Turbine degradation, Ptde
0
Give Exhaust degradation, Pexhde
0
Give FUEL (SynGas               (6.1)   :1,
           Biogas               (1)     :2,
           Biodiesel            (8)     :3,
           Residual Oil         (40.3)  :4,
           Medium Heating Oil   (41)    :5,
           Light Heating Oil    (42.5)  :6,
           Kerozine             (43.2)  :7,
           Russian Natural Gas  (48.6)  :8,
           Algerian Natural Gas (48.9)  :9)
8
Variation of Tambient:1, Variation of Pressure:2 Variation of Altitude:3
```

C.7 Off Design simulation program: Output file (1-shaft)

```
                    1ShaftGTod: VARIATION OF Tamb
-----------------------------------------------------------------------
ENGINE CONFIGURATION      FAR   MASS FLOW      P              T
-----------------------------------------------------------------------
    AMBIENT             0.00000     1.055   P1od= 101.30   T1od= 273.00
1-2 INTAKE              0.00000     1.055   P2od= 100.29   T2od= 273.00
2-3 COMPRESSOR          0.00000     1.055   P3od=1524.70   T3od= 651.77
3-4 PREMASS             0.00000     1.055   P4od=1524.70   T4od= 651.77
4-5 COMBUSTION CHAMBER  0.01586     1.072   P5od=1448.47   T5od=1200.00
5-6 MIXER               0.00000     1.072   P6od=1448.47   T6od=1200.00
6-7 COMPR & POWER TURB  0.00000     1.072   P7od= 101.60   T7od= 677.47
7-8 EXHAUST             0.00000     1.072   P8od= 101.60   T8od= 677.47
PERFORMANCE
UWod=                    0.2323
SWod=                   0.22024
HIod=                     0.765
mfod=                   0.01673
sfcod=                  0.06773
nthod=                   0.3038
niscod=                 0.84775
nistod=                 0.89761
Taod=                   273.0
Pa=                     101.30
Aod=                    0.00
Rcod=                   15.20
Qoutod=                 0.4985
-----------------------------------------------------------------------
ENGINE CONFIGURATION      FAR   MASS FLOW      P              T
-----------------------------------------------------------------------
    AMBIENT             0.00000     1.018   P1od= 101.30   T1od= 283.00
1-2 INTAKE              0.00000     1.018   P2od= 100.29   T2od= 283.00
2-3 COMPRESSOR          0.00000     1.018   P3od=1470.82   T3od= 669.60
3-4 PREMASS             0.00000     1.018   P4od=1470.82   T4od= 669.60
4-5 COMBUSTION CHAMBER  0.01547     1.033   P5od=1397.28   T5od=1200.00
5-6 MIXER               0.00000     1.033   P6od=1397.28   T6od=1200.00
6-7 COMPR & POWER TURB  0.00000     1.033   P7od= 101.60   T7od= 684.34
7-8 EXHAUST             0.00000     1.033   P8od= 101.60   T8od= 684.34
PERFORMANCE
UWod=                    0.2081
SWod=                   0.20447
HIod=                     0.720
mfod=                   0.01574
sfcod=                  0.07115
nthod=                   0.2892
niscod=                 0.84470
nistod=                 0.89439
Taod=                   283.0
Pa=                     101.30
Aod=                    0.00
Rcod=                   14.67
Qoutod=                 0.4770
-----------------------------------------------------------------------
ENGINE CONFIGURATION      FAR   MASS FLOW      P              T
-----------------------------------------------------------------------
    AMBIENT             0.00000     1.055   P1od= 101.30   T1od= 273.00
1-2 INTAKE              0.00000     1.055   P2od= 100.29   T2od= 273.00
2-3 COMPRESSOR          0.00000     1.055   P3od=1586.96   T3od= 656.74
3-4 PREMASS             0.00000     1.055   P4od=1586.96   T4od= 656.74
4-5 COMBUSTION CHAMBER  0.01827     1.074   P5od=1507.61   T5od=1300.00
5-6 MIXER               0.00000     1.074   P6od=1507.61   T6od=1300.00
6-7 COMPR & POWER TURB  0.00000     1.074   P7od= 101.60   T7od= 723.36
7-8 EXHAUST             0.00000     1.074   P8od= 101.60   T8od= 723.36
PERFORMANCE
UWod=                    0.2927
SWod=                   0.27748
HIod=                     0.881
mfod=                   0.01927
sfcod=                  0.06192
nthod=                   0.3323
niscod=                 0.85456
nistod=                 0.90483
Taod=                   273.0
Pa=                     101.30
Aod=                    0.00
Rcod=                   15.82
Qoutod=                 0.5563
-----------------------------------------------------------------------
ENGINE CONFIGURATION      FAR   MASS FLOW      P              T
-----------------------------------------------------------------------
    AMBIENT             0.00000     1.018   P1od= 101.30   T1od= 283.00
1-2 INTAKE              0.00000     1.018   P2od= 100.29   T2od= 283.00
2-3 COMPRESSOR          0.00000     1.018   P3od=1530.88   T3od= 674.75
3-4 PREMASS             0.00000     1.018   P4od=1530.88   T4od= 674.75
4-5 COMBUSTION CHAMBER  0.01787     1.036   P5od=1454.34   T5od=1300.00
5-6 MIXER               0.00000     1.036   P6od=1454.34   T6od=1300.00
6-7 COMPR & POWER TURB  0.00000     1.036   P7od= 101.60   T7od= 730.82
7-8 EXHAUST             0.00000     1.036   P8od= 101.60   T8od= 730.82
```

PERFORMANCE	
UWod=	0.2655
SWod=	0.26084
HIod=	0.831
mfod=	0.01819
sfcod=	0.06444
nthod=	0.3193
niscod=	0.85149
nistod=	0.90158
Taod=	283.0
Pa=	101.30
Aod=	0.00
Rcod=	15.27
Qoutod=	0.5335

1ShaftGTod: VARIATION OF Pamb

ENGINE CONFIGURATION	FAR	MASS FLOW	P	T
AMBIENT	0.00000	0.997	P1od= 101.00	T1od= 288.00
1-2 INTAKE	0.00000	0.997	P2od= 99.99	T2od= 288.00
2-3 COMPRESSOR	0.00000	0.997	P3od=1441.01	T3od= 678.45
3-4 PREMASS	0.00000	0.997	P4od=1441.01	T4od= 678.45
4-5 COMBUSTION CHAMBER	0.01527	1.012	P5od=1368.96	T5od=1200.00
5-6 MIXER	0.00000	1.012	P6od=1368.96	T6od=1200.00
6-7 COMPR & POWER TURB	0.00000	1.012	P7od= 101.30	T7od= 687.68
7-8 EXHAUST	0.00000	1.012	P8od= 101.30	T8od= 687.68
PERFORMANCE				
UWod=	0.1962			
SWod=	0.19676			
HIod=	0.696			
mfod=	0.01523			
sfcod=	0.07301			
nthod=	0.2818			
niscod=	0.84322			
nistod=	0.89282			
Ta=	288.0			
Paod=	101.00			
Aod=	0.00			
Rcod=	14.41			
Qoutod=	0.4653			

ENGINE CONFIGURATION	FAR	MASS FLOW	P	T
AMBIENT	0.00000	1.017	P1od= 103.00	T1od= 288.00
1-2 INTAKE	0.00000	1.017	P2od= 101.97	T2od= 288.00
2-3 COMPRESSOR	0.00000	1.017	P3od=1469.54	T3od= 678.45
3-4 PREMASS	0.00000	1.017	P4od=1469.54	T4od= 678.45
4-5 COMBUSTION CHAMBER	0.01527	1.032	P5od=1396.07	T5od=1200.00
5-6 MIXER	0.00000	1.032	P6od=1396.07	T6od=1200.00
6-7 COMPR & POWER TURB	0.00000	1.032	P7od= 103.31	T7od= 687.68
7-8 EXHAUST	0.00000	1.032	P8od= 103.31	T8od= 687.68
PERFORMANCE				
UWod=	0.2001			
SWod=	0.19676			
HIod=	0.710			
mfod=	0.01553			
sfcod=	0.07301			
nthod=	0.2818			
niscod=	0.84322			
nistod=	0.89282			
Ta=	288.0			
Paod=	103.00			
Aod=	0.00			
Rcod=	14.41			
Qoutod=	0.4745			

ENGINE CONFIGURATION	FAR	MASS FLOW	P	T
AMBIENT	0.00000	0.997	P1od= 101.00	T1od= 288.00
1-2 INTAKE	0.00000	0.997	P2od= 99.99	T2od= 288.00
2-3 COMPRESSOR	0.00000	0.997	P3od=1499.85	T3od= 683.69
3-4 PREMASS	0.00000	0.997	P4od=1499.85	T4od= 683.69
4-5 COMBUSTION CHAMBER	0.01767	1.015	P5od=1424.86	T5od=1300.00
5-6 MIXER	0.00000	1.015	P6od=1424.86	T6od=1300.00
6-7 COMPR & POWER TURB	0.00000	1.015	P7od= 101.30	T7od= 734.45
7-8 EXHAUST	0.00000	1.015	P8od= 101.30	T8od= 734.45
PERFORMANCE				
UWod=	0.2520			
SWod=	0.25271			
HIod=	0.805			
mfod=	0.01762			
sfcod=	0.06578			
nthod=	0.3128			
niscod=	0.85000			
nistod=	0.90000			

```
Ta=                    288.0
Paod=                  101.00
Aod=                   0.00
Rcod=                  15.00
Qoutod=                0.5210
```

```
ENGINE CONFIGURATION      FAR    MASS FLOW       P             T
    AMBIENT            0.00000      1.017    P1od= 103.00   T1od= 288.00
1-2 INTAKE             0.00000      1.017    P2od= 101.97   T2od= 288.00
2-3 COMPRESSOR         0.00000      1.017    P3od=1529.55   T3od= 683.69
3-4 PREMASS            0.00000      1.017    P4od=1529.55   T4od= 683.69
4-5 COMBUSTION CHAMBER 0.01767      1.035    P5od=1453.07   T5od=1300.00
5-6 MIXER              0.00000      1.035    P6od=1453.07   T6od=1300.00
6-7 COMPR & POWER TURB 0.00000      1.035    P7od= 103.31   T7od= 734.45
7-8 EXHAUST            0.00000      1.035    P8od= 103.31   T8od= 734.45
PERFORMANCE
UWod=                   0.2569
SWod=                  0.25271
HIod=                    0.821
mfod=                  0.01797
sfcod=                 0.06578
nthod=                 0.3128
niscod=                0.85000
nistod=                0.90000
Ta=                    288.0
Paod=                  103.00
Aod=                   0.00
Rcod=                  15.00
Qoutod=                0.5313
```

1ShaftGTod: VARIATION OF Altitude

```
ENGINE CONFIGURATION      FAR    MASS FLOW       P             T
    AMBIENT            0.00000      1.000    P1od= 101.32   T1od= 288.15
1-2 INTAKE             0.00000      1.000    P2od= 100.31   T2od= 288.15
2-3 COMPRESSOR         0.00000      1.000    P3od=1444.89   T3od= 678.72
3-4 PREMASS            0.00000      1.000    P4od=1444.89   T4od= 678.72
4-5 COMBUSTION CHAMBER 0.01527      1.015    P5od=1372.65   T5od=1200.00
5-6 MIXER              0.00000      1.015    P6od=1372.65   T6od=1200.00
6-7 COMPR & POWER TURB 0.00000      1.015    P7od= 101.63   T7od= 687.78
7-8 EXHAUST            0.00000      1.015    P8od= 101.63   T8od= 687.78
PERFORMANCE
UWod=                   0.1965
SWod=                  0.19653
HIod=                    0.698
mfod=                  0.01526
sfcod=                 0.0731
nthod=                 0.282
niscod=                0.84318
nistod=                0.89278
Taod=                  288.1
Paod=                 101.32
Aod=                      0.0
Rcod=                  14.40
Qoutod=                0.4665
```

