بسم الله الرحمن الرحيم

Sudan University of Science and Technology

College of Graduate Studies

Optimize the Operational Parameters through Simulation to Increase the Efficiency of Combined Cycle Power Plants

أمثلة عوامل التشغيل باستخدام المحاكاة لرفع كفاءة دورة محطات القدرة المزدوجه

A THESIS SUBMITTED TO THE COLLEGE OF GRADUATE STUDIES, IN FULFILLMENT OF THE REQUIREMENT FOR THE DEGREE OF Ph.D. IN MECHANICAL ENGINEERING

By

Abdallah Mokhtar Mohammed Abdallah

(B.Sc. & M.Sc. Mechanical Engineer)

Supervised by: Dr. Mohammed Eltayeb Mansour

Co-Supervised by: Dr.Tawfig Ahmed Gamal Eldeen

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بِسْمِ اللهِ الرَّحْمَنِ الرَّحِيمِ

ورُ السَّمَاوَاتِ وَالأَرْضِ مَثَلُ نُورِهِ كَمِشْكَاةٍ فِيهَا
احٌ الْمِصْبَاحُ فِي زُجَاجَةٍ الزُّجَاجَةُ كَأَنَّهَا كَوْكَبٌ
يُوقَدُ مِن شَجَرَةٍ مُّبَارَكَةٍ زَيْتُونِةٍ لاَّ شَرْقِيَّةٍ وَلاَ
غَرْبِيَّةٍ يَكَادُ زَيْتُهَا يُضِيءُ وَلَوْ لَمْ تَمْسَسْهُ نَارٌ نُّورٌ عَلَى
ي اللهُ لِنُورِهِ مَن يَشَاءُ وَيَضْرِبُ اللهُ الأَمْثَالَ
لنَّاسِ وَاللهُ بِكُلِّ شَيْءٍ عَلِيمٌ }

صدق الله العظيم

سورة النور الاية(35)

DEDICATION

To My Mother and My Father

On Their Best Upbringing

To My Wife and My Children

On Their Forbearance

To All Mechanical Engineers

Acknowledgment

My gratitude goes to my supervisors, Dr. Mohamed Eltayeb Monsour and Dr.Tawfig Ahmed Jamal Eldeen, for their technical and moral support in the initiation and successful completion of the research. Their valuable criticism, fruitful discussion and continuous guidance were the source of great inspiration to me.

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Abstract

Electric power generation is one of the important factors for the development of peoples and can take an advantage of energy extraction technology of combined cycle in Sudan, which is highly effective in the States instead of remote power generation and combined cycle power plant also be economically feasible for use in sugar refineries in Sudan.

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The purpose of this study is to find the maximum efficiency and minimum cost of power generation in combined cycle power plant by simulate GARRI (1) combined cycle power plant by using ASPEN HYSYS. It aims to conduct a detailed thermodynamic analysis for combined cycle power plant and optimization to attained maximum efficiency by dissertating different scenarios of operating parameters.

The study examined the operational side by passing through all the components of the combined cycle power plant and the mechanism of the system. Block 1 in GARRI (1) combined cycle power plant is used.

The efficiency obtained from ASPEN HYSYS simulator is **31.89%** and that find by mat lab code is 31.42% while the actual of GARRI (1) is **27.4%.** The effect of each operating parameter on the efficiency and power output was extracted by using Microsoft excel in form of graphical charts resulted from the thermodynamic analysis done by using ASPEN HYSYS simulator. The maximum efficiency in the optimum operating parameters is about **33.88%** by using different scenarios. Reference to the results the Parameters that have a significant impact on the overall efficiency and the power output of the plant and it has and economic effect on the results are the fuel mass flow rate and compressor pressure ratio. The study verified and chooses the values to give optimum efficiency and power output under a minimum cost. Research found that air (ambient/outdoor) temperature has a major impact factor effect on the overall efficiency and net power output.

المستخلـص

توليد الطاقة الكهربائية من الأمور الهامة لتنمية الشعوب ويمكن الاستفادة من تكنولوجيا استخراج الطاقة من الدورة المزدوجة في السودان في الولايات بدلاً من توليد الطاقة بالمحطات الصغيرة حيث أن استخدام الدورة المزدوجة في مصانع السكر في السودان ذو جدوى اقتصادية.

الغرض من هذه الدراسة هو إيجاد أقصى قدر من الكفاءة و بأقل تكلفة لتوليد الطاقة في محطة تعمل بدورة مزدوجة.بمحاكاة محطة توليد الطاقة الكهربائية المزدوجة (قري1) باستخدام برنامج (ASPEN HYSYS) وأجراء تحليل حراري مفصل لمحطة التوليد المزدوجة وحساب أقصى قدرة بأعلى كفاءة ممكنة من خلال طرح سيناريوهات مختلفة من متغيرات التشغيل المختلفة.

شملت الدراسة الجانب النظري و مكونات محطة الطاقة المزدوجة وآلية النظام و ذلك لدراسة مربع 1 في محطة قري الحرارية لتوليد الكهرباء في السودان. تم حساب كفاءة التوليد باستخدام برنامج ASPEN HYSYS حيث اعطى 31.89% وايجاد قيمتها باستخدم ماتلاب 31.42%، بالمقارنة مع الكفاءة العملية لمحطة قري الحرارية والتي تساوي 27.4%

تم استخدام تأثيركل متغيرمن متغيرات التشغيل على الكفاءة والطاقة الناتجة للمحطة المصممة بمساعدة برنامج Microsoft excel في شكل مخططات بيانية عن طريق التحليل الحراري باستخدام برنامج ASPEN HYSYS للمحاكاة. تم حساب أقصى كفاءة في إطار معايير التشغيل المحددة بافتراض سيناريوهات مختلفة لمتغيرات التشغيل. وجد أن أقصى كفاءة للمحطة المصممة يمكن أن تصل إلى 33.88٪ تحت متغيرات التشغيل المثلى المحسوبة.باستخدام برنامج (ASPEN HYSYS) وإجراء تحليل حراري مفصل لمحطة التوليد المزدوجة وحساب أقصى قدرة بأعلى كفاءة ممكنة من خلال طرح سيناريوهات مختلفة من متغيرات التشغيل المختلفة. البحث اثبت أن درجة حرارة الهواء (الخارجي/المحيط) لهما تأثير ملحوظ على القدرة المنتجة و الكفاءة.

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List of Abbreviations & Symbols

AC	Absorption chiller
CC	Combined cycle
C.C	Combustion chamber
CCGT	Combined cycle gas turbine
CCPP	Combined cycle power plant
C_P	Specific heat
DIT	Dublin Institute of Technology
e	Base of the natural logarithm
FDOT	Finite dimensional optimization thermodynamics
FST	Finite speed thermodynamics
FTT	Finite time thermodynamics
GT	Gas turbine
HRSG	Heat recovery steam generator
H E	Heat Exchanger
HP	High pressure
h_1	Enthalpy at turbine inlet
h_2	Enthalpy at turbine outlet
h_3	Enthalpy at condense outlet
h_4	Enthalpy at boiler inlet
ISO	International standards organization
L_n	Natural log
LP	Low pressure
LPG	Liquid petroleum gas

P_{t1}	Pressure at compressor inlet
P_{t2}	Pressure at compressor outlet
Q_L	waste heat in the combined cycle
Q_{added}	heat added
Q_{in}	Heat input from the combustion in the actual cycle
q_{in}	Specific heat input
q_{added}	Specific added heat in Rankine cycle
$q_{rejected}$	Rejected heat in Rankine cycle
r_p	compression ratio
S_b	Entropy at H.E outlet to the C.C
S_a	Entropy at compressor outlet to the H.E
T_{ta}	Temperature at compressor outlet to the H.E
T_{tb}	Temperature at H.E outlet to the C.C
T_{t1}	Temperature at compressor inlet
T_{t2}	Temperature at compressor outlet
T_{t3}	Temperature at turbine inlet
T_{t4}	Temperature at turbine outlet
$T_{t2`}$	Temperature at compressor outlet (actual cycle)
$T_{t4`}$	Temperature at turbine outlet (actual cycle)
T_{t5}	Temperature at H.E outlet
TIT	Turbine inlet temperature
W_{56}	Work output of the second turbine in the reheating system
W_{4a}	Work output of the first turbine in the reheating system
w_{cycle}	Net work in Rankine cycle

W_{total}	Work of the combined cycle
W_{gt}	Work output of the gas turbine
W_{st}	Work output of the steam turbine
w_{out}	Work output
Wt	Turbine work
Wc	Compressor work
W_n	Net work

Greek Symbols

η_{th}	Thermal efficiency
η_t	Turbine efficiency
η_c	Compressor efficiency
$\eta_{th\ Rankine}$	Thermal efficiency of Rankine cycle
η_{is}	Isentropic efficiency
η_{gt}	Gas turbine efficiency
η_B	Boiler efficiency
η_{cc}	combined (steam & gas) efficiency
γ	Specific heat ratio
α	temperature ratio of gas turbine inlet to Ambient

CHAPTER I

INTRODUCTION

CHAPTER I

Introduction

1.1 Introduction

Probably a windmill was the first turbine to produce useful work where in there is no pre-compression and no combustion. The characteristic feature of gas turbine as we think of the name today includes a compression process and a heat addition. The gas turbine represents perhaps the most satisfactory way of producing large quantities of power. The thermal efficiency of a gas turbine alone is still quite modest compared with that of steam turbine that we can construct combined plants whose efficiency is of order of 45% [1] and more.

At the end of the last century a new era of electric power as a new type of stations that merged the two sessions of the gas and steam together in so-called combined cycle for higher efficiency and to improve the economic aspects.

The combined cycle has the advantage to use the temperature of the gases from gas turbine cycle to generate steam instead of exhausting gases directly into the atmosphere. More than half of the productive capacity in the cycle comes from the combined cycle so the gas cycle variables play a key role in the proper operation of combined cycle.

In this study, the focus will be on the gas cycle analysis to determine the importance of different variables in achieving better efficiency and give a greater amount of qualitative filled cycle simple in principle.

Successive innovations and ideas to improve the exploitation of energy wasted in the combustion gases expelled and originated the idea of courses dual containing gas turbines built or linked to turbine steam where is the benefit of gases emerging from the cycle gas turbine to turn water into steam and then introduced into the turbine steam, which is connected to a generator Electricity.

1.2 Problem Statement

- To determines the performance of combined cycle power plant and its components through the available methods
- Efforts needed to concentrate on increasing the efficiency of the power plants and the power output with cutting down on the fuel used.
- Gas cycle analysis to determine the importance variables to achieve better output power and efficiency.

1.3 Objectives

1.3.1 General

The purpose of this project is to optimized combined cycle in terms of operating performance to increase the power generated by the gas turbine and efficiency using the exhaust heat under the lowest cost

1.3.2 Secondary

- Identify the variables that affect the performance of the gas cycle and fine better initial values to them.
- Statement of the impact of the efficiency of the gas cycle in the overall efficiency of the combined cycle.
- Select a computer simulation of a combined cycle power plant for the thermodynamic and thermo-economic parameters detailed thermal analysis.
- Consider different scenarios for optimum combined cycle power plant design to find maximum efficiency and minimum cost and discuss factors affecting in combined cycle power plant performance

1.4 Scope of Work:

The problems that emerged during the study of this model of combined cycle plant are:-

- Difficulty of choosing the right program and understanding it for the simulation which contains the components of combined cycle power plant.

- Collecting data specially the design of heat recovery steam generator (HRSG) since in most of the visited sites some information’s and data where considered confidential.
- Time wasted in learning and familiarizing with new programs used in optimization stage.

1.5 Research Methodology:

The completion of this study will to gain results of the following results:

1. Selection of a computer program for combined cycle.
2. The work of the simulation program to identify the thermal and economic variables.
3. Create conditions that maximize the effectiveness of combined cycle power plants.
4. Finding maximum efficiency and minimum cost of power generation in the Plant.

The study is shown in the flow chart figure (1.1) & (appendix A)

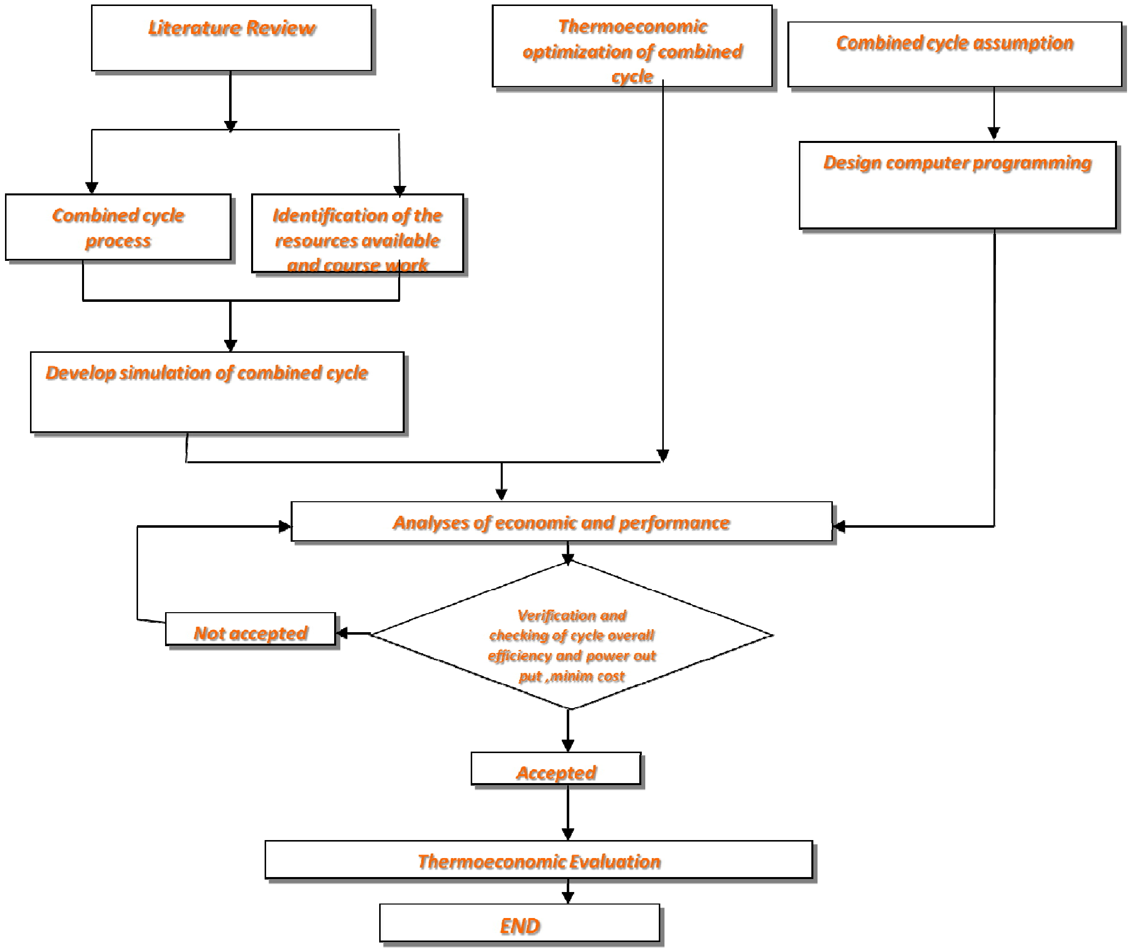


Figure (1.1) Activities of the study

CHAPTER II

LITERATURE REVIEW &
THEORETICAL BACKGROUND

CHAPTER II

Literature Review & Theoretical Background

2.1 Literature Review

Sanjay Kumar and Onkar Single 2013 [2] the study evaluate the performance of Gas-steam Combined cycle having Transpiration Cooled Gas Turbine. The computer code developed on the basis of modeling has been used to compare the transpiration cooled combined cycle performance for varying Turbine Inlet Temperature (TIT). Coolant requirements, steam generation rate in Heat Recovery Steam Generator (HRSG), heat recovery in HRSG, cycle efficiencies, cycle specific work output and specific fuel consumption of the combined cycle are compared for TIT at a fixed pressure ratio of 23:1. The study shows that the coolant required for cooling gas turbine blades increases with increasing TIT. For a fixed maximum steam generation temperature in HRSG the difference between the exhaust gas temperature and steam generation temperature increases with increasing TIT which augments the heat recovery in HRSG. For an increase of TIT from 1600 K to 1700 K the exhaust gas temperature increases from 846 K to 907 K and heat recovery in high pressure (hp) section increases by 6.4 percent. Due to increased heat recovery in HRSG the total steam generation rate increases with increasing TIT. All the above effects improve the efficiency and specific work of each of the topping, bottoming as well as combined cycle with increase of TIT. The study concluded that using transpiration cooling higher TIT's enhances the overall performance of combined cycle by improvement in gas cycle as well as in steam cycle by diminishing the temperature differences in the heat recovery steam generator and by lowering the stack temperature.