```
ENGINE CONFIGURATION      FAR    MASS FLOW       P             T
    AMBIENT            0.00000      0.976    P1od=  98.36   T1od= 286.52
1-2 INTAKE             0.00000      0.976    P2od=  97.37   T2od= 286.52
2-3 COMPRESSOR         0.00000      0.976    P3od=1410.53   T3od= 675.85
3-4 PREMASS            0.00000      0.976    P4od=1410.53   T4od= 675.85
4-5 COMBUSTION CHAMBER 0.01533      0.991    P5od=1340.01   T5od=1200.00
5-6 MIXER              0.00000      0.991    P6od=1340.01   T6od=1200.00
6-7 COMPR & POWER TURB 0.00000      0.991    P7od=  98.65   T7od= 686.70
7-8 EXHAUST            0.00000      0.991    P8od=  98.65   T8od= 686.70
PERFORMANCE
UWod=                   0.1942
SWod=                  0.19903
HIod=                    0.684
mfod=                  0.01496
sfcod=                 0.0724
nthod=                 0.284
niscod=                0.84366
nistod=                0.89328
Taod=                  286.5
Paod=                  98.36
Aod=                    250.0
Rcod=                  14.49
Qoutod=                0.4560
```

```
ENGINE CONFIGURATION      FAR    MASS FLOW       P             T
    AMBIENT            0.00000      1.000    P1od= 101.32   T1od= 288.15
1-2 INTAKE             0.00000      1.000    P2od= 100.31   T2od= 288.15
2-3 COMPRESSOR         0.00000      1.000    P3od=1503.89   T3od= 683.96
```

3-4 PREMASS	0.00000	1.000	P4od=1503.89	T4od= 683.96
4-5 COMBUSTION CHAMBER	0.01767	1.017	P5od=1428.70	T5od=1300.00
5-6 MIXER	0.00000	1.017	P6od=1428.70	T6od=1300.00
6-7 COMPR & POWER TURB	0.00000	1.017	P7od= 101.63	T7od= 734.56
7-8 EXHAUST	0.00000	1.017	P8od= 101.63	T8od= 734.56

PERFORMANCE

UWod=	0.2524
SWod=	0.25247
HIod=	0.807
mfod=	0.01766
sfcod=	0.0658
nthod=	0.313
niscod=	0.84996
nistod=	0.89995
Taod=	288.1
Paod=	101.32
Aod=	0.0
Rcod=	14.99
Qoutod=	0.5223

ENGINE CONFIGURATION	FAR	MASS FLOW	P	T
AMBIENT	0.00000	0.976	P1od= 98.36	T1od= 286.52
1-2 INTAKE	0.00000	0.976	P2od= 97.37	T2od= 286.52
2-3 COMPRESSOR	0.00000	0.976	P3od=1468.13	T3od= 681.06
3-4 PREMASS	0.00000	0.976	P4od=1468.13	T4od= 681.06
4-5 COMBUSTION CHAMBER	0.01773	0.993	P5od=1394.72	T5od=1300.00
5-6 MIXER	0.00000	0.993	P6od=1394.72	T6od=1300.00
6-7 COMPR & POWER TURB	0.00000	0.993	P7od= 98.65	T7od= 733.39
7-8 EXHAUST	0.00000	0.993	P8od= 98.65	T8od= 733.39

PERFORMANCE

UWod=	0.2490
SWod=	0.25510
HIod=	0.791
mfod=	0.01731
sfcod=	0.0654
nthod=	0.315
niscod=	0.85044
nistod=	0.90046
Taod=	286.5
Paod=	98.36
Aod=	250.0
Rcod=	15.08
Qoutod=	0.5104

C.8 Off Design simulation program: Input file (2-shaft)

```
*********************************************************
   INPUT FILE: 2 SHAFT GT OFF DESIGN POINT PERFORMANCE
 *********************************************************
Give Ambient temperature, Pa (90-110Pa)
101.3
Give Ambient temperature, Ta (233-323K)
288
Give Air mass flow, m
1
Give Compressor isentropic eff., nisc (0.85-0.95)
0.85
Give Combustion eff., ncc (0.80-0.995)
0.99
Give Compressor turbine isentropic eff., npolct (0.80-0.97)
0.86
Give Heat exchanger hot eff., nisct (0.84-0.96)
0.89
Give Heat exchanger hot eff., nispt (0.84-0.96)
0.88
Give Intake pressure loss, DPinloss (0%-4%)
1
Give Heat exchanger cold pressure loss, DPcheloss (1%-4%)
1
Give Combustor pressure loss, DPccloss (3%-5%)
5
Give Heat exchanger hot pressure loss, DPhheloss (1%-4%)
1
Give Exhaust pressure loss, DPexhloss (0%-4%)
0
Give Turbine Entry Temperature, TET (K)
1300
Give minimum Compressor pressure ratio, Rc
15
Give Inlet Mach number, Minv
0
Give Exit Mach number, Mexv
1
Give Compressor degradation, Pcdev
0
Give Turbine degradation, Ptdev
0
Give Exhaust degradation, Pexhdev
0
Give FUEL (SynGas               (6.1)    :1,
           Biogas               (1)      :2,
           Biodiesel            (8)      :3,
           Residual Oil         (40.3)   :4,
           Medium Heating Oil   (41)     :5,
           Light Heating Oil    (42.5)   :6,
           Kerozine             (43.2)   :7,
           Russian Natural Gas  (48.6)   :8,
           Algerian Natural Gas (48.9)   :9)
8
Variation of Tambient:1, Variation of Pressure:2 Variation of Altitude:3
```

C.9 Off Design simulation program: Output file (2-shaft)

```
                 2ShaftGTod: VARIATION OF Tamb
-----------------------------------------------------------------------
ENGINE CONFIGURATION      FAR   MASS FLOW       P              T
-----------------------------------------------------------------------
    AMBIENT             0.00000   0.929   P1od= 101.30   T1od= 273.00
1-2 INTAKE              0.00000   0.929   P2od= 100.29   T2od= 273.00
2-3 COMPRESSOR          0.00000   0.929   P3od=1343.00   T3od= 626.82
3-4 PREMASS             0.00000   0.929   P4od=1343.00   T4od= 626.82
4-5 COMBUSTION CHAMBER  0.01668   0.945   P5od=1275.85   T5od=1200.00
5-6 MIXER               0.00000   0.945   P6od=1275.85   T6od=1200.00
6-7 COMPR & POWER TURB  0.00000   0.945   P7od= 317.59   T7od= 890.79
7-8 EXHAUST             0.00000   0.945   P8od= 101.60   T8od= 696.98
8-9 EXHAUST             0.00000   0.945   P9od= 101.60   T9od= 696.98
PERFORMANCE
UWod=                   0.21059
SWod=                   0.22660
HIod=                   0.70867
mfod=                   0.01550
sfcod=                  0.06924
nthod=                   0.2972
niscod=                 0.84775
nisctod=                0.87801
nisptod=                0.87813
Taod=                   273.0
Pa=                       0.00
npolctod=               0.85772
Rcod=                   13.39
Qoutod=                 0.4607

-----------------------------------------------------------------------
ENGINE CONFIGURATION      FAR   MASS FLOW       P              T
-----------------------------------------------------------------------
    AMBIENT             0.00000   0.860   P1od= 101.30   T1od= 283.00
1-2 INTAKE              0.00000   0.860   P2od= 100.29   T2od= 283.00
2-3 COMPRESSOR          0.00000   0.860   P3od=1242.18   T3od= 635.60
3-4 PREMASS             0.00000   0.860   P4od=1242.18   T4od= 635.60
4-5 COMBUSTION CHAMBER  0.01648   0.874   P5od=1180.07   T5od=1200.00
5-6 MIXER               0.00000   0.874   P6od=1180.07   T6od=1200.00
6-7 COMPR & POWER TURB  0.00000   0.874   P7od= 293.93   T7od= 891.86
7-8 EXHAUST             0.00000   0.874   P8od= 101.60   T8od= 709.96
8-9 EXHAUST             0.00000   0.874   P9od= 101.60   T9od= 709.96
PERFORMANCE
UWod=                   0.00000
SWod=                   0.00000
HIod=                   0.00000
mfod=                   0.00000
sfcod=                  0.00000
nthod=                   0.0000
niscod=                 0.00000
nisctod=                0.00000
nisptod=                0.00000
Taod=                   283.0
Pa=                       0.00
npolctod=               0.00000
Rcod=                   12.39
Qoutod=                 0.0000

-----------------------------------------------------------------------
ENGINE CONFIGURATION      FAR   MASS FLOW       P              T
-----------------------------------------------------------------------
    AMBIENT             0.00000   1.066   P1od= 101.30   T1od= 273.00
1-2 INTAKE              0.00000   1.066   P2od= 100.29   T2od= 273.00
2-3 COMPRESSOR          0.00000   1.066   P3od=1606.94   T3od= 659.26
3-4 PREMASS             0.00000   1.066   P4od=1606.94   T4od= 659.26
4-5 COMBUSTION CHAMBER  0.01855   1.086   P5od=1526.59   T5od=1300.00
5-6 MIXER               0.00000   1.086   P6od=1526.59   T6od=1300.00
6-7 COMPR & POWER TURB  0.00000   1.086   P7od= 379.50   T7od= 962.44
7-8 EXHAUST             0.00000   1.086   P8od= 101.60   T8od= 723.54
8-9 EXHAUST             0.00000   1.086   P9od= 101.60   T9od= 723.54
PERFORMANCE
UWod=                   0.29841
SWod=                   0.27983
HIod=                   0.90417
mfod=                   0.01978
sfcod=                  0.06234
nthod=                   0.3300
niscod=                 0.85456
nisctod=                0.88408
nisptod=                0.88495
Taod=                   273.0
Pa=                       0.00
npolctod=               0.86461
Rcod=                   16.02
Qoutod=                 0.5628

-----------------------------------------------------------------------
ENGINE CONFIGURATION      FAR   MASS FLOW       P              T
-----------------------------------------------------------------------
    AMBIENT             0.00000   0.983   P1od= 101.30   T1od= 283.00
1-2 INTAKE              0.00000   0.983   P2od= 100.29   T2od= 283.00
```

```
2-3 COMPRESSOR          0.00000    0.983    P3od=1480.98    T3od= 667.93
3-4 PREMASS             0.00000    0.983    P4od=1480.98    T4od= 667.93
4-5 COMBUSTION CHAMBER  0.01835    1.001    P5od=1406.93    T5od=1300.00
5-6 MIXER               0.00000    1.001    P6od=1406.93    T6od=1300.00
6-7 COMPR & POWER TURB  0.00000    1.001    P7od= 349.96    T7od= 963.60
7-8 EXHAUST             0.00000    1.001    P8od= 101.60    T8od= 737.75
8-9 EXHAUST             0.00000    1.001    P9od= 101.60    T9od= 737.75
PERFORMANCE
UWod=                   0.26000
SWod=                   0.26451
HIod=                   0.82457
mfod=                   0.01804
sfcod=                  0.06525
nthod=                   0.3153
niscod=                 0.85149
nisctod=                0.88135
nisptod=                0.88188
Taod=                   283.0
Pa=                       0.00
npolctod=               0.86151
Rcod=                   14.77
Qoutod=                 0.5235
```