Jim McGovern et al 2010 [3] the study has focused on the theoretical development of a novel combined cycle power generation system using a Sterling Cycle engine as a bottoming cycle on a stationary Otto Cycle engine. Such power plant would find use in small to medium scale power generation scenarios.

The aim of the current work is to provide a details validation procedure for the theoretical model and to developed under the frameworks of what are traditionally called

Finite Time Thermodynamics (FTT) and Finite Speed Thermodynamics (FST). However, although the Finite Time Thermodynamics name is often used, it is apparent that the concept has broadened from the original study of the time parameter to include other constraints typical of such applications.

Finite Dimensional Optimization Thermodynamics (FDOT) include optimization procedures that might usually be ascribed to the literature of Finite Method offers the advantage of providing good qualitative and quantitative agreement with known operating characteristics of real engines using comparatively uncomplicated models.

Eleni T. Bonataki_ and K.C. Giannakoglou (2007) [4] the study is concerned with the techno economic optimization of investments in Combined Cycle Gas Turbine (CCGT) power plants. Three CCGT power plant design problems, with total power output in the range of [170, 200] MW, [360, 400] MW and [720, 800] MW, are analyzed and discussed. In each one of them, the gas turbine is selected from a list of models which are available in the market place and produce, used as a single or multiple units, 120MW, 260MW and 520MW, respectively. The constrained, multi objective optimization is carried out through evolutionary algorithms; the design space consists of continuous and integer variables, where the latter correspond to the selection of the Gas Turbine(GT) model from a database as well as the type of HRSG.

Boonnasaa S. et al. (2003) [5] the Study is to improve the capacity of the combined cycle (CC) power plant which has been operated for 8 years. The research purposes of a steam absorption chiller (AC) to cool intake air to the desired temperature level. It could increase the power output of a gas turbine by about 10.6% and the CC power plant by around 6.24% annually. With payback period about 3.81 years and internal rate of return 40%.

2.2 Theoretical Background

2.2.1 Preface

Considered the power plants operating gas turbine, a relatively recent and stings with different sizes and are usually used during peak load in countries where there is steam or water generation plants. Gas turbine is a cycle to generate power using air as fluid primary operating. The first use of the gas turbine was about the year 1930/19 century by

"George Brighton" in the frequency of engine invented. The rotary actuators are used now and not in reciprocating gas turbines. The first gas turbine used to generate electric power in Switzerland in 1939 and the thermal efficiency of the turbines used until 1954 for 17%[6] .Due to the low efficiency of the compressor and the turbine, as well as the lack of development of adequate in metallurgy for the production of metals bear high temperatures result in limiting use of gas turbines. The highest thermal efficiency of combined cycle obtained from Siemens design in the world in 2016 is 61 %[7]. The use of gas turbines for power generation throughout the day, include the peak period.

2.2.2Gas Turbine Classification

a) Open Cycle Gas Turbine

Gas turbine type in which the air passes through the compressor and to the combustion chamber where fuel is ignited. The hot gas resulting from the combustion enter the turbine and expands, driving the rotor of the turbine that convert the heat capacity to mechanical circulatory, and the exhaust gases rejected to air at the exit of the turbine.

b) Closed Cycle Gas Turbine

Gas turbine cycle in which the air passes through the compressor, to the combustion chamber .The heat exchanger raising the temperature of air under constant pressure and expanded within the turbine converting the heat capacity to mechanical circulatory. The exhaust gases at the exit of the turbine passed through the heat exchanger to the compressor.

2.2.3 Components of Gas Turbine Stations:

a) Air Compressor:

The compressor takes air from the surrounding atmosphere to produce high pressures required for improving the efficiency of the cycle, as well as the air pressure over the mass which increases the energy. This energy is used to drive the gas turbine and produce power. Compressors used are:-

I. **Centrifugal Compressor:**

It consists of centrifugal compressor fan, driving shaft and diffuser. Fan blades mounted on a radial spindle to form corridors spaced for acceleration air and the air is expelled in the radial direction under the influence of centrifugal force. The resulting kinetic energy converts to pressure energy in the diffuser [8]. The compression ratio is relatively low in comparison with the compressor central pivot and does not exceed 5% per single-stage which is not enough to get the required efficiency.

II. **Axial Compressor:**

Axial compressor consists of successive rows of fixed and moving blades to accelerate the deployment of the air. Blades of the compressor can be controlled to suit different loading conditions. Usually a multi-stage axial compressor is used. Diffuser which is found in the compressor is used to increase the pressure of the air and to reduce speed before entering the combustion chamber. Axial Compressor has a higher compression ratio which helps to get high efficiency and lead to less fuel consumption.

b) Combustion Chamber:

Combustion chamber is to mixed compressed air from the r compressor with fuel and burning together by means of a special reignite. Combustion products are introduced to the turbine at the proper temperature and pressure that does not exceed the permissible value at the entrance of the turbine. Combustion chamber can be designed in various forms as follows:-

i. Cylindrical (Cans):

It is a room in the form of cylinders to separate vehicle on the perimeter of the column connector between the compressor and the turbine in a circular motion to receive fuel, flame with equal combustion in cylinders (Cans). This type is used in aircraft engines and is suitable for engines that use centrifugal compressors system

ii. Annular (Annular):

Consists of one large room with several fuel nozzles placed on the perimeter of the room (figure (2.1)) and is called the Ring of Fire to provide fuel. Advantages are simple and its size is smaller than the cylindrical with a high efficiency. Due to the lack of movement of uneven air temperature distribution inside the cylinder.

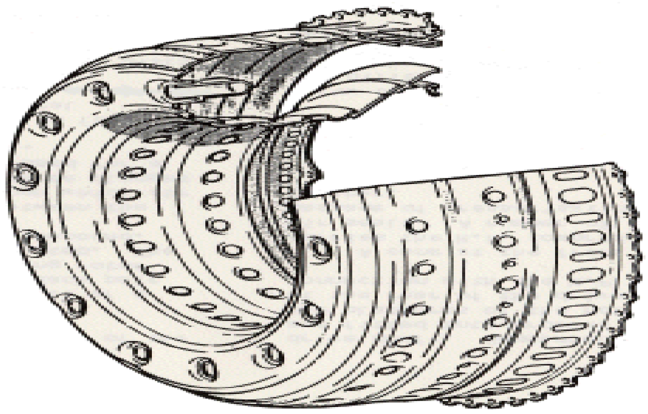


Figure (2.1) annular combustion Chamber

iii. Annular Cylindrical (Can Annular):

Is the integration of a number of cylinders in the annular cylindrical on the cover of the throat going through the air to be distributed to the cylinders through nozzles in the annular casing figure(2.2)[9]. The combustion is stable and hence the carbon deposition in the minimum level of safety with an allowable control of the combustion products within the limit for high temperatures that generate nitrogen oxides

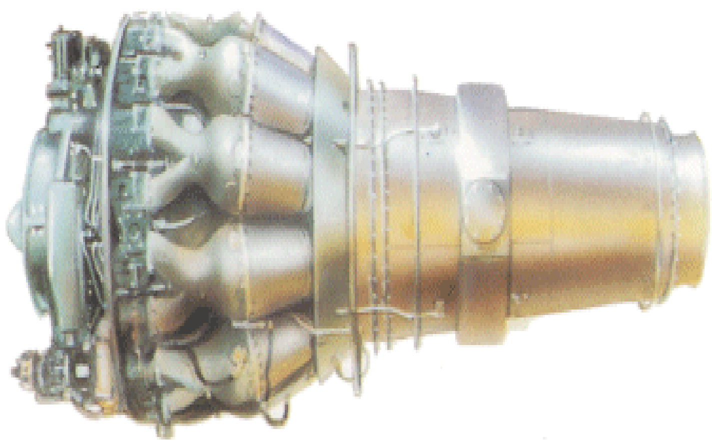


Figure (2.2) Annular cylindrical

c) Turbine:

Horizontal axis turbine tied by hand with the axis of the piston air directly and on the other hand with the generator, a gearbox to reduce the speed of rotation of the turbine to match the speed of rotation of the electrical generator. Interference gases resulting from the combustion turbine drives many feathers on the number of low-pressure (diameter turbine divergent of these side) into the air through the chimney.

Turbine extracted energy from the combustion products and converted into rotational mechanical energy for operation of the compressor and other accessories which ensures the continued rotation of the motor.

d) Electric Generator:

The electrical generator connected to the turbine by a gearbox to reduce the speed.

e) Machinery and Auxiliary Equipment:

Gas turbine plants need some assistance, equipment and machinery as follows:
1. filters the air before it enters the air compressor.
2. Assistant initial operation either a diesel engine or an electric motor.
3. Machines cooling water cooling station.
4. Equipment to measure temperature and pressure in each stage of the work.
5. Equipment for electrical measurement.

2.2.4 Advantages & Disadvantages:

a) Advantages:

1. The size and weight of the gas turbine is less compared to the steam turbine, but the largest compared with diesel engines with high speeds.
2. 2Cylindrical shape of the compressor and the turbine, make it easy to install, operate and reliable.
3. Gas turbines do not need to run in the water and need less amount of lubricating oils.
4. The rate of vibration is less compared with reciprocating engines.
5. Quick response operation compared to the steam turbine.
6. Thermal efficiency of gas turbine nearly identical to the efficiency of the steam turbine of the same size and less than diesel engines
7. Prices lower than other private power generators.

8. More flexibility in the type of fuel used compare to the Ferns 'Diesel' natural gas or coal.

b) Disadvantages:

1. Maximum temperature limited due to metal turbine.
2. Thermal efficiency is lower compared to other power plants.
3. They are not self starting
4. Need special method of cooling

2.2.5 Classification of Gas Turbines:

a) According to Usage

1. Used in Aviation
2. Used in trains.
3. Used in Power Generation.
4. Used in Marine Propulsion
5. Used in oil and Gas Industry.

b) According to Turbine Design:

1. Simple
2. One axis
3. Multiple axes
4. Cooling cycle
5. Reheating

c) According to the Used Fuel Type

1. Liquid fuel
2. Solid fuel
3. Gaseous fuel

2.2.6 Uses of Gas Turbines to Generate Electric Power:

Gas turbines used are simple to produce electrical energy in developing countries as a backup or when the peak times. It is expected to become the head of the unit in power generation in the future at all times. Also be used with steam turbines as a machine to generate power, hot exhaust released from the gas turbine is used to heat the feed water in the steam generator.

2.2.7 Types of Gas Turbines Used In Power Plants:

a) Open Gas Turbine Electric Generator:

The first turbine is driving the compressor and the other turbine exclusively driving the generator. In some cases the exhaust gases of the second gas turbine used to drive a steam

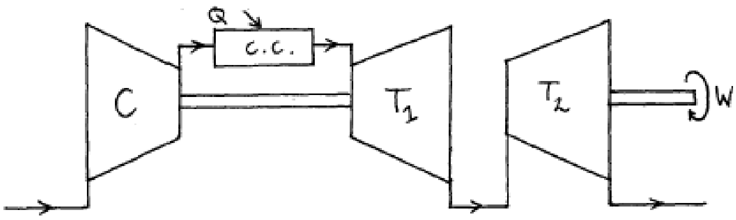


Figure (2.3) Open gas turbine electric generator

turbine in the combined cycle.

b) Open Cycle Gas Turbine With Heat Exchanger:

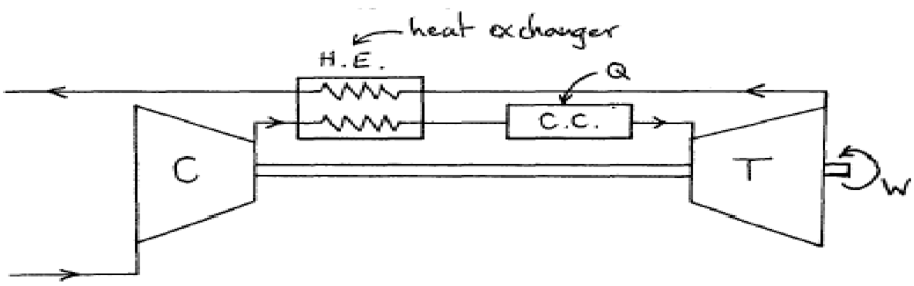


Figure (2.4) Open cycle gas turbine with heat exchanger

c) Open Cycle Gas Turbines With Reheat:

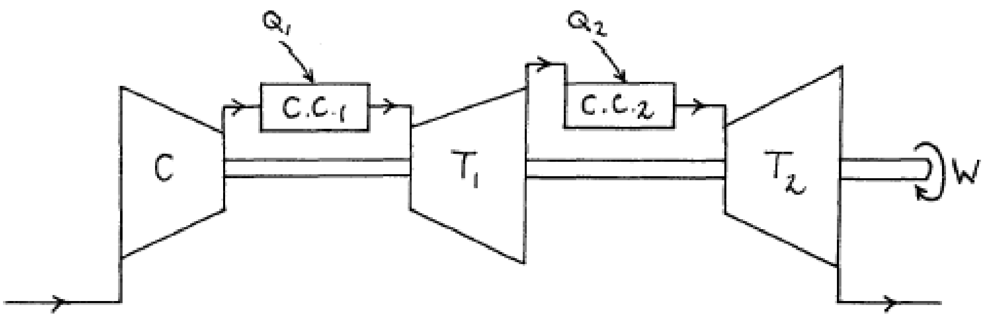


Figure (2.5) Gas turbines open cycle with reheat

It can burn a large amount of fuel without exceeding the temperature of the turbine generator using the re-heating.

c) High and Low Pressure Turbines:

Compressors are axial compound with the corresponding turbine. Each one of them has different speed according to the design. The working fluid from compressor is heated by external source at constant pressure. Air of high pressure and temperature from the

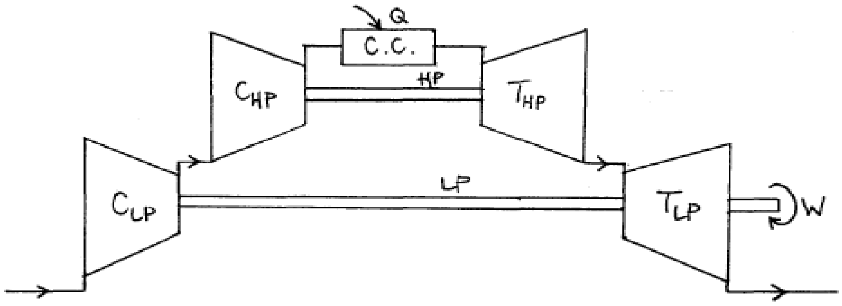


Figure (2.6) High and low pressure turbines

external heater is passed through the gas turbine. The fluid from the turbine is cooled to its original temperature in the cooler using external cooling source.