```
                 2ShaftGTod: VARIATION OF Pamb
--------------------------------------------------------------------------
ENGINE CONFIGURATION      FAR   MASS FLOW       P               T
--------------------------------------------------------------------------
    AMBIENT             0.00000    0.883    P1od= 101.00    T1od= 288.00
1-2 INTAKE              0.00000    0.883    P2od=  99.99    T2od= 288.00
2-3 COMPRESSOR          0.00000    0.883    P3od=1302.49    T3od= 656.11
3-4 PREMASS             0.00000    0.883    P4od=1302.49    T4od= 656.11
4-5 COMBUSTION CHAMBER  0.01644    0.898    P5od=1237.37    T5od=1250.00
5-6 MIXER               0.00000    0.898    P6od=1237.37    T6od=1250.00
6-7 COMPR & POWER TURB  0.00000    0.898    P7od= 308.08    T7od= 928.30
7-8 EXHAUST             0.00000    0.898    P8od= 101.30    T8od= 730.79
8-9 EXHAUST             0.00000    0.898    P9od= 101.30    T9od= 730.79
PERFORMANCE
UWod=                   0.20393
SWod=                   0.23087
HIod=                   0.69860
mfod=                   0.01452
sfcod=                  0.07049
nthod=                   0.2919
niscod=                 0.84667
nisctod=                0.87706
nisptod=                0.87706
Ta=                       0.0
Paod=                   101.00
npolctod=               0.85663
Rcod=                   13.03
Qoutod=                 0.4572
```

```
--------------------------------------------------------------------------
ENGINE CONFIGURATION      FAR   MASS FLOW       P               T
--------------------------------------------------------------------------
    AMBIENT             0.00000    0.901    P1od= 103.00    T1od= 288.00
1-2 INTAKE              0.00000    0.901    P2od= 101.97    T2od= 288.00
2-3 COMPRESSOR          0.00000    0.901    P3od=1328.29    T3od= 656.11
3-4 PREMASS             0.00000    0.901    P4od=1328.29    T4od= 656.11
4-5 COMBUSTION CHAMBER  0.01644    0.916    P5od=1261.87    T5od=1250.00
5-6 MIXER               0.00000    0.916    P6od=1261.87    T6od=1250.00
6-7 COMPR & POWER TURB  0.00000    0.916    P7od= 314.18    T7od= 928.30
7-8 EXHAUST             0.00000    0.916    P8od= 103.31    T8od= 730.79
8-9 EXHAUST             0.00000    0.916    P9od= 103.31    T9od= 730.79
PERFORMANCE
UWod=                   0.20797
SWod=                   0.23087
HIod=                   0.71243
mfod=                   0.01481
sfcod=                  0.07049
nthod=                   0.2919
niscod=                 0.84667
nisctod=                0.87706
nisptod=                0.87706
Ta=                       0.0
Paod=                   103.00
npolctod=               0.85663
Rcod=                   13.03
Qoutod=                 0.4662
```

```
--------------------------------------------------------------------------
ENGINE CONFIGURATION      FAR   MASS FLOW       P               T
--------------------------------------------------------------------------
    AMBIENT             0.00000    0.943    P1od= 101.00    T1od= 288.00
1-2 INTAKE              0.00000    0.943    P2od=  99.99    T2od= 288.00
2-3 COMPRESSOR          0.00000    0.943    P3od=1419.89    T3od= 672.28
3-4 PREMASS             0.00000    0.943    P4od=1419.89    T4od= 672.28
4-5 COMBUSTION CHAMBER  0.01732    0.960    P5od=1348.90    T5od=1300.00
5-6 MIXER               0.00000    0.960    P6od=1348.90    T6od=1300.00
6-7 COMPR & POWER TURB  0.00000    0.960    P7od= 335.63    T7od= 964.17
7-8 EXHAUST             0.00000    0.960    P8od= 101.30    T8od= 744.64
```

```
8-9 EXHAUST               0.00000   0.960   P9od= 101.30   T9od= 744.64
PERFORMANCE
UWod=                     0.24229
SWod=                     0.25683
HIod=                     0.78637
mfod=                     0.01634
sfcod=                    0.06678
nthod=                     0.3081
niscod=                   0.85000
nisctod=                  0.88002
nisptod=                  0.88039
Ta=                         0.0
Paod=                     101.00
npolctod=                 0.86000
Rcod=                     14.20
Qoutod=                   0.5040
```

```
ENGINE CONFIGURATION       FAR   MASS FLOW      P              T
    AMBIENT               0.00000   0.962   P1od= 103.00   T1od= 288.00
1-2 INTAKE                0.00000   0.962   P2od= 101.97   T2od= 288.00
2-3 COMPRESSOR            0.00000   0.962   P3od=1448.01   T3od= 672.28
3-4 PREMASS               0.00000   0.962   P4od=1448.01   T4od= 672.28
4-5 COMBUSTION CHAMBER    0.01732   0.979   P5od=1375.61   T5od=1300.00
5-6 MIXER                 0.00000   0.979   P6od=1375.61   T6od=1300.00
6-7 COMPR & POWER TURB    0.00000   0.979   P7od= 342.27   T7od= 964.17
7-8 EXHAUST               0.00000   0.979   P8od= 103.31   T8od= 744.64
8-9 EXHAUST               0.00000   0.979   P9od= 103.31   T9od= 744.64
PERFORMANCE
UWod=                     0.24709
SWod=                     0.25683
HIod=                     0.80194
mfod=                     0.01667
sfcod=                    0.06678
nthod=                     0.3081
niscod=                   0.85000
nisctod=                  0.88002
nisptod=                  0.88039
Ta=                         0.0
Paod=                     103.00
npolctod=                 0.86000
Rcod=                     14.20
Qoutod=                   0.5140
```

```
               2ShaftGTod: VARIATION OF Altitude
ENGINE CONFIGURATION       FAR   MASS FLOW      P              T
    AMBIENT               0.00000   0.885   P1od= 101.32   T1od= 288.15
1-2 INTAKE                0.00000   0.885   P2od= 100.31   T2od= 288.15
2-3 COMPRESSOR            0.00000   0.885   P3od=1305.21   T3od= 656.24
3-4 PREMASS               0.00000   0.885   P4od=1305.21   T4od= 656.24
4-5 COMBUSTION CHAMBER    0.01644   0.900   P5od=1239.95   T5od=1250.00
5-6 MIXER                 0.00000   0.900   P6od=1239.95   T6od=1250.00
6-7 COMPR & POWER TURB    0.00000   0.900   P7od= 308.72   T7od= 928.32
7-8 EXHAUST               0.00000   0.900   P8od= 101.63   T8od= 730.99
8-9 EXHAUST               0.00000   0.900   P9od= 101.63   T9od= 730.99
PERFORMANCE
UWod=                     0.20417
SWod=                     0.23066
HIod=                     0.69994
mfod=                     0.01455
sfcod=                     0.0705
nthod=                      0.292
niscod=                   0.84663
nisctod=                  0.87702
nisptod=                  0.87701
npolctod=                 0.85659
Paod=                     101.32
Aod=                          0.0
Rcod=                     13.01
Qoutod=                   0.4582
```

```
ENGINE CONFIGURATION       FAR   MASS FLOW      P              T
    AMBIENT               0.00000   0.870   P1od=  98.36   T1od= 286.52
1-2 INTAKE                0.00000   0.870   P2od=  97.37   T2od= 286.52
2-3 COMPRESSOR            0.00000   0.870   P3od=1282.69   T3od= 654.82
3-4 PREMASS               0.00000   0.870   P4od=1282.69   T4od= 654.82
4-5 COMBUSTION CHAMBER    0.01647   0.884   P5od=1218.56   T5od=1250.00
5-6 MIXER                 0.00000   0.884   P6od=1218.56   T6od=1250.00
6-7 COMPR & POWER TURB    0.00000   0.884   P7od= 303.37   T7od= 928.14
7-8 EXHAUST               0.00000   0.884   P8od=  98.65   T8od= 728.86
8-9 EXHAUST               0.00000   0.884   P9od=  98.65   T9od= 728.86
PERFORMANCE
UWod=                     0.20263
SWod=                     0.23295
HIod=                     0.68913
```

```
mfod=                   0.01432
sfcod=                   0.0700
nthod=                    0.294
niscod=                 0.84711
nisctod=                0.87745
nisptod=                0.87749
npolctod=               0.85707
Paod=                    98.36
Aod=                     250.0
Rcod=                   13.17
Qoutod=                 0.4498
```

ENGINE CONFIGURATION	FAR	MASS FLOW	P	T
AMBIENT	0.00000	0.945	P1od= 101.32	T1od= 288.15
1-2 INTAKE	0.00000	0.945	P2od= 100.31	T2od= 288.15
2-3 COMPRESSOR	0.00000	0.945	P3od=1422.81	T3od= 672.41
3-4 PREMASS	0.00000	0.945	P4od=1422.81	T4od= 672.41
4-5 COMBUSTION CHAMBER	0.01732	0.962	P5od=1351.67	T5od=1300.00
5-6 MIXER	0.00000	0.962	P6od=1351.67	T6od=1300.00
6-7 COMPR & POWER TURB	0.00000	0.962	P7od= 336.32	T7od= 964.19
7-8 EXHAUST	0.00000	0.962	P8od= 101.63	T8od= 744.85
8-9 EXHAUST	0.00000	0.962	P9od= 101.63	T9od= 744.85

```
PERFORMANCE
UWod=                   0.24259
SWod=                   0.25661
HIod=                   0.78786
mfod=                   0.01637
sfcod=                   0.0668
nthod=                    0.308
niscod=                 0.84996
nisctod=                0.87998
nisptod=                0.88035
npolctod=               0.85996
Paod=                   101.32
Aod=                       0.0
Rcod=                   14.18
Qoutod=                 0.5051
```

ENGINE CONFIGURATION	FAR	MASS FLOW	P	T
AMBIENT	0.00000	0.929	P1od= 98.36	T1od= 286.52
1-2 INTAKE	0.00000	0.929	P2od= 97.37	T2od= 286.52
2-3 COMPRESSOR	0.00000	0.929	P3od=1398.65	T3od= 671.00
3-4 PREMASS	0.00000	0.929	P4od=1398.65	T4od= 671.00
4-5 COMBUSTION CHAMBER	0.01735	0.945	P5od=1328.72	T5od=1300.00
5-6 MIXER	0.00000	0.945	P6od=1328.72	T6od=1300.00
6-7 COMPR & POWER TURB	0.00000	0.945	P7od= 330.58	T7od= 964.01
7-8 EXHAUST	0.00000	0.945	P8od= 98.65	T8od= 742.62
8-9 EXHAUST	0.00000	0.945	P9od= 98.65	T9od= 742.62

```
PERFORMANCE
UWod=                   0.24069
SWod=                   0.25901
HIod=                   0.77582
mfod=                   0.01612
sfcod=                   0.0663
nthod=                    0.310
niscod=                 0.85044
nisctod=                0.88041
nisptod=                0.88083
npolctod=               0.86044
Paod=                    98.36
Aod=                     250.0
Rcod=                   14.36
Qoutod=                 0.4959
```

C.10 Off Design simulation program: Input file (1-shaft HE)

```
*********************************************************
   INPUT FILE: 1 SHAFT GT HE OFF DESIGN POINT PERFORMANCE
*********************************************************
Give Ambient pressure, Pav (90-110Pa)
101.3
Give Ambient temperature, Tav (233-323K)
288
Give Air mass flow, mv
1
Give Compressor isentropic eff., niscv (0.85-0.95)
0.85
Give Heat exchanger cold eff., nchev (0.84-0.96)
0.95
Give Combustion eff., nccv (0.80-0.995)
0.99
Give Compressor turbine isentropic eff., nistv (0.80-0.97)
0.85
Give Heat exchanger hot eff., nhhev (0.84-0.96)
0.95
Give Intake pressure loss, DPinlossv (0%-4%)
1
Give Heat exchanger cold pressure loss, Dpchelossv (1%-4%)
1
Give Combustor pressure loss, DPcclossv (3%-5%)
5
Give Heat exchanger hot pressure loss, DPhhelossv (1%-4%)
1
Give Exhaust pressure loss, DPexhlossv (0%-4%)
0
Give Turbine Entry Temperature, TET (K)
1100
Give Compressor pressure ratio, Rc
5
Give Inlet Mach number, Minv
0
Give Exit Mach number, Mexv
1
Give Compressor degradation, Pcdev
0
Give Turbine degradation, Ptdev
0
Give Exhaust degradation, Pexhdev
0
Give FUEL (SynGas               (6.1)   :1,
           Biogas               (1)     :2,
           Biodiesel            (8)     :3,
           Residual Oil         (40.3)  :4,
           Medium Heating Oil   (41)    :5,
           Light Heating Oil    (42.5)  :6,
           Kerozine             (43.2)  :7,
           Russian Natural Gas  (48.6)  :8,
           Algerian Natural Gas (48.9)  :9)
8
Variation of Tambient:1, Variation of Pressure:2, Variation of Altitude:3
```

C.11 Off Design simulation program: Output file (1-shaft HE)

```
                       1ShaftGTod HE: VARIATION OF Tamb
-----------------------------------------------------------------------
ENGINE CONFIGURATION      FAR   MASS FLOW       P             T
-----------------------------------------------------------------------
    AMBIENT             0.00000     1.055   P1od= 101.30   T1od= 273.00
1-2 INTAKE              0.00000     1.055   P2od= 100.29   T2od= 273.00
2-3 COMPRESSOR          0.00000     1.055   P3od= 499.33   T3od= 460.68
3-4 PREMASS             0.00000     1.055   P4od= 499.33   T4od= 460.68
4-5 HEAT EXCHANGER COL  0.00000     1.055   P5od= 499.33   T5od= 460.68
5-6 BURNER              0.01503     1.071   P6od= 474.36   T6od=1000.00
6-7 MIXER               0.00000     1.071   P7od= 474.36   T7od=1000.00
7-8 COMPR & POWER TURB  0.00000     1.071   P8od= 101.60   T8od= 729.56
8-9 HEAT EXCHANGER HOT  0.00000     1.071   P9od= 101.60   T9od= 474.12
9-10 EXHAUST            0.00000     1.071  P10od= 101.60  T10od= 474.12
PERFORMANCE
UWod=                   0.12912
SWod=                   0.12239
HIod=                   0.72476
mfod=                   0.01586
sfcod=                   0.1155
nthod=                   0.1782
niscod=                 0.84645
nistod=                 0.84645
Taod=                   273.0
Pa=                     101.30
Aod=                    0.00
Rcod=                    4.98
Qoutod=                 0.2477

-----------------------------------------------------------------------
ENGINE CONFIGURATION      FAR   MASS FLOW       P             T
-----------------------------------------------------------------------
    AMBIENT             0.00000     1.018   P1od= 101.30   T1od= 283.00
1-2 INTAKE              0.00000     1.018   P2od= 100.29   T2od= 283.00
2-3 COMPRESSOR          0.00000     1.018   P3od= 481.68   T3od= 472.83
3-4 PREMASS             0.00000     1.018   P4od= 481.68   T4od= 472.83
4-5 HEAT EXCHANGER COL  0.00000     1.018   P5od= 481.68   T5od= 472.83
5-6 BURNER              0.01476     1.033   P6od= 457.60   T6od=1000.00
6-7 MIXER               0.00000     1.033   P7od= 457.60   T7od=1000.00
7-8 COMPR & POWER TURB  0.00000     1.033   P8od= 101.60   T8od= 735.71
8-9 HEAT EXCHANGER HOT  0.00000     1.033   P9od= 101.60   T9od= 485.97
9-10 EXHAUST            0.00000     1.033  P10od= 101.60  T10od= 485.97
PERFORMANCE
UWod=                   0.11516
SWod=                   0.11316
HIod=                   0.68673
mfod=                   0.01502
sfcod=                   0.1227
nthod=                   0.1677
niscod=                 0.84341
nistod=                 0.84341
Taod=                   283.0
Pa=                     101.30
Aod=                    0.00
Rcod=                    4.80
Qoutod=                 0.2410

-----------------------------------------------------------------------
ENGINE CONFIGURATION      FAR   MASS FLOW       P             T
-----------------------------------------------------------------------
    AMBIENT             0.00000     1.055   P1od= 101.30   T1od= 273.00
1-2 INTAKE              0.00000     1.055   P2od= 100.29   T2od= 273.00
2-3 COMPRESSOR          0.00000     1.055   P3od= 523.70   T3od= 465.83
3-4 PREMASS             0.00000     1.055   P4od= 523.70   T4od= 465.83
4-5 HEAT EXCHANGER COL  0.00000     1.055   P5od= 523.70   T5od= 465.83
5-6 BURNER              0.01743     1.073   P6od= 497.51   T6od=1100.00
6-7 MIXER               0.00000     1.073   P7od= 497.51   T7od=1100.00
7-8 COMPR & POWER TURB  0.00000     1.073   P8od= 101.60   T8od= 792.09
8-9 HEAT EXCHANGER HOT  0.00000     1.073   P9od= 101.60   T9od= 482.14
9-10 EXHAUST            0.00000     1.073  P10od= 101.60  T10od= 482.14
PERFORMANCE
UWod=                   0.16911
SWod=                   0.16030
HIod=                   0.84063
mfod=                   0.01839
sfcod=                   0.1023
nthod=                   0.2012
niscod=                 0.85456
nistod=                 0.85456
Taod=                   273.0
Pa=                     101.30
Aod=                    0.00
Rcod=                    5.22
Qoutod=                 0.2582

-----------------------------------------------------------------------
ENGINE CONFIGURATION      FAR   MASS FLOW       P             T
-----------------------------------------------------------------------
    AMBIENT             0.00000     1.018   P1od= 101.30   T1od= 283.00
1-2 INTAKE              0.00000     1.018   P2od= 100.29   T2od= 283.00
```

2-3 COMPRESSOR	0.00000	1.018	P3od= 505.19	T3od= 478.16
3-4 PREMASS	0.00000	1.018	P4od= 505.19	T4od= 478.16
4-5 HEAT EXCHANGER COL	0.00000	1.018	P5od= 505.19	T5od= 478.16
5-6 BURNER	0.01716	1.035	P6od= 479.93	T6od=1100.00
6-7 MIXER	0.00000	1.035	P7od= 479.93	T7od=1100.00
7-8 COMPR & POWER TURB	0.00000	1.035	P8od= 101.60	T8od= 798.88
8-9 HEAT EXCHANGER HOT	0.00000	1.035	P9od= 101.60	T9od= 494.20
9-10 EXHAUST	0.00000	1.035	P10od= 101.60	T10od= 494.20
PERFORMANCE				
UWod=	0.15280			
SWod=	0.15015			
HIod=	0.79831			
mfod=	0.01747			
sfcod=	0.1075			
nthod=	0.1914			
niscod=	0.85149			
nistod=	0.85149			
Taod=	283.0			
Pa=	101.30			
Aod=	0.00			
Rcod=	5.04			
Qoutod=	0.2514			

1ShaftGTod HE: VARIATION OF Pamb

ENGINE CONFIGURATION	FAR	MASS FLOW	P	T
AMBIENT	0.00000	0.997	P1od= 101.00	T1od= 288.00
1-2 INTAKE	0.00000	0.997	P2od= 99.99	T2od= 288.00
2-3 COMPRESSOR	0.00000	0.997	P3od= 471.92	T3od= 478.85
3-4 PREMASS	0.00000	0.997	P4od= 471.92	T4od= 478.85
4-5 HEAT EXCHANGER COL	0.00000	0.997	P5od= 471.92	T5od= 478.85
5-6 BURNER	0.01463	1.012	P6od= 448.32	T6od=1000.00
6-7 MIXER	0.00000	1.012	P7od= 448.32	T7od=1000.00
7-8 COMPR & POWER TURB	0.00000	1.012	P8od= 101.30	T8od= 738.71
8-9 HEAT EXCHANGER HOT	0.00000	1.012	P9od= 101.30	T9od= 491.84
9-10 EXHAUST	0.00000	1.012	P10od= 101.30	T10od= 491.84
PERFORMANCE				
UWod=	0.10837			
SWod=	0.10869			
HIod=	0.66678			
mfod=	0.01459			
sfcod=	0.12661			
nthod=	0.1625			
niscod=	0.84194			
nistod=	0.84194			
Ta=	288.0			
Paod=	101.00			
Aod=	0.00			
Rcod=	4.72			
Qoutod=	0.2371			

ENGINE CONFIGURATION	FAR	MASS FLOW	P	T
AMBIENT	0.00000	1.017	P1od= 103.00	T1od= 288.00
1-2 INTAKE	0.00000	1.017	P2od= 101.97	T2od= 288.00
2-3 COMPRESSOR	0.00000	1.017	P3od= 481.26	T3od= 478.85
3-4 PREMASS	0.00000	1.017	P4od= 481.26	T4od= 478.85
4-5 HEAT EXCHANGER COL	0.00000	1.017	P5od= 481.26	T5od= 478.85
5-6 BURNER	0.01463	1.032	P6od= 457.20	T6od=1000.00
6-7 MIXER	0.00000	1.032	P7od= 457.20	T7od=1000.00
7-8 COMPR & POWER TURB	0.00000	1.032	P8od= 103.31	T8od= 738.71
8-9 HEAT EXCHANGER HOT	0.00000	1.032	P9od= 103.31	T9od= 491.84
9-10 EXHAUST	0.00000	1.032	P10od= 103.31	T10od= 491.84
PERFORMANCE				
UWod=	0.11051			
SWod=	0.10869			
HIod=	0.67998			
mfod=	0.01488			
sfcod=	0.12661			
nthod=	0.1625			
niscod=	0.84194			
nistod=	0.84194			
Ta=	288.0			
Paod=	103.00			
Aod=	0.00			
Rcod=	4.72			
Qoutod=	0.2418			

ENGINE CONFIGURATION	FAR	MASS FLOW	P	T
AMBIENT	0.00000	0.997	P1od= 101.00	T1od= 288.00
1-2 INTAKE	0.00000	0.997	P2od= 99.99	T2od= 288.00
2-3 COMPRESSOR	0.00000	0.997	P3od= 494.95	T3od= 484.27
3-4 PREMASS	0.00000	0.997	P4od= 494.95	T4od= 484.27
4-5 HEAT EXCHANGER COL	0.00000	0.997	P5od= 494.95	T5od= 484.27
5-6 BURNER	0.01703	1.014	P6od= 470.20	T6od=1100.00
6-7 MIXER	0.00000	1.014	P7od= 470.20	T7od=1100.00
7-8 COMPR & POWER TURB	0.00000	1.014	P8od= 101.30	T8od= 802.19