2.2.8 Analysis of Simple Cycle:

Model for gas turbines with constant pressure cycle:

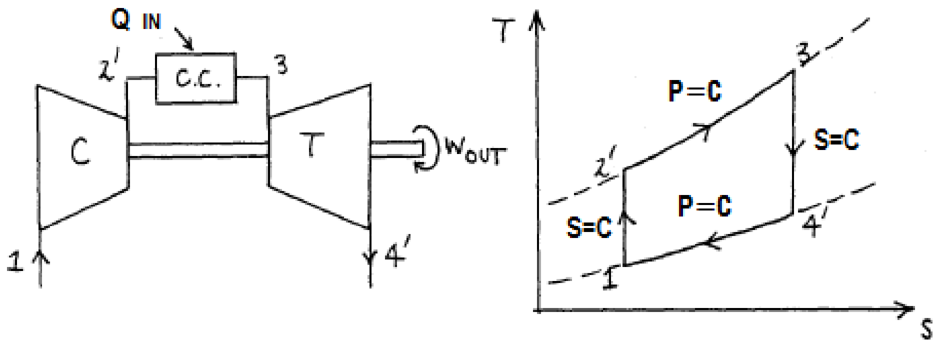


Figure (2.7) Analysis of simple cycle

Assumptions:

1) Isentropic and expansion compression cycle
2) No loss in pressure
3) The specific heat (C_P) is constant.
4) Air is a working fluid.

a) Thermal Efficiency:

$$\eta_{th} = \frac{w_{out}}{q_{in}} = \frac{(w_t - w_c)}{q_{in}} \qquad (2.1)$$

Where:

W_t =turbine work (kj/kg)

W_C =compressor work(kj/kg)

q_{in} =Heat input (kj/kg)

since all of the processes of expansion and compression isentropic then energy equation.

$$W_c = \Delta h_{tc} = c_p (T_{t2} - T_{t1}) \qquad (2.2)$$

$$W_t = \Delta h_{tt} = c_p (T_{t3} - T_{t4}) \qquad (2.3)$$

No work while adding heat therefore:

$$q = \Delta h_{tcc} = c_p (T_{t3} - T_{t2}) \qquad (2.4)$$

$$\eta_{th} = \frac{c_p (T_{t3} - T_{t4'}) - c_p (T_{t2'} - T_{t1}}{c_p (T_{t3} - T_{t2'})} \qquad (2.5)$$

Define the compression ratio:

$$r_p = P_{t2} / P_{t1} \qquad (2.6)$$

Hence: $$\frac{Tt2}{Tt1} = rp^{\frac{\gamma - 1}{\gamma}} = \frac{Tt3}{Tt4} \qquad (2.7)$$

$$\eta_{th} = \frac{\left[T_{t3}(1 - r_p^{\frac{1-\gamma}{\gamma}}) - T_{t2'}\left(1 - r_p^{\frac{1-\gamma}{\gamma}}\right)\right]}{T_{t3} - T_{t2'}} \qquad (2.8)$$

$$\eta_{th} = 1 - r_p^{\frac{1-\gamma}{\gamma}} \qquad (2.9)$$

:. The thermal efficiency of this cycle depends only on the ratio of compressibility and specific heat ratio γ .

b) Work Output:

$$W = Wt - Wc \qquad (2.10)$$

$$W = C_p T_{t3}\left(1 - \frac{T_{t4}}{T_{t3}}\right) - C_p T_{t1}\left(\frac{T_{t2}}{T_{t1}} - 1\right) \qquad (2.11)$$

$$W = C_p T_{t3}\left(1 - r_p^{\frac{1-\gamma}{\gamma}}\right) - C_p T_{t1}\left(r_p^{\frac{\gamma-1}{\gamma}} - 1\right) \qquad (2.12)$$

Where: $\frac{T_{t2}}{T_{t1}} = r_p^{\frac{1-\gamma}{\gamma}} = \frac{T_{t3}}{T_{t4}}$ (2.13)

$$\chi = r_p^{\frac{1-\gamma}{\gamma}} \qquad (2,14)$$

The total actual work out put:

$$W = C_p T_{t3}\left(1 - \frac{1}{\chi}\right) - C_p T_{t1}(\chi - 1) \qquad (2.15)$$

The maximum value for the work done is obtained when:

$$\frac{\partial w}{\partial \chi} = \frac{C_p T_{t3}}{\chi^2} - C_p T_{t1} \qquad (2.16)$$

Put: $\chi^2 = \frac{T_{t3}}{T_{t1}}$ (2.17)

Given that: $r_p^{\frac{\gamma-1}{\gamma}} = \frac{T_{t2}}{T_{t1}} = \frac{T_{t3}}{T_{t4}}$ (2.18)

$$\frac{T_{t3}}{T_{t4}} = \underbrace{\frac{T_{t2'}}{T_{t4}}}_{\chi} = \underbrace{\frac{T_{t3}}{T_{t4'}}}_{\chi} \quad (2.19)$$

$$T_{t2'} = T_{t4'} \quad (2.20)$$

$$r_p = \left(\frac{T_{t3}}{T_{t1}}\right)^{\frac{\gamma}{2(\gamma-1)}} \quad (2.21)$$

T-S requirement of the diagram that achieves $T_2 = T_4$ to give maximum work net decides that the form of the cycle.

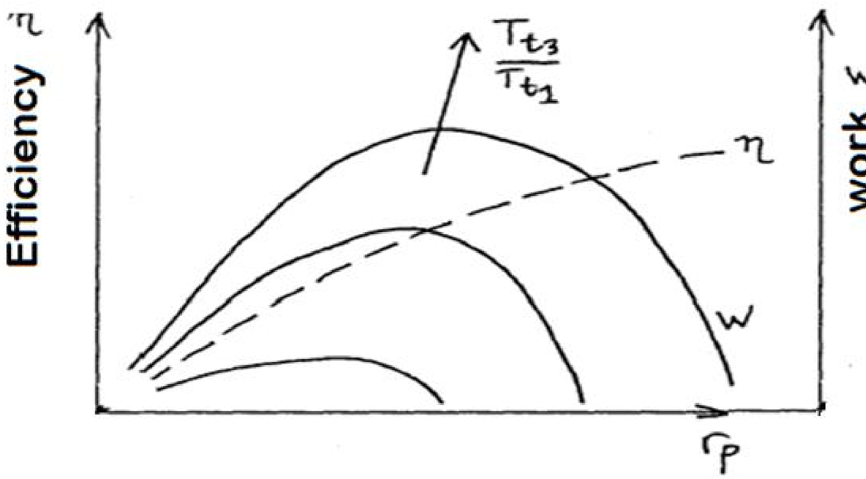


Figure (2.8) Efficiency& Work out put with pressure ratio

c) How the gas turbine cycle operates as thermal engine:

$$S_b - S_a = C_p Ln\left(\frac{Ttb}{Tta}\right) - R\, Ln\left(\frac{p_{tb}}{p_{ta}}\right) \quad (2.22)$$

Along the line of constant pressure

$$S_b - S_a = C_p L_n\left(\frac{T_{tb}}{T_{ta}}\right) \Rightarrow \left(\frac{T_{tb}}{T_{ta}}\right) = \ell^{\left(sb-sa/Cp\right)}$$ (2.23)

Thus lines of constant pressure curves exponentially in the diagram of (T-S), Figure (2.6) these curves diverge increase entropy.

Also:

$$\frac{Tt4'}{Tt1} = \ell^{(s4'-s1)/cp} = \ell^{(s3-s2)/cp} = \frac{Tt3}{Tt2'} \Rightarrow \frac{Tt4'}{Tt3} = \frac{Tt1}{Tt2'}$$ (2.24)

Give wok of the turbine:

$$W_t = C_p(T_{t3} - T_{t4'}) = C_p T_{t4'}\left(\frac{T_{t3}}{T_{t4'}} - 1\right) = C_p T_{t4'}\left(\frac{T_{t2'}}{T_{t1}} - 1\right) = \underbrace{\frac{T_{t4'}}{T_{t1}}}_{\succ 1}\underbrace{C_p(T_{t2'} - T_{t1})}_{w_c}$$ (2.25)

$$W_n = W_t - W_c \quad \succ 0$$ (2.26)

Thus cycle (gas turbine) thermal engine completely based on the deviation lines of constant pressure in the diagram of (T – S), Figure (2.6)

d) Isentropic Efficiency

Isentropic efficiency of the compressor and turbine has the actual and the ideal work for a specific compression ratio.

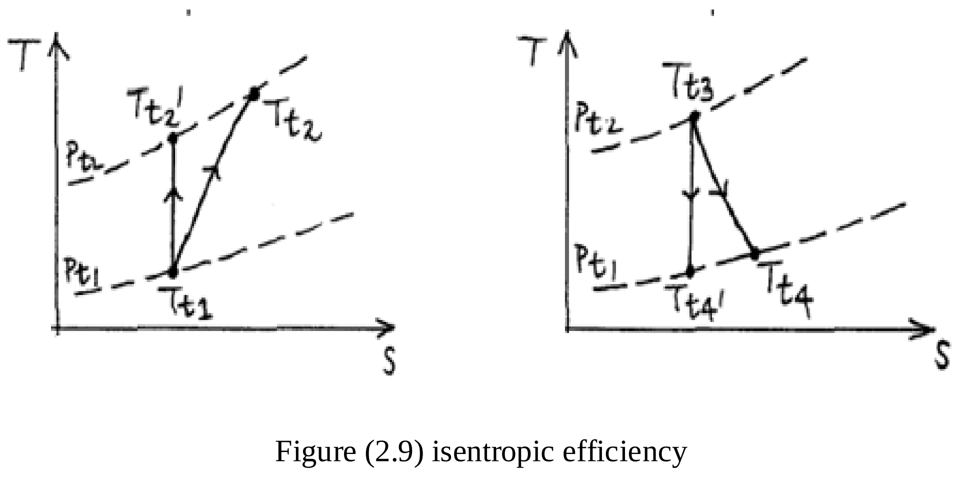


Figure (2.9) isentropic efficiency

Observed in the process of compression that the actual work done the highest ideal work.
Know the isentropic efficiency of the compressor
isentropic efficiency = actual work / Ideal work

$$\eta_{is} = \frac{C_p(T_{t2'} - T_{t1})}{C_p(T_{t2} - T_{t1})} = \frac{T_{t2'} - T_{t1}}{T_{t2} - T_{t1}} \qquad (2.27)$$

e) Non-Ideal Cycle (Actual):

The cycle includes non-ideal work for real compressor and turbine assumes a lack loss in spite of the pressure with the least pressure due to combustion is estimated at 3%.

Using the energy equation:

$$W = C_p(T_3 - T_4) - C_p(T_2 - T_1) \qquad (2.28)$$

W = work out put
Heat input from the combustion:

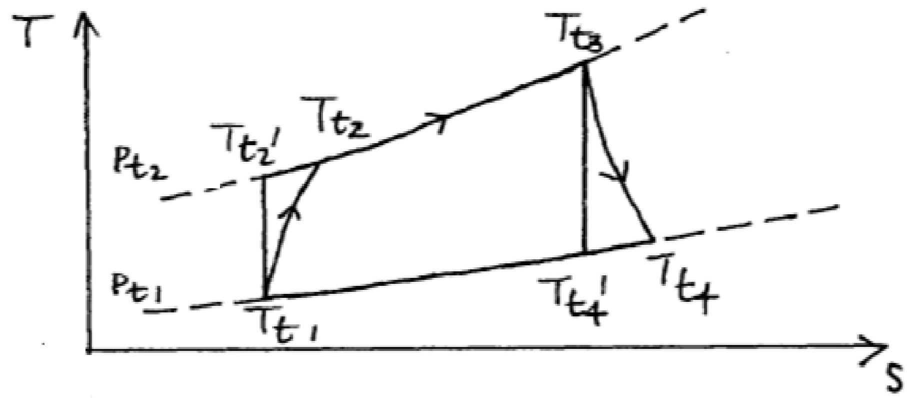


Fig (2.10) non-ideal Cycle (actual)

$$Q_{in} = C_p(T_2 - T_1) \qquad (2.29)$$

f) Thermal Efficiency for Non-Ideal Cycle (Actual):

Thermal efficiency is a work done to the total heat input:

$$\eta_{th} = \frac{W}{Qin} = \frac{Cp\,(T3 - T4) - Cp\,(T2 - T1)}{Cp\,(T3 - T2)} \qquad (2.30)$$

Now enter the actual values of temperature, point 2 and point 4:

$$\eta_{th} = \frac{\eta_t (T_{t3} - T_{t4'}) - \frac{1}{\eta_c}(T_{t2'} - T_{t1})}{T_{t3} - T_{t2}} \qquad (2.31)$$

For $\alpha = T_3 / T_1$ (2.32)

$$\eta_{th} = \frac{\eta_t \alpha \left(1 - r_p^{\frac{1-\gamma}{\gamma}}\right) - \frac{1}{\eta_c}\left(r_p^{\frac{\gamma-1}{\gamma}} - 1\right)}{\alpha - \frac{T_{t2}}{T_{t1}}} \qquad (2.33)$$

$$\frac{T_{t2}}{T_{t1}} = \frac{T_{t2} - T_{t1}}{T_{t1}} + 1 = \frac{1}{\eta_c}\frac{T_{t2'} - T_{t1}}{T_{t1}} + 1 = \frac{1}{\eta_c}\left(r_p^{\frac{\gamma-1}{\gamma}} - 1\right) + 1 \qquad (2.34)$$

$$\eta_{th} = \frac{\eta_t \alpha \left(1 - r_p^{\frac{1-\gamma}{\gamma}}\right) - \frac{1}{\eta_c}\left(r_p^{\frac{\gamma-1}{\gamma}} - 1\right)}{\alpha - \frac{1}{\eta_c}\left(r_p^{\frac{\gamma-1}{\gamma}} - 1\right) - 1} \qquad (2.35)$$

After re-arrangements:

$$\eta_{th} = \frac{\left(\eta_c \eta_t \alpha r_p^{\frac{1-\gamma}{\gamma}} - 1\right)}{\frac{\eta_c(\alpha - 1)}{\left(r_p^{\frac{\gamma-1}{\gamma}} - 1\right)} - 1} \qquad (2.36)$$

1 - Comparing this result with those in the ideal cycle the temperature exceeds the limits of maximum efficiency is dependent on metal turbine.