```
8-9 HEAT EXCHANGER HOT 0.00000     1.014   P9od= 101.30   T9od= 500.17
9-10 EXHAUST           0.00000     1.014  P10od= 101.30  T10od= 500.17
PERFORMANCE
UWod=                  0.14479
SWod=                  0.14522
HIod=                  0.77600
mfod=                  0.01698
sfcod=                 0.11027
nthod=                 0.1866
niscod=                0.85000
nistod=                0.85000
Ta=                    288.0
Paod=                  101.00
Aod=                   0.00
Rcod=                   4.95
Qoutod=                0.2474
-------------------------------------------------------------------------------
ENGINE CONFIGURATION      FAR   MASS FLOW       P              T
-------------------------------------------------------------------------------
    AMBIENT            0.00000     1.017   P1od= 103.00   T1od= 288.00
1-2 INTAKE             0.00000     1.017   P2od= 101.97   T2od= 288.00
2-3 COMPRESSOR         0.00000     1.017   P3od= 504.75   T3od= 484.27
3-4 PREMASS            0.00000     1.017   P4od= 504.75   T4od= 484.27
4-5 HEAT EXCHANGER COL 0.00000     1.017   P5od= 504.75   T5od= 484.27
5-6 BURNER             0.01703     1.034   P6od= 479.51   T6od=1100.00
6-7 MIXER              0.00000     1.034   P7od= 479.51   T7od=1100.00
7-8 COMPR & POWER TURB 0.00000     1.034   P8od= 103.31   T8od= 802.19
8-9 HEAT EXCHANGER HOT 0.00000     1.034   P9od= 103.31   T9od= 500.17
9-10 EXHAUST           0.00000     1.034  P10od= 103.31  T10od= 500.17
PERFORMANCE
UWod=                  0.14766
SWod=                  0.14522
HIod=                  0.79137
mfod=                  0.01731
sfcod=                 0.11027
nthod=                 0.1866
niscod=                0.85000
nistod=                0.85000
Ta=                    288.0
Paod=                  103.00
Aod=                   0.00
Rcod=                   4.95
Qoutod=                0.2523


                      1ShaftGTod HE: VARIATION OF Pamb
-------------------------------------------------------------------------------
ENGINE CONFIGURATION      FAR   MASS FLOW       P              T
-------------------------------------------------------------------------------
    AMBIENT            0.00000     1.000   P1od= 101.32   T1od= 288.15
1-2 INTAKE             0.00000     1.000   P2od= 100.31   T2od= 288.15
2-3 COMPRESSOR         0.00000     1.000   P3od= 473.19   T3od= 479.03
3-4 PREMASS            0.00000     1.000   P4od= 473.19   T4od= 479.03
4-5 HEAT EXCHANGER COL 0.00000     1.000   P5od= 473.19   T5od= 479.03
5-6 BURNER             0.01463     1.014   P6od= 449.53   T6od=1000.00
6-7 MIXER              0.00000     1.014   P7od= 449.53   T7od=1000.00
7-8 COMPR & POWER TURB 0.00000     1.014   P8od= 101.63   T8od= 738.79
8-9 HEAT EXCHANGER HOT 0.00000     1.014   P9od= 101.63   T9od= 492.02
9-10 EXHAUST           0.00000     1.014  P10od= 101.63  T10od= 492.02
PERFORMANCE
UWod=                  0.10852
SWod=                  0.10855
HIod=                  0.66839
mfod=                  0.01462
sfcod=                 0.12673
nthod=                 0.1624
niscod=                0.84189
nistod=                0.84189
Taod=                  288.1
Paod=                  101.32
Aod=                      0.00
Rcod=                    4.72
Qoutod=                0.2378
-------------------------------------------------------------------------------
ENGINE CONFIGURATION      FAR   MASS FLOW       P              T
-------------------------------------------------------------------------------
    AMBIENT            0.00000     0.976   P1od=  98.36   T1od= 286.52
1-2 INTAKE             0.00000     0.976   P2od=  97.37   T2od= 286.52
2-3 COMPRESSOR         0.00000     0.976   P3od= 461.94   T3od= 477.08
3-4 PREMASS            0.00000     0.976   P4od= 461.94   T4od= 477.08
4-5 HEAT EXCHANGER COL 0.00000     0.976   P5od= 461.94   T5od= 477.08
5-6 BURNER             0.01467     0.990   P6od= 438.84   T6od=1000.00
6-7 MIXER              0.00000     0.990   P7od= 438.84   T7od=1000.00
7-8 COMPR & POWER TURB 0.00000     0.990   P8od=  98.65   T8od= 737.83
8-9 HEAT EXCHANGER HOT 0.00000     0.990   P9od=  98.65   T9od= 490.11
9-10 EXHAUST           0.00000     0.990  P10od=  98.65  T10od= 490.11
PERFORMANCE
UWod=                  0.10735
SWod=                  0.11000
```