2 - Loss of compression $\eta_c = \prec 1$ offset partially by reducing Q_{in}. 3 - Loss in the turbine $\eta_t = \prec 1$ is losing from the system and it appears in the numerator once.

g) Specific Power:

Is the ratio of output power per unit mass flow through the gas turbine.
Of the energy equation

$$W_n = W_t - W_c = C_p(T_3 - T_4) - C_p(T_2 - T_1) \tag{2.37}$$

$$= C_p T_{t3}\left(1 - \frac{T_{t4}}{T_{t3}}\right) - C_p T_{t1}\left(\frac{T_{t2}}{T_{t1}} - 1\right) \tag{2.38}$$

Include

$\eta_t, \eta_t \, and \, r_p$:

$$W_n = C_p T_{t3}\eta_t\left(1 - rp^{\frac{1-\gamma}{\gamma}}\right) - C_p\frac{T_{t1}}{\eta_c}\left(rp^{\frac{\gamma-1}{\gamma}} - 1\right) \tag{2.39}$$

Equation can be arranged to form dimensionless the following:

$$\frac{W_n}{C_p T_{t1}} = \left(rp^{\frac{1-\gamma}{\gamma}} - 1\right) - \left(\frac{\eta_t \alpha}{rp^{\frac{\gamma-1}{\gamma}}} - \frac{1}{\eta_c}\right) \tag{2.40}$$

Diagram Fig (2.11) for the derived relationships for efficiency η_{th} and work W for the cycle against the actual:

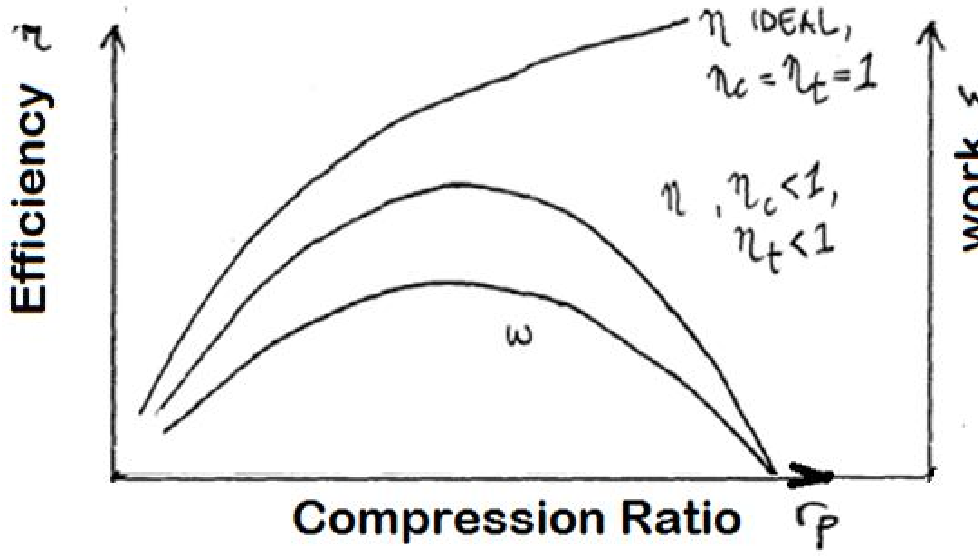


Figure (2.11) Specific power

1 - Increase the isentropic efficiency of the compressor and turbine give a small increase in thermal efficiency and a significant increase in work out put.

2 - Increase T_3 means a very large increase in work and some increase in thermal efficiency.

3 - Increase the isentropic efficiency of the turbine has a great effect on the thermal efficiency of the increased isentropic efficiency of the compressor.

h) Other Cycles:

It is possible, in the gas turbine cycle, to indicate that the thermal efficiency and the work in Compression ratio and isentropic efficiency and temperature entering the turbine.

1. Compression ratio and inlet temperature of the turbine to determine the turbine materials.

2. Isentropic efficiency (for the compressor and turbine) can impose its values (80-85)% and expect a substantial increase in the future.

i) Reheating System:

Filled the turbine can be increased by adding heater and another turbine.

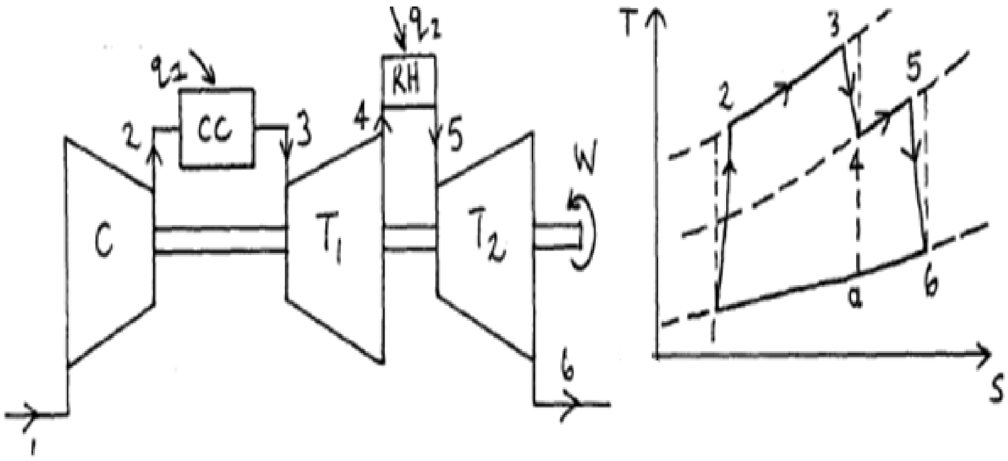


Figure (2.12) The Reheating system

$$W_t = C_p[(T_{t3} - T_{t4}) + (T_{t5} - T_{t6})] \& W_{56} \succ W_{4a} \quad (2.41)$$

$$q = C_p[(T_{t3} - T_{t2}) + (T_{t5} - T_{t4})] \quad (2.42)$$

The effect of re-heating:

1. Increase the work of the turbine and thus the output power.

2. Thermal efficiency and decreasing because $\eta_{th} = w/q$ the increase in w relatively more than increases the Qin.

j) Compressor Re-Cooling System:

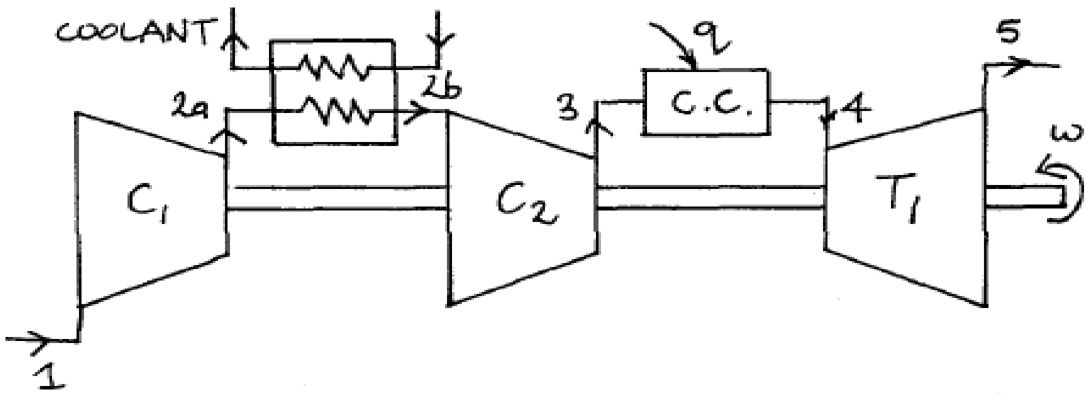


Figure (2.13) Compressor re-cooling system

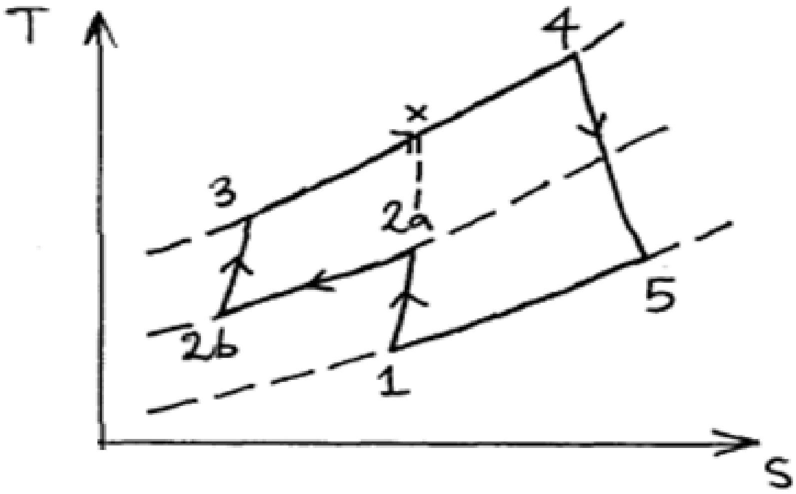


Figure (2.14) Compressor re-cooling system T-S Diagram

$$W_c = C_p[(T_{t2a} - T_{t1}) + (T_{t3} - T_{t2b})] \,\&\, W_c \prec C_p(T_{t\chi} - T_{t1}) \tag{2.43}$$

$$q = Cp(Tt4 - Tt3) \succ Cp(Tt4 - Tt\chi) \tag{2.44}$$

Effect of re-cooling:

1. Power output largest emerging because the compressor is less work.

2. Thermal efficiency is reduced because the relative increase in the work more than a modification in the excess heat added.

k) Regenerator System:

The heat exchanger is used to convert the heat emerging from the turbine to the fluid after leaving the compressor.

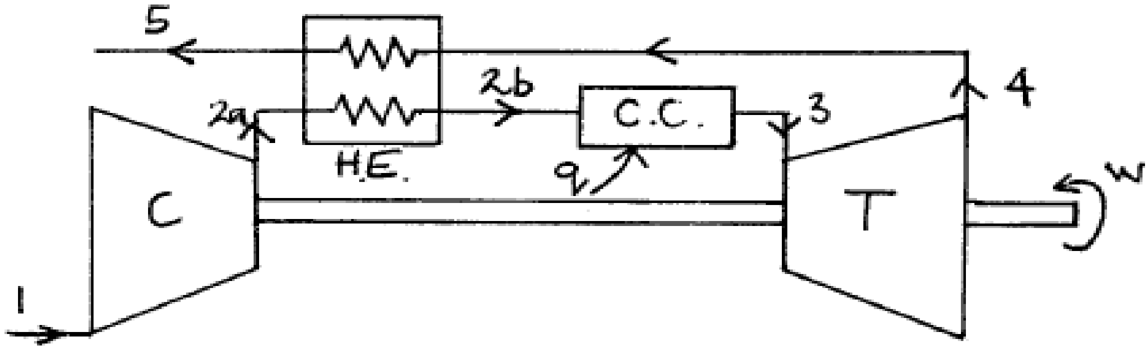


Figure (2.15) Regenerator system

The regeneration cycle is used when operating the gas turbine individually. In this case it uses the temperature of the exhausts gases out.

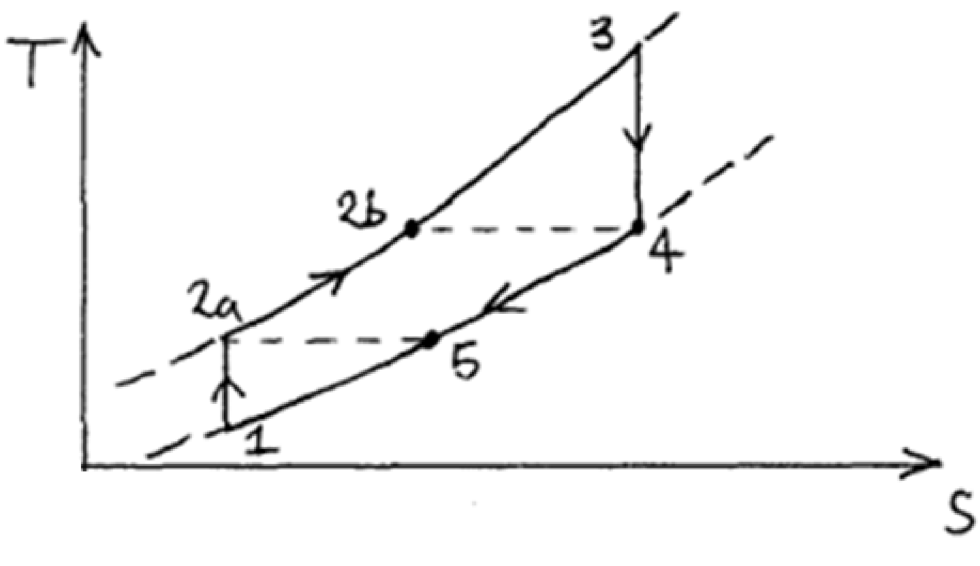


Figure (2.16) Regenerator system T-s Diagram

The efficiency of the heat exchanger:

$$\varepsilon=\frac{T_{2b}-T_{2a}}{T_4-T_{2a}}\approx\frac{T_{t2b}-T_{t2a}}{T_{t4}-T_{t2a}} \tag{2.45}$$

The low Mach number in the heat exchanger between turbine and compressor the process assumed ideal.

Total work:

$$W_n=W_t-W_c=C_p(T_{t3}-T_{t4})-C_p(T_{t2a}-T_{t1}) \tag{2.46}$$

The heat added: $Q_{added}=C_p(T_{t3}-T_{t2b})$ (2.47)

$$W_t=C_p(T_{t3}-T_{t4}) \tag{2.48}$$

Thermal efficiency:

$$\eta_{th}=\frac{W_t-W_c}{Q}=1-\frac{W_c}{Q}=1-\frac{T_{t2a}-T_{t1}}{T_{t3}-T_{t4}} \tag{2.49}$$

$$\eta_{th}=1-\frac{T_{t1}}{T_{t3}}\frac{\left(\frac{T_{t2a}}{T_{t1}}-1\right)}{\left(1-\frac{T_{t4}}{T_{t3}}\right)}=1-\frac{T_{t1}}{T_{t3}}\left(\frac{r_p^{\frac{\gamma-1}{\gamma}}-1}{1-r_p^{1-\gamma/\gamma}}\right)=1-\frac{T_{t1}}{T_{t3}}r_p^{\frac{\gamma-1}{\gamma}} \tag{2.50}$$

For gas turbines without regeneration $\eta_{th}=1-r_p^{\frac{1-\gamma}{\gamma}}$ (2.51)

Note:

1. Regeneration cycle can take a higher thermal efficiency than the basic cycle.

2. Improve the thermal efficiency.

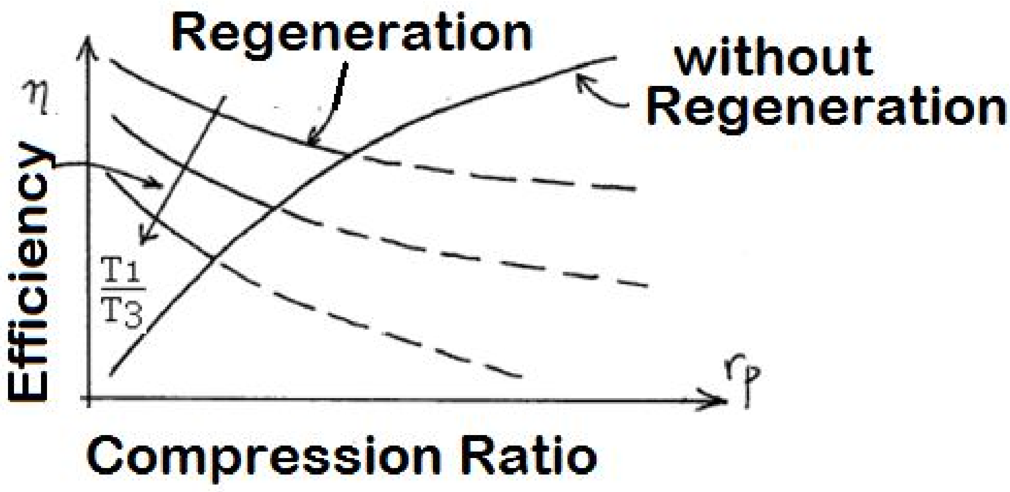


Fig (2.17) Efficiency & pressure ratio

2.3 Steam Cycles

2.3.1 Preface

In the steam cycle the water is heated in the boiler, the vapor with high temperature and pressure passes on the blades of the turbine, to the condenser and condensate into water that pumped back to the boiler (Figure (2.18))

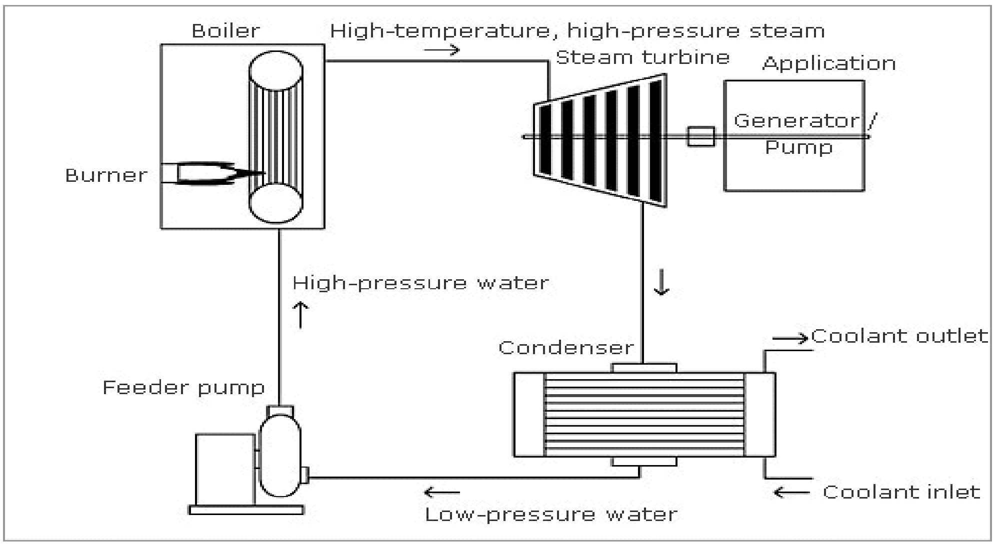


Figure (2.18) steam power plant component

2.3.2 Rankine Cycle Analysis

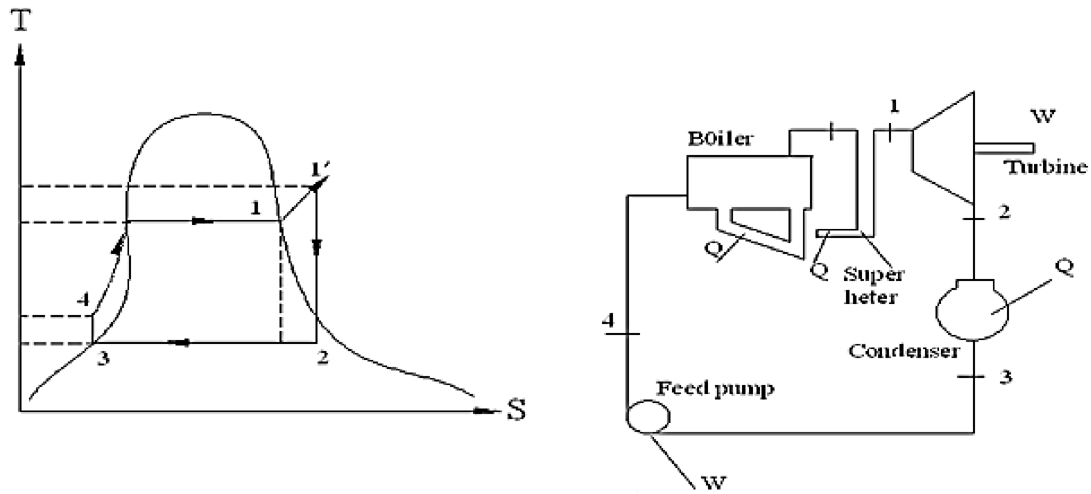


Figure (2.19) Rankine cycle

$$w_{cycle} = h_1 - h_2 (KJ/Kg) \qquad (2.52)$$

$$q_{added} = h_1 - h_4 (KJ/Kg) = h_1 - h_3 (KJ/Kg) \qquad (2.53)$$

$$q_{rejected} = h_2 - h_3 (KJ/Kg) \qquad (2.54)$$

$$\eta_{thRnkin} = w_{cycle} / q_{added} = (h_1 - h_2)/(h_1 - h_3) \qquad (2.55)$$

a) Effect of Pressure & Temperature on Rankine Cycle Efficiency

The efficiency of Rankine cycle can improve by:

1. Reducing the pressure result in condense steam and thus lowering its temperature.
2. Raising the temperature at which the evaporation result in raising the pressure to generates steam.

d) Rankine Cycle Reheat:

When steam, leaves the turbine, it is typically wet. The presence of water causes erosion of the turbine blades. To prevent this, steam is extracted from high pressure turbine and then it is reheated in the boiler and sent back to the low pressure turbine

e) Rankine Cycle Regeneration:

Regeneration helps improve the Rankine cycle efficiency by preheating the feed water into the boiler. Regeneration can be achieved by open feed water heaters or closed feed water heaters. In open feed water heaters, a fraction of the steam bled steam exiting a high pressure turbine is mixed with the feed water at the same pressure. In closed system, the steam bled from the turbine is not directly mixed with the feed water, and therefore, the two streams can be different pressures.

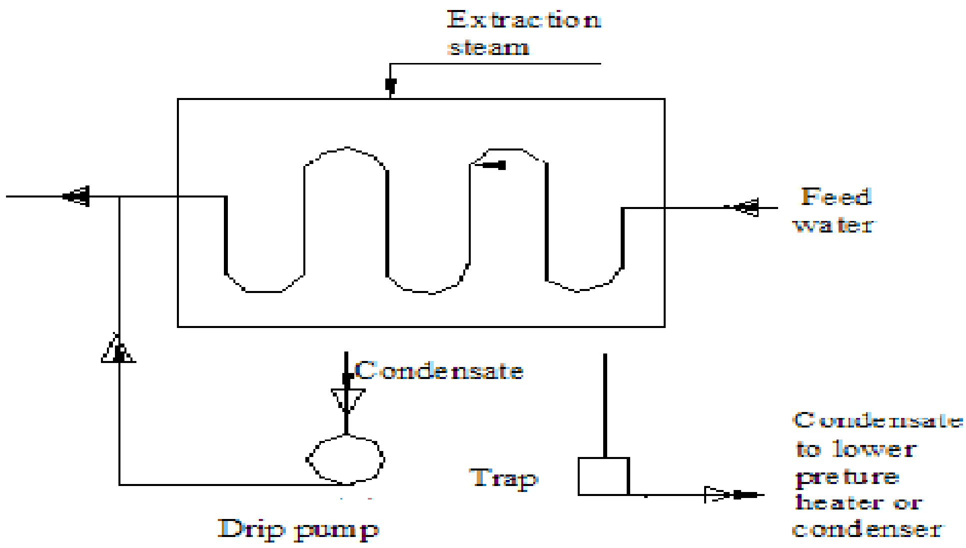


Figure (2.20) Regeneration closed feed water

Arrangement of the heaters in the power plant can be in multiple ways differ in detail, among them the following arrangements:

- Open heaters.

-Heaters unlocked compresses the steam heating intensifies after the feeding tube .

- Heaters unlocked with traps permit the exit of condensed water heated to the next.

Efficiency increases with increase of the number of thermal heaters but the costs of plant increases due to the need to feed pump with each heater. So it is when there is need to a large number of heaters It is recommended not to use open heaters and closed heaters used.

d) Ideal Fluid for Steam Cycle

The fluid used is water since it cheap, available and persistence of chemically and despite the fact that the properties of the water far from the properties of ideal fluid for steam cycles. The following is a summary of the characteristics desired in the ideal fluid as in the (T – S), diagram, Figure (2.21) for such a fluid

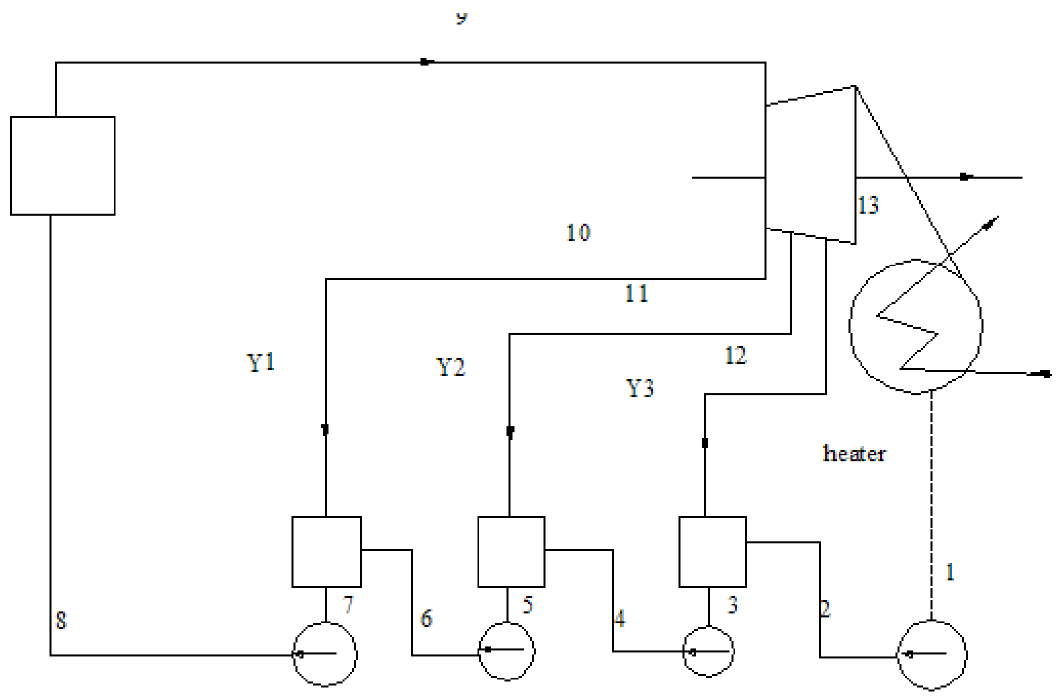


Figure (2.21): Steam power with three open heaters

1. The critical temperature of the fluid is clearly higher when the maximum temperature permitted for use for metal facility. In this case, there is no reason to roasting steam as it can add most of the heat at the highest temperature. The saturation pressure corresponding to the maximum temperature allowed being preferable to use the average pressure (20 to 50 bar) in order to reduce construction and maintenance costs.

2. For small specific heat of the liquid, any tendency of liquid saturation curve is great and so is the amount of heat needed to reach the boiling temperature is small and thus increasing the amount of heat added at the maximum temperature.

3. The latent heat of evaporation, for high consumption of the two types of low fluid leads to the small size of facility to a certain capacity. As well as the size of the business is getting smaller as increased density of the fluid.

4. Vapor saturation curve leads to lack of much lower degree of evaporation (0.9 <x) after expansion without having to use the re-heating.

5. The saturation pressure at the minimum temperature is higher than atmospheric pressure. Thus eliminating the need to use intensive, that prevents air leakage into the group there is no means of combining all of these properties that will be required and at the same time cheap fixed installation is non-toxic and does not cause corrosion of metals.

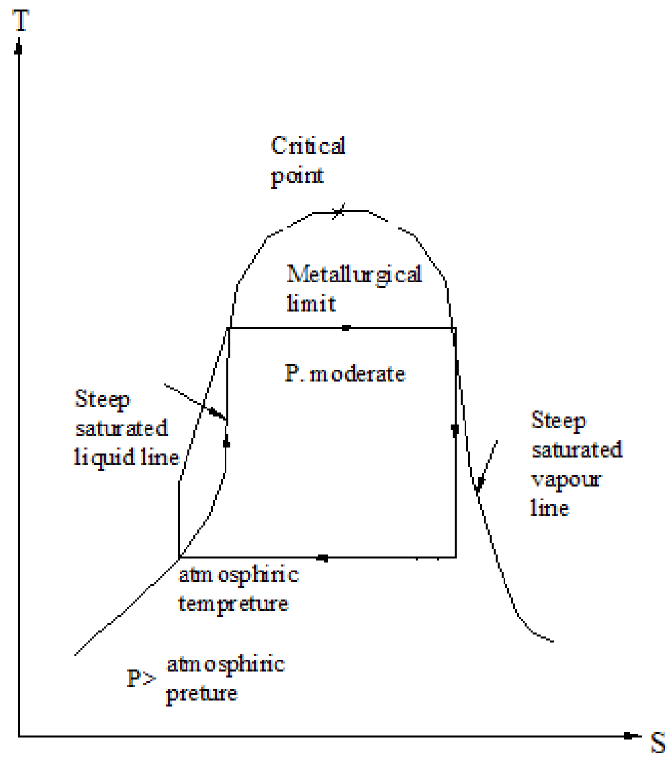


Figure (2.22): T-S diagram Ideal fluid for steam cycle

2.4 Combined Cycle

2.4.1 Preface

Combined cycle used heat rejected from a cycle (called the upper cycle) as input heat or supply for a second cycle with a lower temperature (called the lower cycle). Mercury cycles and steam combined established in the forties of the last century, the early cycle are combined, but not proved its trustworthiness. The term is usually used (combined cycle) currently to mean gas cycle combined with steam cycle Figure(2.23).

It has been a cycle of mercury and water combined-established in the forties of the last century, it was one of the first combined cycles, but it proved its trustworthiness for the following reasons:

1. Lack of mercury in the process of life and high cost.

2. The gas produced from mercury are toxic.

3. Mercury interacts with all the material.

2.4.2 Combined Cycle Power Plant Advantages:

(i) Energy generation is clean i.e., it's the most acceptable technology from an ecological standpoint.

(ii) High efficiency factor.

(iii) Minimal land requirement

(iv) Minimal water requirements.

(v) Fast operations. The station starts and shuts downs quickly, so it is possible to operate the facility both for base and peak load.

(vi) Construction time is short; accordingly, less time is required to repay the investment.

(vii) High level of automation and smaller number of staff required.

(viii) A wide range of fuels can be used, including natural gas, diesel oil, fuel oil and crude oil (flexibility in the use of different types of fuel)

(ix) Exploit the advantages of cycles (steam and gas).

(x) Combined cycle acceptable because a raid emerging from the gas turbine high temperature enables the use of a simple and effective. Also assured because the fluid operating smoothly and at a reasonable cost and are not toxic.

2.4.3 Types of Combined Cycle:

a) Combined Cycle Without Additional Ignition:

This type of combined cycle is the most prevalent and is the basis of most dual-energy plants. In this kind the gas turbine exhaust are fed directly to the heat exchanger (Heat Exchanger) where the heat transmitted to the steam. This session performance depends largely on the on the heat exchanger, which is often called the term boiler (Heat Recovery Steam Generator) (HRSG) or utter heater (Boiler). Cool the combustion gases pass through the heat exchanger because they lose heat to the water inside the boiler which is heated until it becomes saturated with water and boil until it becomes saturated steam then becomes superheated steam (Superheated steam / steam gas).

Cycle of the gas turbine determines the temperature of the combustion gases emerging from the gas turbine and entering the heat exchanger.