```
HIod=                   0.65441
mfod=                   0.01432
sfcod=                  0.12543
nthod=                  0.1640
niscod=                 0.84237
nistod=                 0.84237
Taod=                   286.5
Paod=                    98.36
Aod=                     250.00
Rcod=                     4.74
Qoutod=                 0.2318
-------------------------------------------------------------------------------
ENGINE CONFIGURATION      FAR   MASS FLOW       P              T
-------------------------------------------------------------------------------
    AMBIENT             0.00000     1.000   P1od= 101.32   T1od= 288.15
1-2 INTAKE              0.00000     1.000   P2od= 100.31   T2od= 288.15
2-3 COMPRESSOR          0.00000     1.000   P3od= 496.28   T3od= 484.46
3-4 PREMASS             0.00000     1.000   P4od= 496.28   T4od= 484.46
4-5 HEAT EXCHANGER COL  0.00000     1.000   P5od= 496.28   T5od= 484.46
5-6 BURNER              0.01702     1.017   P6od= 471.47   T6od=1100.00
6-7 MIXER               0.00000     1.017   P7od= 471.47   T7od=1100.00
7-8 COMPR & POWER TURB  0.00000     1.017   P8od= 101.63   T8od= 802.29
8-9 HEAT EXCHANGER HOT  0.00000     1.017   P9od= 101.63   T9od= 500.35
9-10 EXHAUST            0.00000     1.017  P10od= 101.63  T10od= 500.35
PERFORMANCE
UWod=                   0.14504
SWod=                   0.14508
HIod=                   0.77791
mfod=                   0.01702
sfcod=                  0.11036
nthod=                  0.1864
niscod=                 0.84996
nistod=                 0.84996
Taod=                   288.1
Paod=                   101.32
Aod=                       0.00
Rcod=                     4.95
Qoutod=                 0.2481
-------------------------------------------------------------------------------
ENGINE CONFIGURATION      FAR   MASS FLOW       P              T
-------------------------------------------------------------------------------
    AMBIENT             0.00000     0.976   P1od=  98.36   T1od= 286.52
1-2 INTAKE              0.00000     0.976   P2od=  97.37   T2od= 286.52
2-3 COMPRESSOR          0.00000     0.976   P3od= 484.48   T3od= 482.47
3-4 PREMASS             0.00000     0.976   P4od= 484.48   T4od= 482.47
4-5 HEAT EXCHANGER COL  0.00000     0.976   P5od= 484.48   T5od= 482.47
5-6 BURNER              0.01707     0.993   P6od= 460.26   T6od=1100.00
6-7 MIXER               0.00000     0.993   P7od= 460.26   T7od=1100.00
7-8 COMPR & POWER TURB  0.00000     0.993   P8od=  98.65   T8od= 801.22
8-9 HEAT EXCHANGER HOT  0.00000     0.993   P9od=  98.65   T9od= 498.41
9-10 EXHAUST            0.00000     0.993  P10od=  98.65  T10od= 498.41
PERFORMANCE
UWod=                   0.14314
SWod=                   0.14667
HIod=                   0.76135
mfod=                   0.01666
sfcod=                  0.10944
nthod=                  0.1880
niscod=                 0.85044
nistod=                 0.85044
Taod=                   286.5
Paod=                    98.36
Aod=                     250.00
Rcod=                     4.98
Qoutod=                 0.2419
```

APPENDIX D

D.1 Saturated water – Temperature table [54]

Table D.1: Saturated water – Temperature table

		Specific volume, m^3/kg		Internal energy, kJ/kg			Enthalpy, kJ/kg			Entropy, kJ/(kg · K)		
Temp., T °C	Sat. press., P_{sat} kPa	Sat. liquid, v_f	Sat. vapor, v_g	Sat. liquid, u_f	Evap., u_{fg}	Sat. vapor, u_g	Sat. liquid, h_f	Evap., h_{fg}	Sat. vapor, h_g	Sat. liquid, s_f	Evap., s_{fg}	Sat. vapor, s_g
0.01	0.6113	0.001000	206.14	0.0	2375.3	2375.3	0.01	2501.3	2501.4	0.000	9.1562	9.1562
5	0.8721	0.001000	147.12	20.97	2361.3	2382.3	20.98	2489.6	2510.6	0.0761	8.9496	9.0257
10	1.2276	0.001000	106.38	42.00	2347.2	2389.2	42.01	2477.7	2519.8	0.1510	8.7498	8.9008
15	1.7051	0.001001	77.93	62.99	2333.1	2396.1	62.99	2465.9	2528.9	0.2245	8.5569	8.7814
20	2.339	0.001002	57.79	83.95	2319.0	2402.9	83.96	2454.1	2538.1	0.2966	8.3706	8.6672
25	3.169	0.001003	43.36	104.88	2304.9	2409.8	104.89	2442.3	2547.2	0.3674	8.1905	8.5580
30	4.246	0.001004	32.89	125.78	2290.8	2416.6	125.79	2430.5	2556.3	0.4369	8.0164	8.4533
35	5.628	0.001006	25.22	146.67	2276.7	2423.4	146.68	2418.6	2565.3	0.5053	7.8478	8.3531
40	7.384	0.001008	19.52	167.56	2262.6	2430.1	167.57	2406.7	2574.3	0.5725	7.6845	8.2570
45	9.593	0.001010	15.26	188.44	2248.4	2436.8	188.45	2394.8	2583.2	0.6387	7.5261	8.1648
50	12.349	0.001012	12.03	209.32	2234.2	2443.5	209.33	2382.7	2592.1	0.7038	7.3725	8.0763
55	15.758	0.001015	9.568	230.21	2219.9	2450.1	230.23	2370.7	2600.9	0.7679	7.2234	7.9913
60	19.940	0.001017	7.671	251.11	2205.5	2456.6	251.13	2358.5	2609.6	0.8312	7.0784	7.9096
65	25.03	0.001020	6.197	272.02	2191.1	2463.1	272.06	2346.2	2618.3	0.8935	6.9375	7.8310
70	31.19	0.001023	5.042	292.95	2176.6	2469.6	292.98	2333.8	2626.8	0.9549	6.8004	7.7553
75	38.58	0.001026	4.131	313.90	2162.0	2475.9	313.93	2321.4	2635.3	1.0155	6.6669	7.6824
80	47.39	0.001029	3.407	334.86	2147.4	2482.2	334.91	2308.8	2643.7	1.0753	6.5369	7.6122
85	57.83	0.001033	2.828	355.84	2132.6	2488.4	355.90	2296.0	2651.9	1.1343	6.4102	7.5445
90	70.14	0.001036	2.361	376.85	2117.7	2494.5	376.92	2283.2	2660.1	1.1925	6.2866	7.4791
95	84.55	0.001040	1.982	397.88	2102.7	2500.6	397.96	2270.2	2668.1	1.2500	6.1659	7.4159
	Sat. press., MPa											
100	0.10135	0.001044	1.6729	418.94	2087.6	2506.5	419.04	2257.0	2676.1	1.3069	6.0480	7.3549
105	0.12082	0.001048	1.4194	440.02	2072.3	2512.4	440.15	2243.7	2683.8	1.3630	5.9328	7.2958
110	0.14327	0.001052	1.2102	461.14	2057.0	2518.1	461.30	2230.2	2691.5	1.4185	5.8202	7.2387
115	0.16906	0.001056	1.0366	482.30	2041.4	2523.7	482.48	2216.5	2699.0	1.4734	5.7100	7.1833
120	0.19853	0.001060	0.8919	503.50	2025.8	2529.3	503.71	2202.6	2706.3	1.5276	5.6020	7.1296
125	0.2321	0.001065	0.7706	524.74	2009.9	2534.6	524.99	2188.5	2713.5	1.5813	5.4962	7.0775
130	0.2701	0.001070	0.6685	546.02	1993.9	2539.9	546.31	2174.2	2720.5	1.6344	5.3925	7.0269
135	0.3130	0.001075	0.5822	567.35	1977.7	2545.0	567.69	2159.6	2727.3	1.6870	5.2907	6.9777
140	0.3613	0.001080	0.5089	588.74	1961.3	2550.0	589.13	2144.7	2733.9	1.7391	5.1908	6.9299
145	0.4154	0.001085	0.4463	610.18	1944.7	2554.9	610.63	2129.6	2740.3	1.7907	5.0926	6.8833
150	0.4758	0.001091	0.3928	631.68	1927.9	2559.5	632.20	2114.3	2746.5	1.8418	4.9960	6.8379
155	0.5431	0.001096	0.3468	653.24	1910.8	2564.1	653.84	2098.6	2752.4	1.8925	4.9010	6.7935
160	0.6178	0.001102	0.3071	674.87	1893.5	2568.4	675.55	2082.6	2758.1	1.9427	4.8075	6.7502
165	0.7005	0.001108	0.2727	696.56	1876.0	2572.5	697.34	2066.2	2763.5	1.9925	4.7153	6.7078
170	0.7917	0.001114	0.2428	718.33	1858.1	2576.5	719.21	2049.5	2768.7	2.0419	4.6244	6.6663
175	0.8920	0.001121	0.2168	740.17	1840.0	2580.2	741.17	2032.4	2773.6	2.0909	4.5347	6.6256
180	1.0021	0.001127	0.19405	762.09	1821.6	2583.7	763.22	2015.0	2778.2	2.1396	4.4461	6.5857
185	1.1227	0.001134	0.17409	784.10	1802.9	2587.0	785.37	1997.1	2782.4	2.1879	4.3586	6.5465
190	1.2544	0.001141	0.15654	806.19	1783.8	2590.0	807.62	1978.8	2786.4	2.2359	4.2720	6.5079
195	1.3978	0.001149	0.14105	828.37	1764.4	2592.8	829.98	1960.0	2790.0	2.2835	4.1863	6.4698

Table D.2: Saturated water – Temperature table (continue)

		Specific volume, m^3/kg		Internal energy, kJ/kg			Enthalpy, kJ/kg			Entropy, kJ/(kg · K)		
Temp., T °C	Sat. press., P_{sat} MPa	Sat. liquid, v_f	Sat. vapor, v_g	Sat. liquid, u_f	Evap., u_{fg}	Sat. vapor, u_g	Sat. liquid, h_f	Evap., h_{fg}	Sat. vapor, h_g	Sat. liquid, s_f	Evap., s_{fg}	Sat. vapor, s_g
200	1.5538	0.001157	0.13736	850.65	1744.7	2595.3	852.45	1940.7	2793.2	2.3309	4.1014	6.4323
205	1.7230	0.001164	0.11521	873.04	1724.5	2597.5	875.04	1921.0	2796.0	2.3780	4.0172	6.3952
210	1.9062	0.001173	0.10441	895.53	1703.9	2599.5	897.76	1900.7	2798.5	2.4248	3.9337	6.3585
215	2.104	0.001181	0.09479	918.14	1682.9	2601.1	920.62	1879.9	2800.5	2.4714	3.8507	6.3221
220	2.318	0.001190	0.08619	940.87	1661.5	2602.4	943.62	1858.5	2802.1	2.5178	3.7683	6.2861
225	2.548	0.001199	0.07849	963.73	1639.6	2603.3	966.78	1836.5	2803.3	2.5639	3.6863	6.2503
230	2.795	0.001209	0.07158	986.74	1617.2	2603.9	990.12	1813.8	2804.0	2.6099	3.6047	6.2146
235	3.060	0.001219	0.06537	1009.89	1594.2	2604.1	1013.62	1790.5	2804.2	2.6558	3.5233	6.1791
240	3.344	0.001229	0.05976	1033.21	1570.8	2604.0	1037.32	1766.5	2803.8	2.7015	3.4422	6.1437
245	3.648	0.001240	0.05471	1056.71	1546.7	2603.4	1061.23	1741.7	2803.0	2.7472	3.3612	6.1083
250	3.973	0.001251	0.05013	1080.39	1522.0	2602.4	1085.36	1716.2	2801.5	2.7927	3.2802	6.0730
255	4.319	0.001263	0.04598	1104.28	1596.7	2600.9	1109.73	1689.8	2799.5	2.8383	3.1992	6.0375
260	4.688	0.001276	0.04221	1128.39	1470.6	2599.0	1134.37	1662.5	2796.9	2.8838	3.1181	6.0019
265	5.081	0.001289	0.03877	1152.74	1443.9	2596.6	1159.28	1634.4	2793.6	2.9294	3.0368	5.9662
270	5.499	0.001302	0.03564	1177.36	1416.3	2593.7	1184.51	1605.2	2789.7	2.9751	2.9551	5.9301
275	5.942	0.001317	0.03279	1202.25	1387.9	2590.2	1210.07	1574.9	2785.0	3.0208	2.8730	5.8938
280	6.412	0.001332	0.03017	1227.46	1358.7	2586.1	1235.99	1543.6	2779.6	3.0668	2.7903	5.8571
285	6.909	0.001348	0.02777	1253.00	1328.4	2581.4	1262.31	1511.0	2773.3	3.1130	2.7070	5.8199
290	7.436	0.001366	0.02557	1278.92	1297.1	2576.0	1289.07	1477.1	2766.2	3.1594	2.6227	5.7821
295	7.993	0.001384	0.02354	1305.2	1264.7	2569.9	1316.3	1441.8	2758.1	3.2062	2.5375	5.7437
300	8.581	0.001404	0.02167	1332.0	1231.0	2563.0	1344.0	1404.9	2749.0	3.2534	2.4511	5.7045
305	9.202	0.001425	0.019948	1359.3	1195.9	2555.2	1372.4	1366.4	2738.7	3.3010	2.3633	5.6643
310	9.856	0.001447	0.018350	1387.1	1159.4	2546.4	1401.3	1326.0	2727.3	3.3493	2.2737	5.6230
315	10.547	0.001472	0.016867	1415.5	1121.1	2536.6	1431.0	1283.5	2714.5	3.3982	2.1821	5.5804
320	11.274	0.001499	0.015488	1444.6	1080.9	2525.5	1461.5	1238.6	2700.1	3.4480	2.0882	5.5362
330	12.845	0.001561	0.012996	1505.3	993.7	2498.9	1525.3	1140.6	2665.9	3.5507	1.8909	5.4417
340	14.586	0.001638	0.010797	1570.3	894.3	2464.6	1594.2	1027.9	2622.0	3.6594	1.6763	5.3357
350	16.513	0.001740	0.008813	1641.9	776.6	2418.4	1670.6	893.4	2563.9	3.7777	1.4335	5.2112
360	18.651	0.001893	0.006945	1725.2	626.3	2351.5	1760.5	720.3	2481.0	3.9147	1.1379	5.0526
370	21.03	0.002213	0.004925	1844.0	384.5	2228.5	1890.5	441.6	2332.1	4.1106	0.6865	4.7971
374.14	22.09	0.003155	0.003155	2029.6	0	2029.6	2099.3	0	2099.3	4.4298	0	4.4298

D.2 LiBr/Water solution specific enthalpy [60]

The specific enthalpy of the solution (in kJ/kgr), is the sum of the salt enthalpy plus the enthalpy of saturated pure water plus an equivalent coefficient:

$$\mathrm{h(t, x)} = x \cdot h_{LiBr}(t) + (1-x) \cdot h_w(t) + \Delta h(t, x) \tag{D-1}$$

where:

t, is temperature in ^{o}C,

x, is the mass fraction (for example x=0.5),

$h_{LiBr}(t)$, is the salt enthalpy and can be calculated as:

$$h_{LiBr}(t) = \sum_{i=0}^{4} a_i \cdot t^i \tag{D-2}$$

where:

a_i takes the corresponding value from the following values:

$$a_i = \begin{Bmatrix} 508.68 \\ -118.6241 \\ 9.85946 \cdot 10^{-2} \\ -2.50979 \cdot 10^{-5} \\ 4.15801 \cdot 10^{-8} \end{Bmatrix}$$

$h_w(t)$, is the enthalpy of saturated pure water and can be calculated as:

$$h_w(t) = b_0 + b_1 \cdot t + b_2 \cdot t^2 + b_3 \cdot t^3 \tag{D-3}$$

where:

$$b_i = \begin{Bmatrix} 0.22156863 \\ 4.1969 \\ -0.0008993808 \\ 0.00000400756794 \end{Bmatrix}$$

$\Delta h(t, x)$, is equivalent coefficient and can be calculated as:

$$\Delta h(t, x) = x \cdot (1-x) \cdot \sum_{k=0}^{4} \sum_{j=0}^{3} b_{kj} \cdot (2 \cdot x - 1)^k \cdot t^j \tag{D-4}$$

where:

$$b_{kj} = \begin{Bmatrix} -1021.61 & 36.8773 & -0.186051 & -0.00000751277 \\ -533.308 & 40.2847 & -0.191198 & 0 \\ 483.628 & 39.9142 & -0.199213 & 0 \\ 1155.13 & 33.3572 & -0.1782580 & 0 \\ 640.622 & 13.1032 & -0.07751010 & 0 \end{Bmatrix}$$

Notice: the above values are **not** considered as elements of a mathematic table.

D.3 LiBr/Water solution density [60]

$$\rho(t,x) = \sum_{j=1}^{5} x^{j-1}\left[c_{0,j-1} + \left(c_{1,j-1} + t \cdot c_{2,j-1}\right)\right] \qquad \text{(D-5)}$$

where:

$\rho(t,x)$, in kgr/m^3

t, is temperature in ^{o}C,

c are given by:

$$c = \begin{Bmatrix} 999{,}1 & 7{,}74931 & 0{,}00536509 & 0{,}00134988 & -0{,}00000308671 \\ -0{,}0239865 & -0{,}0128346 & 0{,}000207232 & -0{,}00000908213 & 0{,}0000000994788 \\ -0{,}00390453 & -0{,}0000555855 & 0{,}0000109879 & -0{,}000000239834 & 0{,}00000000153514 \end{Bmatrix}$$

Notice: the above values are **not** considered as elements of a mathematic table.

D.4 LiBr/Water solution temperature determination method [66]

$$h(t,x) = A + B \cdot t + C \cdot t^2 \qquad \text{(D-6)}$$

where:

h is in kJ/kgr

t, is temperature in ^{o}F [^{o}C=(^{o}F-32)/1.8],

A= $-1015.07 + 79.5387 \cdot x - 2{,}358016 \cdot x^2 + 0.03031583 \cdot x^3 - 1.400261 \cdot 10^{-4} \cdot x^4$

B= $4.68108 - 3.037766 \cdot 10^{-1} \cdot x + 8.44845 \cdot 10^{-3} \cdot x^2 - 1.047721 \cdot 10^{-4} \cdot x^3 + 4.80097 \cdot 10^{-7} \cdot x^4$

C= $-4.9107 \cdot 10^{-3} + 3.83184 \cdot 10^{-4} \cdot x - 1.078963 \cdot 10^{-5} \cdot x^2 + 1.3152 \cdot 10^{-7} \cdot x^3 - 5.897 \cdot 10^{-10} \cdot x^4$

where:

$x = 40 \div 70$

Thus, from: $\qquad$ (D-6) $\Rightarrow$ $t = \dfrac{-B + \sqrt{\Delta}}{2 \cdot C}$ $\qquad$ (D-7)

where:

$\Delta = B^2 - 4 \cdot C \cdot (A - h)$

D.5 Absorption cooling simulation program input file