It must not be less than the temperature of the water entering the heat exchanger for the acid dew point of the flue gas heat exchanger otherwise will damage quickly because of the chemical erosion. This is because the heat transfer coefficient between the water pipe boiler much larger than the

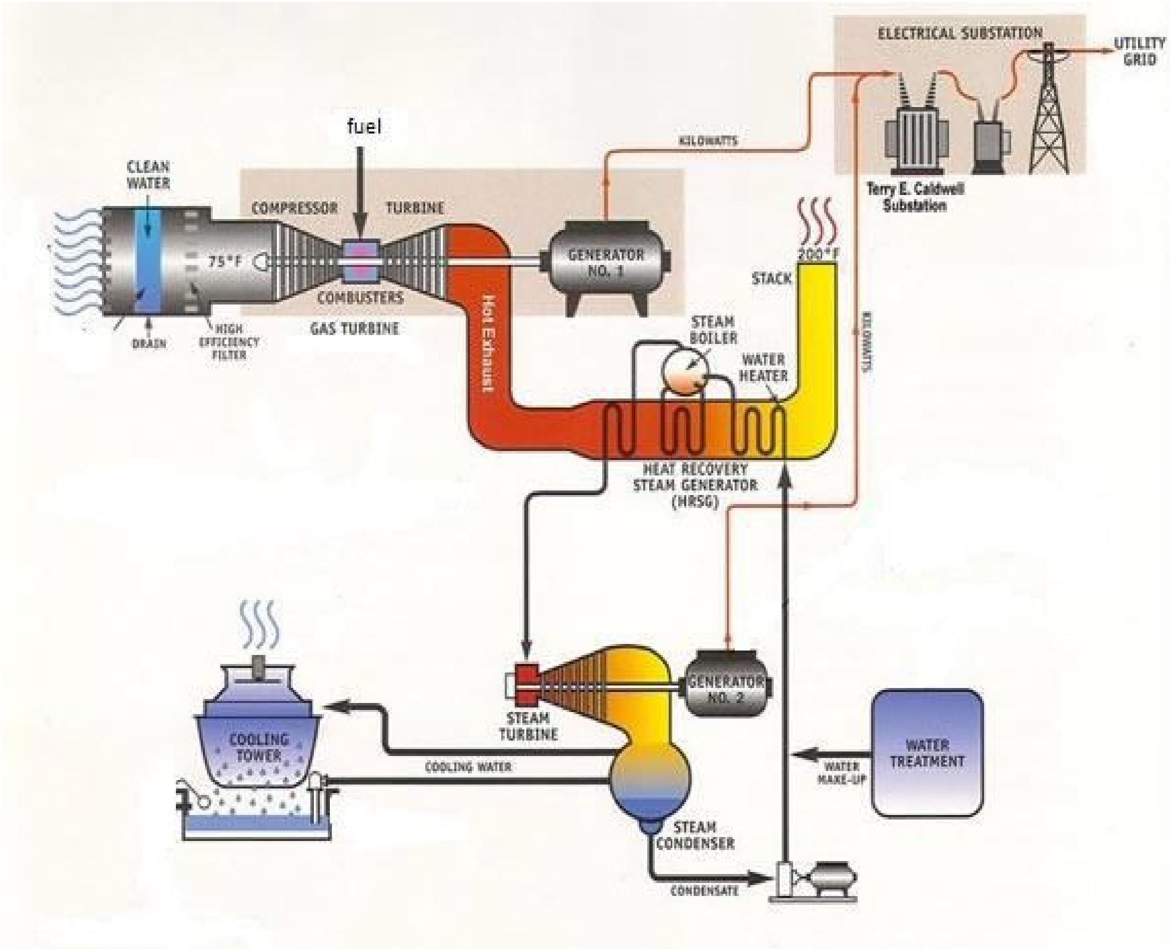


Figure (2.23) combined cycle power plant

coefficient of heat transfer between the gas and the pipes of the boiler, so the temperature of boiler tube is approximately equal to the temperature of the water, and therefore, if the temperature of the pipe is less than the dew point of acid gases burnt, which often contain the sulfur gases burnt intensified on the walls of the pipe and cause corrosion acidic her.

Usually prove the internal temperature of the water to the boiler at about 100 m, and this depends on the amount of sulfur in the fuel. If a clean fuel such as natural gas, the internal temperature of the water to the boiler can be less. Also that the minimum difference in temperature between the gas and steam, which is called a point narrow (Pinch point) must not be too small, otherwise it shall be the boiler too large and therefore high cost. Practically This should be the difference is greater than about 20 m and must preserve this difference the lower part of the boiler with hot gas, as in Figure (2.24). Where should remain the difference between the temperature of the hot gas and steam temperature gaseous larger than about 20 m. Can determine the materials used in the steam turbine maximum temperature of the gaseous vapor. [10]

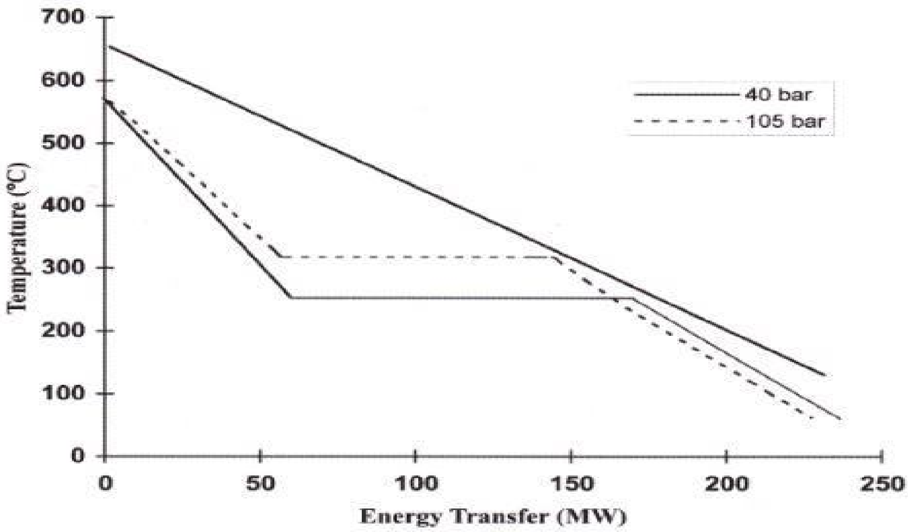


Figure (2.24) shows the scheme temperature with the amount of shifting the heat exchanger to heat

b) Combined Cycle With Additional Ignition:

Contains exhaust gas turbine a sufficient amount of oxygen (15% by volume) being able to be used in the process of burning a new one. In this type of cycles are dual uses hot combustion gases emerging from the gas turbine combustion chamber where it is attached to boiler re-ignited to raise the temperature. Ignition uses overtime to increase the production of steam and raise the temperature of the gaseous steam. The temperature of the combustion gases expelled from the gas turbine is a large ~ 600 m can rise to ~ 815 m using the ignition additional boiler in a basic design, the height

Increases the amount of steam produced roughly doubled and therefore the ability of the steam turbine. Can increase the temperature to ~ 1100 if the walls lined refractory material, and the walls if you use chilled water temperature can be increased more.

2.4.4 Electricity Market Requires Power Plants to be:

1. With high efficiency.
2. With low financial cost of capital.
3. Flexible operation.
4. Trustworthy full.

2.4.5 Combined Cycle Arrangement Systems:

a) System per Shaft

Consists of one gas turbine, one steam turbine, one generator and one heat exchanger, where the gas turbine and steam turbine electricity generator are online on a single column. The important advantages of this arrangement is the simplicity, speed of operation and that it trusted more than the order with multiple columns. It also features greater flexibility when maintenance increase operating flexibility through the use of hydraulic clutch pedal until the steam turbine can be separated when the station's management or operation of the gas turbine in simple cycle.

b) **System Multi-Shaft**

A gas turbine and a generator is connected to one or several gas turbines which is connected to a generator and several heat exchangers provide steam through the pipe joint compound to a separate steam turbine connected to a generator. Columns multi-system costs about 5% of the system per column. System features multi-pole higher efficiency when running under load core [11].

2.4.6 Thermal Efficiency of Combined Cycle

Increased thermal Overall efficiency for combined cycle about 40% to 56% in the past decade. Efficiency has become today's 60% [12] in all cases; the efficiency is greater than the efficiency of simple gas cycle gas or steam cycle individually.

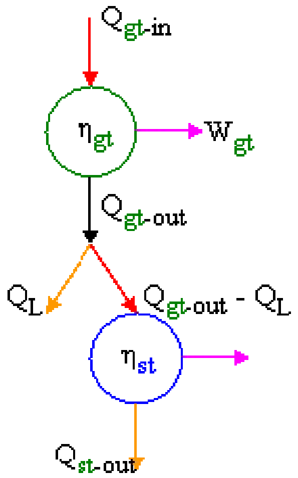


Fig (2.25) Thermal efficiency of combined cycle

Total Work(W_{total}) =work output of the gas turbine (Wgt) work output of the steam turbine (Wst)

Work piece total (W_{total}) = efficiency gas turbine (ηgt) × heat supplied to the gas turbine ($Q_{gt\text{-}in}$) + efficiency turbine steam (η_{st}) ×) heat rejected from the gas turbine (Q gt-out) - waste heat (QL) {.

$$\eta_{gt} = W_{gt} / Q_{gt-in} \qquad \textit{(gas turbine efficiency)} \qquad (2.56)$$

$$\eta_{st} = W_{st} / Q_{st-in} = W_{st} / Q_{gt-out} - Q_L \qquad \text{(steam turbine efficiency)} \qquad (2.57)$$

$$\begin{aligned} W_{total} &= W_{gt} + W_{st} \\ &= \eta_{gt} \times Q_{gt-in} + \eta_{st} \times (Q_{gt-out} - Q_L) \\ &= \eta_{gt} \times Q_{gt-in} + \eta_{st} \times Q_{gt-out} (1 - Q_L / Q_{gt-out}) \\ &= \eta_{gt} \times Q_{gt-in} + \eta_{st} \times (1 - \eta_{gt})(1 - Q_L / Q_{gt-out}) \end{aligned} \qquad (2.58)$$

Can be considered the item (QL / Qgt-out -1) heat exchanger efficiency (= efficiency of exploitation) which refers to the portion of the heat rejected from the course that you used the upper and lower cycle symbolized by the η_{gt} (efficiency heat exchanger originally equal to the product of the efficiency in the exploitation Jump efficiency, and efficiency is assumed here that the Jump = 100%, and thus the efficiency of the heat exchanger is equal to the efficiency of exploitation).

Calculated the total thermal efficiency of the combined cycle η_{cc} as follows:

$$\begin{aligned}
\eta_{cc} &= W_{total} / Q_{gt-in} \\
&= Q_{gt-in} \left\{ \eta_{gt} + \eta_{st} \eta_{gt} \left(1 - \eta_{gt} \right) \right\} / Q_{gt-in} \\
&= \eta_{gt} + \eta_{st} \eta_{gt} \left(1 - \eta_{gt} \right) \\
&= \eta_{gt} \left\{ \eta_{st} \eta_{gt} - \eta_{st} \eta_{gt} \eta_{gt} \right\}
\end{aligned} \qquad (2.59)$$

Possible to make the item $\eta_{cc} = \eta_{st}\, \eta_{gt}$ (efficiency of the steam plant, which includes the steam cycle and the heater together) where:

$$\begin{aligned}
\eta_{cc} &= W_{st} / \left(Q_{gt-out} - Q_L \right) \times \left(1 - Q_L / Q_{gt-out} \right) \\
&= W_{st} \Big/ \left(Q_{gt-out} \left(1 - Q_l / Q_{gt-out} \right) \right) \times \left(1 - Q_l / Q_{gt-out} \right)
\end{aligned} \qquad (2.60)$$

$\eta_{cc} = W_{st} / Q_{gt-out}$	(combined (steam & gas) efficiency

2.4.7 Heating Water Feeding Steam Heat Exchanger Drain

Traditionally used steam Leech of the steam turbine unit to raise the temperature of the water heater nourishing for about 100 ° C in order to raise the efficiency of the steam cycle. But if you have installed vapor pressure to raise the temperature of the water fed to the heater will raise the temperature of the combustion gases in the chimney, there is no need to draw the same amount of heat from the combustion gases. And therefore less efficient use of the heat in the gases expelled and this is a disadvantage of the traditional system of attrition. Less than the mass of steam passing through the turbine after attrition and thus less work and this also disadvantages.

CHAPTER III

GARRI COMBINE CYCLE POWER PLANT

CHAPTER III

Garri Combine Cycle Power Plant

3.1 Introduction

The process of converting the energy in a fuel into electric power involves the creation of mechanical work, which is then transformed into electric power by a generator. Depending on the fuel type and thermodynamic process the overall efficiency of this conversion can be as low as 20%. Simple gas turbine (GT) plants average just fewer than 30% efficiency on natural gas and around 25% on fuel oil.

To increase the overall efficiency of electric power plants, multiple processes can be combined to recover and utilize the residual heat energy in hot gases. In combined cycle mode, power plants can achieve electrical efficiencies up to 60%.Combined cycle operation employs a heat recovery steam generator (HRSG) that captures heat from high temperature exhaust gases to produce steam, which is then supplied to steam turbine to generate additional electric power which based on Rankine cycle.

The common type of combined cycle power plant utilizes gas turbine is called a Combined Cycle Gas Turbine (CCGT) plant. Because GT have low efficiency in simple cycle operation, the output produced by the steam turbine accounts for about half of the CCGT plant output

3.1.1 Combined Cycle Principle of Operation

The HRSG is basically a heat exchanger or rather a series of heat exchangers. The HRSG can rely a natural circulation or utilize forced circulation using pump. As the hot exhaust gases flow past the heat exchanger tubes in which hot water circulates, heat is absorbed causing the creation steam in the tube. The tubes are arranged in sections, or modules, each serving different functions in the production dry superheated steam.

The economizer is a heat exchanger that preheats the water to approach saturation temperature. As the hot exhaust gases flow past the evaporator tubes, heat is absorbed causing the creation of steam in tubes.

The superheated steam produced by the HRSG is supply to the steam turbine where it expands through the turbine blades, imparting rotation to the turbine shaft. The energy delivered to the generator drive shaft is converted into electricity. After exiting the steam turbine, the steam is sent to a condenser which routes the condensed water back to the HRSG.

3.1.2 Design Configuration:

Designs and configurations for HRSGs and steam turbines depend on the exhaust gas characteristics, steam requirements, and expected power plant operations. Because the exhaust gases from a gas turbine can react HRSGs for GTs may produce steam at multiple pressure levels to optimize energy recovery. The high pressure steam in a large CCGT plant can reach 40 – 110 bars. With a pressure HRSG, the steam turbines will typically have multiple steam admission points.

3.1.3 Plant Configuration [13]

The combine cycle power plant consists of two 206B combine cycle, two gas turbine, two HRSGs and one steam turbine are used in per block.

Each gas turbine type is PG600iB; its output is approximately 40MW for ISO condition. The gas turbine is normally operated with double fuel, LPG and light oil .The gas turbine generator, which is driven at 3000 rpm, is air cooler. Each gas turbine exhausts gas lead to its associated HRSG. There is diverter damper between the gas turbine exhaust and the HRSG. It allows the gas turbine to operate either in open cycle mode or in combined cycle mode.

The exhaust gas flow and temperature characteristics at the gas turbine exhaust will be changed with their load. The HRSGs are of the single – pressure type the main stream lines of each HRSG are led to the steam turbine. Steam turbine is condensing type with extraction steam.

3.1.4 Combined Cycle Power Plant –How it work

The CCPP works to produce electricity and capture waste heat from the gas turbine to increase efficiency and electrical output as follows:-

1. Gas turbine burns fuel
   - The gas turbine compresses air and mixes it with fuel that is heated to a very high temperature. The hot air-fuel mixture moves through the gas turbine blades, making them spin
   - The fast-spinning turbine drives a generator that converts a portion of the spinning energy into electricity
2. Heat recovery system captures exhaust
   - A heat recovery Steam Generator(HRSG) captures exhaust heat from the gas turbine that would otherwise escape through the exhaust stack
   - The HRSG creates steam from the gas turbine exhaust heat and delivers it to the steam turbine
3. Steam turbine delivers additional electricity
   - The steam turbine sends its energy to the generator drive shaft, where it is converted into additional electricity

3.1.5 Operating Scheme of the CCPP[13]:

Ambient air is filtered and led to the compressor of the gas turbine where it is compressed and fed to the combustors. In the combustors the compressed air is heated up to the turbine inlet temperature, the fuel combust before expanding in the turbine. After expansion the flue gases is led to the HRSG. Steam is generated in the HRSG by heat transfer from the flue to the feed water. The HRSG is a single pressure boiler. From the two parallel HRSGs the superheated HP steam is fed to the steam turbine

The expanded steam is condensed in a water cooled condenser. In order to obtain optimum utilization of the steam, the pressure at the exhaust is optimized to the condenser cooling system.

There are three reasons why inlet air temperature has a significant influence on the power out put and efficiency of the gas turbine:-

1. Increasing the ambient temperature reduces the density of the air consequently reduces the air mass flow into the compressor as constant volume engine. This the main reason for changes in the gas turbine power plant.
2. The specific power consumed by the compressor increases proportionally to the air intake temperature without a corresponding increase in the output from the turbine.
3. As the air temperature consumed rises and the mass flow decreases, the pressure ratio within the turbine is reduced. Due to the swallowing capacity of the turbine section and the reduced mass flow, the pressure at the turbine inlet of the gas turbine is reduced and applied inversely, of courses to the compressor, however, because its output is less than that of the turbine, the total balance is negative.

The feed water is fed back to the HRSGs by using three constant speed , high pressure feed – water pumps and by two constant speed low pressure feed water pumps. One pump is at standby and is switched on automatically in case of failure of a running pump. A minimum flow feed water pump check valve is provided to ensure minimum flow through the feed water pump to increase the operational flexibility during start up, shut down and abnormal operating conditions separated bypass stations for high pressure steam source. The high pressure bypass stations are designed to accommodate 100% of the maximum steam production in to the condenser. Each bypass station consists of an isolation valve, a steam pressure reducing valve, a de-superheating station with the associated measurement, control and protection device. Main condensate is used for de-superheating the steam down to the saturation point before entering in the condenser

The Liquid Petroleum Gas (LPG) heating steam is from auxiliary boiler when unit start- up and in simple cycle. In normal operation LPG heating steam is from steam turbine extraction. The extraction pressure and temperature are designed to meet the LPG heating required.

3.1.6 Gas Turbine [13]

a) Frame Six Gas Turbine

The turbine and accessories are mounted on a common base sized within normal shipping limits and shipped as an integral assembly that has been successfully completed a fired no load factory test. Advantages of package concept are obvious, such as fast operation, minimum site investment are:-

1- Low installed cost owing to standardization and to factory assembly and test, this make the installation of the station easy and keep the cost per installed kilowatt low because the package power station is quickly ready to be put in to operation.

2- Minimum site investment: the required space and site preparations are minimum; the overall dimensions of the slab type foundation are only 38*6 m^2

3- Low stand – by cost: fast start- up and shut down reduce conventional stand- by costs. The power requirements to keep the plant in stand – by condition are significantly lower than those for other types of prime movers.

4- Maximum application flexibility: the package power plant may be operated either in parallel with existing plant or as a completely isolated station, these units have been used widely for base, peaking and even emergency service.

5- Quick start is one of the significant features of this power unit. Only about one quarter of an hour is need from standstill to hooking up with the power grid.

6- Control reliability: the “SPEEDTRONIC” control provides accurate sequential control and instant protection .The station can be equipped for remote control, starting synchronizing and loading

7- Maintenance cost is comparatively low.

b) PG6001 (B) Model Gas Turbine Generating Set Performance

1. Site condition require atmospheric condition and field condition atmospheric pressure: 96.6 kPa

Compressor inlet condition: 40C, 38% relative humidity

Inlet loss: 101mm H_2O

Exhaust loss: 350mm H_2O

Equipment operating time: less than 100hr

2. PG6001 (B) Bass load performance

Table (3.1) Base load Performance

Fuel type	**LPG**	**Distillate oil**
Generator terminal output (KW)	32366	31602
Heat rate (LHV)(KJ/KWh)	12235	12341
Heat consumption (LHV) KJ/h*$\mathbf{10^6}$	396	390
Exhaust gas flow (Kg /h)* $\mathbf{10^3}$	450	452
Exhaust temperature (°C)	564	564

c) Main Technical Date of Unit

1. Emission Data (Estimated, LPG)

Table (3.2) Emission data

Von brand number	above 94
NO_x at base load, ISO condition	244ppmvd

2. Sound Data (Estimated)

Table (3.3) estimated Sound data

At 121.92m. with level 2 silencing	60d BA
At .61ft.or “near filed”	90d BA with unit full lagged

3. Estimated Lateral Critical Speeds

(GT. ONLY, AS ON FACTORY NO LOAD TEST)

Table (3.4) estimated lateral critical speeds

1^{st}	1658-1877 rpm
2^{nd}	3256-3908-rpm
3^{rd}	7049-7360rpm

4. Startup Times

Table (3.5) Startup times

Type	normal start (min)	fast start (mini: sec)
GT start (to FSNL)	12	7:10
LOADING	4	2.00
Total	16	9.10

5. Note: Including (For Gas Turbine and Generator)

Trip	25.4mm/s

6. Weights and Dimensions (Approx)

Table (3.6): Weights and dimensions

Weight (t)	Dimensions (L X W X H) (m)
97.2	12.55*3.28*3.8
10.45	5

Materials

Table (3.7) Parts & Material

Part	**materials**
Turbine:	
1st stage buckets	IN-738WITH COATING /GTD 111(Or)
2nd stage buckets	in-738
3rd stage buckets	u-500
1st stage nozzle	fsx-414
2nd stage nozzle	fsx-414
3rd stage nozzle	fsx-414
Wheel	Cr-Mo-v steel
Shell	nodular iron
Tie bolt	crucible 422
Compressor:	
Rotor blades	403-ss switch Ni-Cd coating
Stator blades	403-ss switch Ni-Cd coating
Rotor	Ni-Cr-mo-v steel /Cr-mo-v
inlet, forward and shaft casing	Gray Iron
Discharge casing	nodular iron
Combustor:	
Liner	Hastelloy X
Trans. Piece	Hastelloy X
OUTER CASING	A 516(PRESSURE VESSEL QUAL)

8. Parts Estimated Life's

Table (3.8): Parts estimated life's

No	name of part	hours to Inspection	No of repairs	total life (hours)
1	combustion liners	8,000-10,000	2	24,000-30.000
2	transition pieces	8,000-10,000	2	24,000-30.000
3	1st stage nozzle	24,000	2	72,000
4	1st stage buckets	24,000	NTL	24,000
5	2nd stage nozzle	24,000	3	69,000
6	2nd stage buckets	24,000	2	72,000(or 4,800 Statrs)
7	3rd stage nozzle	84,000	3	192,000
8	3 rd stage buckets	24,000	2	72,000 (or 4,800 Statrs)
9	compressor blade	100,000	NTL	100,000

9. MS6001B Inspection Intervals

Table (3.9): MS6001B Inspection Intervals

Starting frequency	Fuel	inspection intervals	Starting per fired hours
1/1000	Gas	24	48
	Distillate	24	48
1/100	Gas	12-16	24-32
	Distillate	12-16	24-32
1/50	Gas	10-14	20-28
	Distillate	10-14	20-28
1/5 up to 1/50	Gas	9-12 or 800	18-24 or 16,000

Note:

1- No of fire hrs corrected = f1 * No of hrs base +6* No of hrs peak on any fuel

F1:1.0(gas) F1:1.5(distillate)

2- No of starts corrected= No of normal starts +20 *No of emergency starts

+8 *No of trips + 2*No of fast load starts

d) Exhaust Gas System

Under the G.T. single cycle condition or HRSG load regulation condition or the condition of HRSG out of service, Exhaust Gas System will bypass G.T. exhaust gas into the air directly.

Included in this system are G.T. outlet transition duct, the three – way diverter damper, bypass stack support, silencer, bypass stack and relevant expansion joints and transition duct.

The three – way diverter damper is of motor driven type with single valve plate. Dual – layer flexible metallic hard sealing with high-pressure sealing air is assumed for three-way diverter damper sealing construction so as to ensure a high sealing efficiency.

Single valve plate for three –way diverter dam per can ensure that the gas turbine exhaust gas opening will not be blocked in the case of disoperation .Moreover, The amount of flue gas can be adjusted with ease and high efficiency by ways of locating the valve plate in different positions which plays an important role in boiler start- up and load regulation.

3.1.7 Water/Steam Cycle

Thermodynamic system of Gas-Steam Combine Cycling power station including Gas Turbine (GT) system, Heat Recovery Steam Generator including Gas Turbine (GT) system, Heat Recovery Steam Generator (HRSG), Steam Turbine (ST) steam and water system.

Two blocks of Gas- Steam Combine Cycling units are designed for this project, each block consists of two (2) Gas Turbine, two (2) HRSGs and one (1) Steam Turbine.

The water/steam system mainly including HRSG steam and water system, steam turbine, generator and steam turbine by – pass system condenser and condensate pump, feed water pump, gland steam cooler, valves and pipes etc. The Water/ Steam flow diagram refers to appendix (C)

3.1.8 HRSG and Its Auxiliary System

The HRSG is natural circulating, no- additional firing horizontal. AII the auxiliary system including, deaerator and its storage tank for two HRSG, low – pressure circulation system, sampling system HRSG inlet and outlet flue- gas duct flue by –pass stack and diverter and feed-water system.

a) The Main Steam and Bypass System

The main steam system connects the two (2) HRSGs supply steam to one (1) steam turbine. The superheated steam piping of each HRSG supplies the steam headers located on this steam turbine side. Steam is directed from this header to the steam turbine stop valves.

The steam turbine bypass system allows each HRSG to be started independently. The bypass steam is expanded and de-superheated prior to entering the condenser. The bypass system is so dimensioned as to condition 100% of the flow of each HRSG at the normal operating pressure and allow:

- Conditioning of the steam piping from cold state
- HRSG pressure change rate can be controlled during startup
- Steam dump in case of steam turbine load rejection, so that the gas turbine can be kept operating. The main steam and bypass system diagram refer to appendix (D)

b) Deaerator Heating Steam System

Feed-water is heated and desecrated in spray type feed-water tank, this horizontal cylindrical the water dissolved oxygen content to acceptable value by the HRSGs. The feed-water heating before startup is provided auxiliary boiler steam fuel oil is light diesel oil. But if fuel is LPG, FEED-water heating steam before startup is provided also by auxiliary boiler. In normal operation, the heating steam is provided by the HRSG low temperature evaporators or, by the HP drummed low load, and diffused into the water mass by vertical perforated tubes. The deaerator heating steam system Diagram reefs to appendix (E).

c) Feed- Water System

The boiler feed-water system consists of three pumps: two pumps for two HRSG, the third pump is standby unit.

Feed- water tank level should keep sufficient pressure at feed water pumps suction under any conditions. The feed – water pumps are installed under the feed water tank to feed the HRSG economizers

The pumps are the horizontal centrifugal type, the multistage pumps are splash lubricated and directly driven at constant speed by electric motors. Sealing is performed by stuffing – box gland packing, each pump is installed on a foundation together with its motor and is equipped with a recirculation line through a check valve for minimum flow operation of each motor- driven feed-water pump. The feed- water flow to each HRSG is controlled by the drum level control valves.

d) Condensate System

Two condensate pumps, one in operation and the other stand- by, are provided for one steam turbine to transits condensate through turbine gland steam cooler to the desecrator. Each pump is equipped with a recirculation line for minimum flow operation of each motor- driven condensate pump. Some spray water from outlet of condensate pumps are provided for turbine bypass at temperature, the condensate system diagram appendix (F).

e) Circulating Cooling Water System

The cooling water system provides cooling water to the steam turbine condenser. The warm water is returned to a multi – cell induced mechanic draft cooling tower where it is cooled and collected in the tower basin. The condenser is equipped with a tube cleaning device of the sponge ball type. The Circulating Cooling Water System Diagram appendix (G)

f) Service Cooling Water System

Cooling water is fed to auxiliary equipment of gas turbine is from circulating water pump room and return to circulating water cooling tower. Steam turbine and gas turbine cooling water, steam turbine generator air cooler cooling water is from main cooling water system .

Service water comes from circulating water pump room is used to cool pump bearing and return to circulating water pump room. Backwater will be discharged to waste water pit if service water is polluted, the service and cooling water system diagram appendix(H)

g) Make- up Water and Filling Water System

One storage water tank is used to provided water to two units Make – up water while one storage water tank is used for the two hot well condensers by using two desecrator filling water pump The make – up water and filling water system diagram appendix (I)

3.1.9 Instrument and Service Compressor Air System

Three screw type air compressors and two air– drying devices are provided for instrument and control. Two air storage tanks are provided. The compressed air for the LPG treatment device, demi water plant etc is provided. The instrument and service compressed Air system Diagram appendix (J)

3.1.10 Condenser Air Evacuation System

Two 100% liquid-ring vacuum pump are provided to one steam turbine, one in work during normal operation while the two may in work during start- up, the condenser air evacuation system diagram appendix (K)

3.1.11 HRSG Low Pressure Circulating System

Each HRSG is provided with two low pressure circulating pumps; one is in operation while the other in the stand – by condition, The HRSG low pressure circulating system diagram appendix (L)

3.1.12 LPG Heating Steam System

At the auxiliary boiler outlet there are two lines, one is connected to the LPG heating during unit start-up and at single cycle operation, the other is connected to the deaerator for feed water heating during the unit start-up. There is one auxiliary boiler only for this project the fuel of auxiliary boiler will be light diesel

In normal operation (combined cycle) the LPG heating steam is coming from steam turbine extraction

3.1.13 Heat Recovery Steam Generator (HRSG)

a) General

The heat recovery system generator components are boiler inlet duct, HRSG boiler proper, main stack, relevant duct accessories and its control system.

HRSG is mainly used to recover heat from G.T exhaust gas, produce HP steam to be led to steam Turbine, driving generator. In addition HRSG boiler can produce LP steam to be provided for deaerators.

b) Type of Boiler

The boiler is a natural circulation water-tube boiler horizontal flue and vertical helical finned pipe, Located in horizontal flue are helical finned pipes of heat-transmitting components. The LPG heating steam system diagram refers to appendix (M).