```
Give strong solution mass fraction, x4 (0.6-0.65)
0.646
Give concentration difference between strong-weak solution, Dx (0.04-0.055)
0.051
Give absorber weak solution outlet temperature, t2 (28-45oC)
42.4
Give temperature difference of the solution heat exchanger, Dt (10-20oC)
15.8
Give desorber solution outlet temperature, t4 (80-120oC)
98.8
Give evaporator temperature, t10 (5-7oC)
5.1
Give liquid carryover percentage from evaporator, Dm (0.02-0.03)
0.025
Give low pressure, p1 (0.86-1.05kPa)
0.876
Give high pressure, p2 (8.6-10.5kPa)
8.687
Give solution pump isentropic efficiency, pump(0.9-0.99)
0.99
Give gas turbine exhaust heat exchanger effectiveness, nEHE(0.70-0.90)
0.8
```

D.6 Absorption cooling simulation program output file (Qe: 300÷2100kW with step 300kW)

Point	h(kJ/kgr)	mc(kgr/sec)	pc(kPa)	T(K)	x
1	103.4655	1673.579	0.876	308.03	0.5950
2	116.6210	1673.579	8.687	315.40	0.5950
3	183.4489	1673.579	8.687	349.15	0.5950
4	247.6344	1541.454	8.687	371.80	0.6460
5	175.0784	1541.454	8.687	331.20	0.6460
6	175.0784	1541.454	0.876	330.64	0.6460
7	2664.5952	132.125	8.687	365.80	0.0000
8	179.4170	132.125	8.687	315.73	0.0000
9	179.4170	132.125	0.876	278.10	0.0000
10	2510.7185	128.902	0.876	278.10	0.0000
11	21.2496	3.223	0.876	278.10	0.0000

```
   COP=      0.68195
 effHE=      0.71986
    Qa=    420.4230652 MW
    Qc=    328.3533325 MW
    Qd=    426.7596130 MW
    Qe=    300.0000000 MW
     W=     13.1555233 MW
QEGTco=    362.7456665 MW
```

Point	h(kJ/kgr)	mc(kgr/sec)	pc(kPa)	T(K)	x
1	90.1536	3347.158	0.876	301.18	0.5950
2	116.6210	3347.158	8.687	315.40	0.5950
3	183.4489	3347.158	8.687	349.15	0.5950
4	247.6344	3082.909	8.687	371.80	0.6460
5	175.0784	3082.909	8.687	331.20	0.6460
6	175.0784	3082.909	0.876	330.64	0.6460
7	2664.5952	264.249	8.687	365.80	0.0000
8	179.4170	264.249	8.687	315.73	0.0000
9	179.4170	264.249	0.876	278.10	0.0000
10	2510.7185	257.804	0.876	278.10	0.0000
11	21.2496	6.445	0.876	278.10	0.0000

```
   COP=      0.68183
 effHE=      0.71986
    Qa=    885.4033203 MW
    Qc=    656.7066650 MW
    Qd=    853.5192261 MW
    Qe=    600.0000000 MW
     W=     26.4674606 MW
QEGTco=    725.4913330 MW
```

Point	h(kJ/kgr)	mc(kgr/sec)	pc(kPa)	T(K)	x
1	76.9961	5020.738	0.876	294.41	0.5950
2	116.6210	5020.738	8.687	315.40	0.5950
3	183.4489	5020.738	8.687	349.15	0.5950
4	247.6344	4624.364	8.687	371.80	0.6460
5	175.0784	4624.364	8.687	331.20	0.6460
6	175.0784	4624.364	0.876	330.64	0.6460
7	2664.5952	396.374	8.687	365.80	0.0000
8	179.4170	396.374	8.687	315.73	0.0000
9	179.4170	396.374	0.876	278.10	0.0000
10	2510.7185	386.706	0.876	278.10	0.0000
11	21.2496	9.668	0.876	278.10	0.0000

```
   COP=      0.68187
 effHE=      0.71986
    Qa=   1394.1654053 MW
    Qc=    985.0601196 MW
    Qd=   1280.2789307 MW
    Qe=    900.0000000 MW
     W=     39.6249733 MW
QEGTco=   1088.2371826 MW
```

Point	h(kJ/kgr)	mc(kgr/sec)	pc(kPa)	T(K)	x
1	63.8687	6694.316	0.876	287.65	0.5950
2	116.6210	6694.316	8.687	315.40	0.5950
3	183.4489	6694.316	8.687	349.15	0.5950
4	247.6344	6165.818	8.687	371.80	0.6460
5	175.0784	6165.818	8.687	331.20	0.6460
6	175.0784	6165.818	0.876	330.64	0.6460
7	2664.5952	528.499	8.687	365.80	0.0000
8	179.4170	528.499	8.687	315.73	0.0000
9	179.4170	528.499	0.876	278.10	0.0000
10	2510.7185	515.608	0.876	278.10	0.0000
11	21.2496	12.890	0.876	278.10	0.0000

```
   COP=      0.68190
 effHE=      0.71986
```

Qa= 1946.7658691 MW
Qc= 1313.4133301 MW
Qd= 1707.0384521 MW
Qe= 1200.0000000 MW
W= 52.7523308 MW
QEGTco= 1450.9826660 MW

Point	h(kJ/kgr)	mc(kgr/sec)	pc(kPa)	T(K)	x
1	50.7577	8367.896	0.876	280.90	0.5950
2	116.6210	8367.896	8.687	315.40	0.5950
3	183.4489	8367.896	8.687	349.15	0.5950
4	247.6344	7707.272	8.687	371.80	0.6460
5	175.0784	7707.272	8.687	331.20	0.6460
6	175.0784	7707.272	0.876	330.64	0.6460
7	2664.5952	660.623	8.687	365.80	0.0000
8	179.4170	660.623	8.687	315.73	0.0000
9	179.4170	660.623	0.876	278.10	0.0000
10	2510.7185	644.511	0.876	278.10	0.0000
11	21.2496	16.113	0.876	278.10	0.0000

COP= 0.68192
effHE= 0.71986
Qa= 2543.1689453 MW
Qc= 1641.7668457 MW
Qd= 2133.7978516 MW
Qe= 1500.0000000 MW
W= 65.8633499 MW
QEGTco= 1813.7282715 MW

Point	h(kJ/kgr)	mc(kgr/sec)	pc(kPa)	T(K)	x
1	37.6490	10041.476	0.876	274.15	0.5950
2	116.6210	10041.476	8.687	315.40	0.5950
3	183.4489	10041.476	8.687	349.15	0.5950
4	247.6344	9248.728	8.687	371.80	0.6460
5	175.0784	9248.728	8.687	331.20	0.6460
6	175.0784	9248.728	0.876	330.64	0.6460
7	2664.5952	792.748	8.687	365.80	0.0000
8	179.4170	792.748	8.687	315.73	0.0000
9	179.4170	792.748	0.876	278.10	0.0000
10	2510.7185	773.413	0.876	278.10	0.0000
11	21.2496	19.335	0.876	278.10	0.0000

COP= 0.68194
effHE= 0.71986
Qa= 3183.4338379 MW
Qc= 1970.1202393 MW
Qd= 2560.5578613 MW
Qe= 1800.0000000 MW
W= 78.9720688 MW
QEGTco= 2176.4743652 MW

Point	h(kJ/kgr)	mc(kgr/sec)	pc(kPa)	T(K)	x
1	24.5286	11715.055	0.876	267.39	0.5950
2	116.6210	11715.055	8.687	315.40	0.5950
3	183.4489	11715.055	8.687	349.15	0.5950
4	247.6344	10790.182	8.687	371.80	0.6460
5	175.0784	10790.182	8.687	331.20	0.6460
6	175.0784	10790.182	0.876	330.64	0.6460
7	2664.5952	924.873	8.687	365.80	0.0000
8	179.4170	924.873	8.687	315.73	0.0000
9	179.4170	924.873	0.876	278.10	0.0000
10	2510.7185	902.315	0.876	278.10	0.0000
11	21.2496	22.558	0.876	278.10	0.0000

COP= 0.68195
effHE= 0.71986
Qa= 3867.7121582 MW
Qc= 2298.4733887 MW
Qd= 2987.3171387 MW
Qe= 2100.0000000 MW
W= 92.0924759 MW
QEGTco= 2539.2197266 MW

APPENDIX E

E.1 Genset Plant Prices

Equipment-only prices for a skid-mounted single fuel gas turbine, electric generator, air intake with basic filter and silencer, exhaust stack, basic starter and controls, gearbox (if needed), conventional combustion system (unless otherwise designated as D or DLE for dry low emissions design).

Quoted FOB (Free On Board, i.e. excluding shipment and installation costs) the factory in 2004 US dollars. Prices can vary considerably depending on the scope of plant equipment, geographical area, special site requirements, currency fluctuations and competitive market conditions. [79]

Model	Base Load Rating	Heat Rate Btu/kWh	LHV Efficiency	Budget Plant Price	Price per kW
VPS1	515 kW	15,980 Btu	21.3%	$445,000	$864
ST6L-813	848 kW	13,100 Btu	26.0%	$678,000	$799
Makila TI	1050 kW	12,580 Btu	27.1%	$880,000	$838
Saturn 20	1210 kW	14,025 Btu	24.3%	$698,000	$577
Heron H-1	1407 kW	7953 Btu	42.9%	$1,065,000	$755
M1A13D	1475 kW	14,230 Btu	24.0%	$940,000	$638
KG2-3C	1500 kW	21,200 Btu	16.1%	$1,100,000	$736
KG2-3E	1895 kW	21,420 Btu	15.9%	$1,240,000	$654
ST18A	1960 kW	11,240 Btu	30.4%	$1,200,000	$611
OGT2500	2730 kW	12,515 Btu	27.3%	$1,435,000	$526
UGT-2500	2850 kW	11,970 Btu	28.5%	$1,390,000	$488
M1T13D	2900 kW	14,440 Btu	23.6%	$1,625,000	$560
VPS3	3105 kW	12,675 Btu	26.9%	$1,530,000	$493
Centaur 40	3520 kW	12,240 Btu	27.9%	$1,520,000	$432
VPS4	3570 kW	11,715 Btu	29.1%	$1,610,000	$451
501-KB5	3940 kW	11,626 Btu	29.4%	$1,675,000	$425
GTES-4	4100 kW	14,130 Btu	24.1%	$1,230,000	$300
ST40	4040 kW	10,310 Btu	33.1%	$1,800,000	$446
Centaur 50	4600 kW	11,630 Btu	29.3%	$1,700,000	$370
Mercury 50	4600 kW	8865 Btu	38.5%	$2,200,000	$478
PGT5	5220 kW	12,720 Btu	26.8%	$1,900,000	$364
Typhoon 5.25	5250 kW	11,200 Btu	30.5%	$2,090,500	$398
501-KB7	5300 kW	10.790 Btu	31.6%	$1,848,000	$348
M7A-01D	5380 kW	11,650 Btu	29.3%	$1,900,000	$353
Taurus 60	5500 kW	11,225 Btu	30.4%	$1,960,000	$356
M7A-01	5512 kW	11,530 Btu	29.6%	$1,800,000	$326
GE5	5520 kW	11,130 Btu	30.7%	$2,000,000	$364
THM1203A	5760 kW	15,185 Btu	22.5%	$2,970,000	$515
PGT5B	5900 kW	10,700 Btu	31.9%	$2,100,000	$347
GTES-6	6200 kW	12,780 Btu	26.7%	$1,705,000	$275
501-KH5 (steam injection)	6420 kW	8560 Btu	39.9%	$2,415,000	$376
UGT-6000	6700 kW	10,830 Btu	31.5%	$2,120,000	$316
M7A-02D	6720 kW	11,260 Btu	30.3%	$2,250,000	$335

Tornado	6750 kW	10,820 Btu	31.5%	$2,660,000	$394
M7A-02	6915 kW	11,190 Btu	30.5%	$2,150,000	$311
Taurus 70	7520 kW	10,100 Btu	33.8%	$2,670,000	$355
Tempest	7910 kW	10,940 Btu	31.2%	$2,945,000	$372
UGT-6000+	8300 kW	10,650 Btu	32.0%	$2,350,000	$283
THM1304-9	8640 kW	12,340 Btu	27.6%	$3,860,000	$447
THM1304-10	9320 kW	12,170 Btu	28.0%	$3,980,000	$427
Mars 90	9450 kW	10,710 Btu	31.9%	$3,600,000	$381
UGT-10000	10,000 kW	10,220 Btu	34.2%	$3,350,000	$335
G3142J	10,450 kW	13,320 Btu	25.6%	$3,750,000	$359
Mars 100	10,690 kW	10,520 Btu	32.4%	$4,050,000	$379
THM1304-11	10,760 kW	11,460 Btu	29.8%	$4,230,000	$393
PGT10B	11,700 kW	10,660 Btu	32.0%	$4,700,000	$402
GTES-12	12,000 kW	10,240 Btu	33.3%	$3,000,000	$250
Cyclone DLE	12,875 kW	9820 Btu	34.8%	$4,650,000	$361
SB60-1	13,570 kW	11,490 Btu	29.7%	$5,930,000	$437
PGT16	13,750 kW	9670 Btu	35.3%	$6,750,000	$491
LM1600PE	13,750 kW	9750 Btu	35.0%	$7,000,000	$509
LM1600DLE	13,750 kW	9865 Btu	34.6%	$7,500,000	$545
H-15	13,800 kW	11,010 Btu	31.0%	$4,900,000	$355

Model	Base Load Rating	Heat Rate Btu/kWh	LHV Efficiency	Budget Plant Price	Price per kW
Titan 130	14,270 kW	9750 Btu	35.0%	$4,875,000	$342
MF111B	14,570 kW	11,020 Btu	31.0%	$6,200,000	$425
Avon	14,580 kW	12,100 Btu	28.2%	$5,376,000	$368
GTES-16	16,000 kW	9790 Btu	34.9%	$4,000,000	$250
UGT-10000 STIG. (steam injection)	16,000 kW	7950 Btu	43.0%	$4,500,000	$281
UGT-16000	16,300 kW	11,230 Btu	30.4%	$3,950,000	$242
LM1600-PB STIG (steam injection)	16,900 kW	8605 Btu	39.7%	$8,280,000	$490
GT35	17,000 kW	10,600 Btu	32.2%	$6,000,000	$353
UGT-15000	17,500 kW	9750 Btu	35.0%	$5,275,000	$301
L20A	17,640 kW	9950 Btu	34.3%	$5,500,000	$312
LM2000	19,500 kW	9810 Btu	34.8%	$8,190,000	$420
UGT-15000+	20,000 kW	9970 Btu	34.2%	$6,100,000	$305
PGT25	22,450 kW	9395 Btu	36.3%	$9,200,000	$410
LM2500PE.	22,800 kW	9280 Btu	36.8%	$9,175,000	$402
GT10B	24,770 kW	9985 Btu	34.2%	$7,495,000	$303
UGT-15000 STIG (steam injection)	25,000 kW	8130 Btu	42.0%	$6,700,000	$268
RB211-6556DLE	24,125 kW	9985 Btu	34.2%	$7,900,000	$327
FT8	25,490 kW	8950 Btu	38.1%	$9,400,000	$368
UGT-25000	26,700 kW	9310 Btu	36.6%	$6,940,000	$260
PG5371PA	26,830 kW	12,025 Btu	28.4%	$6,680,000	$249
H-25	27,500 kW	10,097 Btu	33.8%	$7,290,000	$265
RB211-6562 DLE	27,520 kW	9415 Btu	36.2%	$10,074,000	$323
LM2500PH (steam injection)	28,280 kW	8325 Btu	41.0%	$10,500,000	$371
LM2500+PK	28,600 kW	8860 Btu	38.5%	$9,500,000	$332
GT10C	29,060 kW	9480 Btu	36.0%	$8,495,000	$292
RB211-6562	29,500 kW	9225 Btu	37.0%	$9,256,000	$314
RB211-6762 DLE	29,500 kW	9055 Btu	37.7%	$9,890,000	$335
MS5002E	29,680 kW	9570 Btu	35.7%	$7,700,000	$259
MF-221	30,000 kW	10,670 Btu	32.0%	$10,000,000	$333
RB211-6761 DLE	32,120 kW	8680 Btu	39.3%	$10,610,000	$330

Model	Base Load Rating	Heat Rate Btu/kWh	LHV Efficiency	Budget Plant Price	Price per kW
PG6561B	39,620 kW	10,710 Btu	31.9%	$10,100,000	$255
UGT-25000 STIG. (steam injection)	40,100 kW	7990 Btu	42.7%	$8,200,000	$204
PG6581B	42,100 kW	10,640 Btu	32.1%	$10,740,000	$255
PG6591C	42,300 kW	9410 Btu	36.3%	$11,100,000	$262
LM6000PD	42,330 kW	8310 Btu	41.1%	$10,200,000	$241
LM6000PD(DLE)	42,400 kW	8200 Btu	41.6%	$10,700,000	$252
LM6000PC	43,400 kW	8115 Btu	42.0%	$9,835,000	$226
GTX100	45,000 kW	9215 Btu	37.0%	$11,300,000	$251
LM6000PC Sprint (dry)	46,780 kW	8095 Btu	42.2%	$10,900,000	$233
LM6000PC Sprint (water injection)	50,080 kW	8430 Btu	40.5%	$12,800,000	$252
W251B11/12	49,500 kW	10,450 Btu	32.6%	$10,400,000	$210
FT8 Twin	51,350 kW	8890 Btu	38.4%	$14,800,000	$288
Trent 50 DLE	51,500 kW	8105 Btu	42.1%	$15,100,000	$293
GT8C2	56,300 kW	10,600 Btu	32.2%	$12,900,000	$229
Trent 60 (water injection)	58,200 kW	8355 Btu	40.9%	$15,660,000	$270
V64.3	63,000 kW	9790 Btu	34.8%	$14,175,000	$225
V64.3A	67,700 kW	9730 Btu	35.1%	$14,800,000	$219
PG6101FA	70,200 kW	9980 Btu	34.2%	$14,700,000	$210
PG6111FA	75,900 kW	9760 Btu	35.0%	$15,900,000	$209
PG7121EA	85,100 kW	10,430 Btu	32.8%	$14,830,000	$174
GT11NM	87,900 kW	10,040 Btu	34.0%	$15,400,000	$174
LMS100 (water injection)	104,000 kW	7170 Btu	47.6%	$22,900,000	$220
LMS100 (steam injection)	112,150 kW	6850 Btu	49.8%	$24,900,000	$222
UGT-110000	114,500 kW	9480 Btu	35.0%	$14,000,000	$122
GT11N2	115,400 kW	10,150 Btu	33.6%	$18,900,000	$164
W501D5A	120,500 kW	9840 Btu	34.7%	$18,700,000	$155
PG9171E	126,100 kW	10,100 Btu	33.8%	$18,900,000	$150
Model	Base Load Rating	Heat Rate Btu/kWh	LHV Efficiency	Budget Plant Price	Price per kW
M701DA	144,100 kW	9810 Btu	34.8%	$22,300,000	$155
V94.2	163,300 kW	9905 Btu	34.4%	$24,500,000	$150
PG9231EC	169,200 kW	9770 Btu	34.9%	$26,700,000	$158
PG7241FA	171,700 kW	9360 Btu	36.5%	$28,500,000	$166
GT13E2	172,200 kW	9375 Btu	36.4%	$26,700,000	$155
V84.3A	180,000 kW	8980 Btu	38.0%	$30,700,000	$170
PG7251FB	184,400 kW	9215 Btu	37.0%	$29,400,000	$160
M501F	185,400 kW	9230 Btu	37.0%	$27,950,000	$151
GT24	187,700 kW	9250 Btu	36.9%	$34,700,000	$184
V94.2A	188,200 kW	9360 Btu	36.5%	$28,400,000	$151
W501FD2	198,300 kW	8985 Btu	38.0%	$28,900,000	$146
PG9331FA	243,000 kW	9360 Btu	36.4%	$35,960,000	$148
PG9351FA	255,600 kW	9250 Btu	36.9%	$38,900,000	$152
PG7001H..	260,000 kW	8640 Btu	39.5%	$41,000,000	$158
M501G	264,000 kW	8730 Btu	38.5%	$37,900,000	$143
W501G.	266,300 kW	8685 Btu	39.3%	$37,300,000	$140
PG9371FB	268,800 kW	9040 Btu	37.7%	$39,900,000	$148
M701F	270,300 kW	8930 Btu	38.2%	$43,200,000	$160
M701G	271,000 kW	8820 Btu	38.7%	$44,715,000	$165
V94.3A	272,400 kW	8745 Btu	39.0%	$40,000,000	$147
GT26	280,900 kW	8910 Btu	38.3%	$41,700,000	$148
M701G2	334,000 kW	8630 Btu	39.5%	$51,500,000	$154