3.1.14 LPG Fuel

a) Gas Turbine Model: PG6001B / Fuel: LPG/ Basic Load

Table (3.10) Basic Load Data

NO	Item	Unit	Numerical Value			
1	Unit operating condition	/	Combined cycle, basic load			
2	Ambient temperature	C	15	15.6	40	47.2
3	Ambient pressure	KPa	101.325	96.70	96.6	96.23
4	Relative humidity	%	60	27	38	28
5	G.T. exhaust flow rate	t/h	523	498	450	435
6	G.T. exhaust temperature	C	547	548	564	569
7	G.T. exhaust composition					
	N_2	%	74.49	74.89	73.65	73.54
	NO_2	%	7.69	7.73	7.63	7.26
	H_2O	%	9.31	8.82	10.22	10.31
	SO_2	%	0	0	0	0
	O_2	%	8.51	8.55	8.77	8.89
8	Boiler rated steam Pressure	Mpa (g)	6.9	6.9	6.9	6.9
9	Boiler rated steam temperature	C	462	464	468	468
10	Boiler rated steam output	t/h	69.61	66.63	63.78	62.74
11	Boiler blow down rate	%	1	1	1	1
12	LP desecrating boiler rated Operating pressure	Mpa (g)	0.2	0.2	0.2	0.2
13	LP desecrating boiler steam output	t/h	9.32	8.81	7.47	7.07
14	Boiler exhaust gas temperature	C	157.5	156.8	153.9	153.0
15	Boiler feed water temperature	C	104	104	104	104
16	Heat recover utilization efficiency	%	72.25	72.42	73.77	74.16
17	Flue gas resistance inside boiler	Pa	<3500(T-shaped arrangement)			
18	Boiler overall dimensions	mm	7190×5500×16630			

b) G.T Model: PG6001B / Fuel: LPG/ Partial Load

Table (3.11) Partial Load Data

NO	Item	Unit	Numerical Value			
1	Unit operating condition	/	Combined cycle, Partial load			
2	G.T. Load	%	50	75	90	100
3	Ambient temperature	C	40	40	40	40
4	Ambient pressure	kpa	96.6	96.6	96.6	96.6
5	Relative humidity	%	38	38	38	38
6	G.T. exhaust flow rate	th	391	391	424	450
7	G.T. exhaust temperature	C	463	559	557	564
8	G.T. exhaust composition					
	N_2	%	74.61	73.83	73.73	73.65
	NO_2	%	5.35	6.98	7.19	7.36
	H_2O	%	8.6	9.82	10.4	10.22
	SO_2	%	0	0	0	0
	O_2	%	11.98	9.37	9.4	8.77s
9	Boiler rated steam Pressure (gage pressure)	Mpa (g)	5.2	6.4	6.9	6.9
10	Boiler rated steam temperature	C	462	468	468	468
11	Boiler rated steam output	t/h	31.35	42.22	58.81	63.78
12	Boiler blow down rate	%	1	1	1	1
13	LP deaerating boiler rated Operating pressure	Mpa (g)	0.2	0.2	0.2	0.2
14	LP deaerating boiler steam output	t/h	6.87	5.08	7.15	7.47
15	Boiler exhaust gas temperature	C	153.7	148.8	153.3	153.9
16	Deaeraing boiler Feed water temperature	C	104	104	104	104
17	Heat recover utilization efficiency	%	67.60	74.38	73.50	73.77
18	Flue gas resistance inside boiler	Pa	<3500(T-shape darranggement)			
19	Boiler overall dimensions	mm	7190×5500×16630			

3.1.15 Boiler Heating Surfaces Thermodynamic Properties (for Design Condition)

a) Diesel Oil as G.T. Fuel.

i. Design Condition:

Table (3.12) Diesel Oil as G.T. Fuel Design Condition

NO	Item	Unit	Super-heater	Evaporating Tube Bank	Economi-zer	LP Boiler
1	Inlet flue gas temperature	C	564	488.3	297.3	191.3
2	Outlet flue gas temperature	C	488.3	297.3	191.3	153.9
3	Inlet medium temperature	C	288.6	288.6	104	133.4
4	Outlet medium temperature	C	468	288.6	281.5	133.4
5	Temperature difference	C	129.9	60.8	41.7	36.1
6	Average flue gas temperature	C	512.7	349.4	234.5	169.4
7	Average flue gas speed	m/s	17.4	14.9	12.2	10.4
8	Heating area	M2	2596.3	8381.5	6942.3	2967.7

Ambient Temperature=40 C, Atmosphere Pressure=0.966Bar, RH=38%

Pinch Point: 8.7 C (Design Condition, Diesel)

b) PLG as G.T. Fuel.

i. Design Condition:

Ambient Temperature=40 C, Atmosphere Pressure=0.966Bar, RH=38%

3.1.16 Boiler Performance Features

a) A Safe and Reliable Operation

With gas – turbine exhaust heat recovery natural – circulation boilers widely used both at home and abroad to serve as its prototype, the type of boiler is advanced and proven.

Table (3.13) PLG as G.T. Fuel Design Condition

NO	Item	Unit	Super heater	Evaporating Tube Bank	Eco-nomizer	LP Boiler
1	Inlet flue gas temperature	C	564	488.3	297.4	191.0
2	Outlet flue gas temperature	C	488.3	297.4	191.0	153.9
3	Inlet medium temperature	C	288.6	288.6	104	133.4
4	Outlet medium temperature	C	468	288.6	281.5	133.4
5	Temperature difference	C	129.9	61.0	41.7	36.0
6	Average flue gas temperature	C	512.7	349.7	234.5	169.3
7	Average flue gas speed	m/s	17.3	14.9	12.1	10.4
8	Heating area	M2	2596.3	8381.5	6942.3	2967.7

Pinch Point: 8.8 C (Design Condition, LPG)

b) Incorporation of the Specific Features of Both Natural and Forced Circulation Boilers

With the adoption of a natural circulation mode on the boiler water and steam side, and a forced circulation mode on the gas side, the proposed boiler incorporates the characteristic feature of both the natural and forced circulating boilers.

c) High Maneuverability

A small water capacity with a rapid start-up and a high maneuverability constitute the main features of this type of boiler. Its start-up performance excel those of analogous types of foreign and home-made boiler. These features are especially important for gas turbine generating stations often subjected to a high frequency of stat-ups and shutdowns

d) Saving in Operation Expenses

No forced – circulation pumps are needed for the present boiler which leads to not only the elimination of an unreliable equipment item, but also significant saving in auxiliary power consumption and a low operation cost.

e) A good Gas Flow Section Shape

The type of boiler has a more favorable length and width ratio as compared with that of an ordinary gas –turbine exhaust heat recovery boiler with the boiler furnace fully and properly filled with flue gases, a uniform heat load and a small heat excursion in gas flow distribution.

3.2 Simulation of Combined Cycle Plant

3.2.1 Preface:

The simulation of combined cycle by using different simulators which are HYSYS® and CyclePad is presented. These programs specifically designed to make a computer simulation without need for its determination where the station data interference accurately within the program that can make a numerical analysis according to the type of study and depending on the desired results. After simulating the plant then comparing the designs using different simulators with the original plant design of GARRI station and comparing the result from HYSYS simulator and Cyclepad simulator to choose the perfect software that gives us the nearest result to that result taken from GARRI plant. Data of GARRI power plant will represent, after that HYSYS and Cyclepad models will be made. After comparing results the perfect simulation software is selected for the model.

3.2.2 ASPEN HYSYS® Simulation Software [14]:

HYSYS is powerful engineering simulation tool, has been uniquely created with respect to the program architecture, interface design, engineering capabilities, and interactive operation. The integrated steady state and dynamic modeling capabilities, where the same model can be evaluated from either perspective with full sharing of process information, represent significant advancement in the engineering software industry.

The various components that comprise HYSYS provide an extremely approach to steady state modeling. At a fundamental level, the comprehensive selection of operations and property methods allows you to model a wide range of processes with confidence. Perhaps even more important is how the HYSYS approach to modeling maximizes your return on simulation time through increased process understanding.

HYSYS is such powerful software for simulation. It's widely used in universities and colleges in research, development, modeling and design. For all these reasons we choose HYSYS to make simulation in this research.

3.2.3 Cyclepad Simulation Software [15]:

Cyclepad software is made by Professor Chih Wu (2004) in United States. It has been used at Northwestern University, the US Naval Academy and Oxford University. It is powerful, mature, user friendly package developed to simulate thermodynamic devices and cycles.

CyclePad offers students a rich, exploratory learning environment in which they apply their theoretical thermodynamics knowledge by constructing thermodynamic cycles, performing a wide range of efficiency analyses. it has been in active use in a range of thermodynamics courses at the Naval Academy and elsewhere since 1996. it makes it possible for engineering students to engage in design activities earlier in the curriculum than would otherwise be possible. Qualitative evaluations of CyclePad have shown that students who use CyclePad have a deeper understanding of thermodynamics equations and technical terms. CyclePad supports an unguided approach to exploration and design. While active learning and intense exploration have been shown to be more effective for learning and transfer than more highly directed. So For all these reasons we choose Cyclepad to make simulation in this research.

3.3 GARRI Station of Sudan Power Plant:

GARRI station power plant is one of the major sources of electric power in Sudan. It builds north of Khartoum in Garri city. It contain two combined cycle gas turbine station, every station had two blocks which each block consist two gas turbines and one steam turbine. GARRI generation is approximately 60 MW of power.

The data of this project are based on the document of GARRI Power Plant Station of Sudan. And below we will present the layout of one block on GARRI and its generation and also efficiency on this chapter.

3.3.1 Actual Data of GARRI Station Process:

Different tables of data of the process is used in the project. The real data is extracted of the combined cycle of GARRI process, to produced total energy, as the auxiliary one used, and the total efficiency of the cycle. Figure (3.1) shows the performance of Block 1 of GARRI power plant which explains the net efficiency of block with losses and without losses, net generation, and net heat rate and fuel consumption.

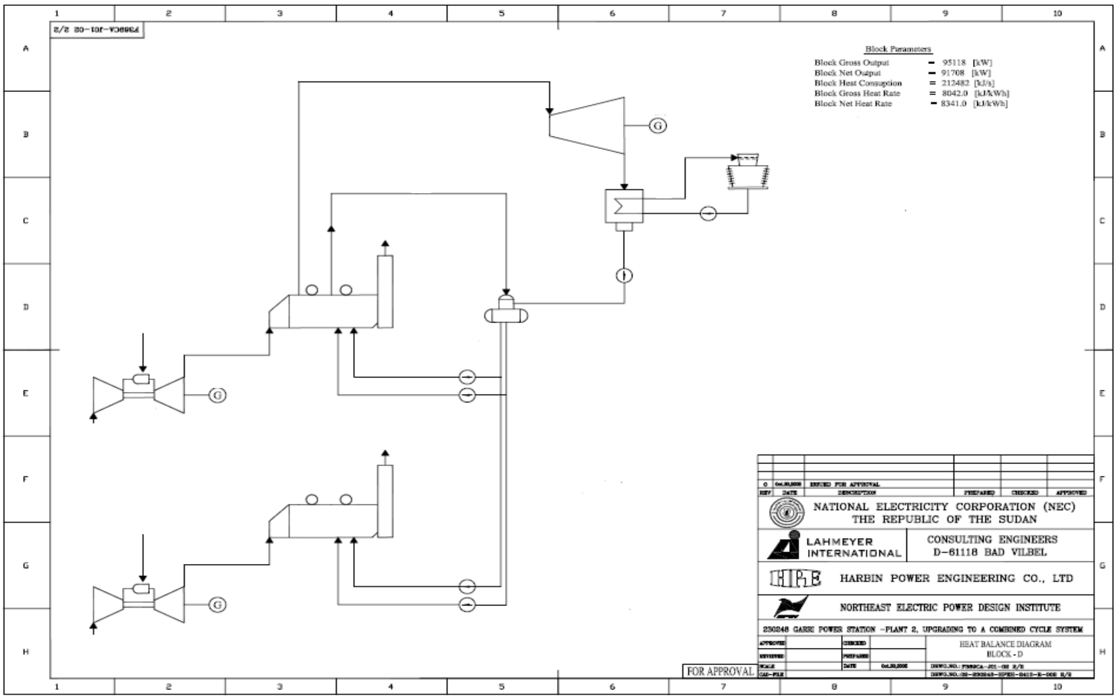


Figure (3.1): layout of one block of GARRI plant

Table (3.14): Performance of Block1 of GARRI Power Plant

ITEM	UNIT	PLANT 1 BLOCK 1				
		GT1	GT2	ST1	Block 1	
Gross Generation	MW	32.57	30.58	31.97	95.12	
Station Service	MW	1.43	0.53		1.96	
Net Generation	MW	31.14	30.05	31.97	93.16	
Service Hour	hr	24.00	24.00	24.00		
No. of Start	time	335	1,137	192		
No. of Start	time	0.00	0.00	0.00		
Operating Hours	hr	65,058.84	63,865.85	52,550.67		
Availability	%	100.00	100.00	100.00	100.00	
Capacity Factor	%	89.61	84.57	69.21	81.61	
hcgo Consumption	ton	0.00	0.00		0.00	
FO Consumption	ton	201.40	191.00		392.40	
FO Consumption	Litre	239,761.90	227,380.95		467,142.86	
Gross Generation	MWh	0.03	0.03	0.03	0.10	
Gross Generation Cost	Ton/MWh	6,183.04	6,246.93		4,125.40	
Gross Generation Cost	SDG/KWh	13,472.83	13,612.07		8,989.25	
Gross Heat Rate	kJ/kWh	283,547,803.40	286,478,135.73		189,186,816.38	
Net Heat Rate	kJ/kWh	296,567,530.42	291,531,669.16		193,167,216.99	
Specific Heat Rate	kg/kWh	6,183.04	6,246.93		4,125.40	
Gross Efficiency	%	29.29	28.16		37.30	28.7
Net Efficiency	%	27.87	26.93		36.20	27.4
LPG Heating Value (LHV)	kJ/kg	45,125.0				
LPG COST	SDG/Ton	600.0				

.

Table (3.15): Energy produced by GARRI station process

Generated energy	**(MW)**
Gas Turbine	63.148
Steam Turbine	31.970
TOTAL	95.118

Table (3.16): Auxiliary energy by GARRI station process

Auxiliary energy	**(MW)**
2 Compressors	26.600
Recycle HP pump	0.945
Block power input	0.1414
Steam recycling pump	0.550
Refrigeration tower	0.388
Transformation losses	7.680
Others …	0.5935
TOTAL	36.8979

Table (3.17): Net energies and efficiency

Net energies	**(MW)**
Net energy	58.220
Generated net heat	212.482
LPG consumption (kg/h)	4708.7
Plant efficiency	27.4 %

Next we will show simulation of GARRI power plant by using HYSYS and CyclePad Simulators.

3.4 Simulation of Combined Cycle Power Plant with HYSYS® Simulator:

3.4.1 Type of Simulation:

The steady-state is a characteristic of a condition, such as value, rate, periodicity, or amplitude, exhibiting only negligible change over an arbitrary long period of time (infinite).

3.4.2 Assumptions:

- I have supposed the camera of combustion of the process from GARRI station as a conversion reactor in the HYSYS®.
- The conversion is 100% in the reactor.
- In the compressor and the turbines the efficiencies are adiabatic.
- The components of the natural gas are: methane, ethane and nitrogen.
- The natural gas in the feed comes directly at the pressure of 23 bars.
- It's supposed worthless the mechanical losses.
- I supposed that isn’t losses on the conversion energy.
- The losses of heat have been rejected in each unit (turbine, compressor, boiler and HRSG adiabatic).

3.4.3 Constraints:

The Constraints are the restrictions or limitations that placed on requirements or design of your process. The constraints of the process are shown in the table (4.18):

Table (3.18): Constraints values

Constraints	values
Temperature combustion	< 1500 C
Temperature steam turbine	< 600 C
Pressure cycle steam	< 170 bars

3.4.4 Fluid Packages:

In HYSYS®, all necessary information pertaining to pure component flash and physical property calculations in contained within the Fluid Package. This approach allows you to define all the required information inside a single entity. There are four keys advantages to this approach and are listed below:

- All associated information is defined in a single location, allowing for easy creation and modification of the information.

- Fluid packages can be exported can be exported and imported as completely defined packages for use in any simulation.
- Fluid packages can be cloned, which simplifies the task of making small changes to a complex Fluid package.
- Multiple Fluid Packages can be used in the same simulation; however; they are all defined inside the common Simulation Basis Manager.

Fluid package to simulate a Gas Power Plant was selected. The package that treat hydrocarbons flow (natural gas), and the results of each fluid package, were compared with the data of GARRI station process.

The compared results of temperature, pressure and work for the i operations with the different data for each fluid package are shown in table(4.19)

Table (3.19): Temperature and Pressure data for each fluid package tested.

	SRK	**GARRI**
T(C) exit compressor	384.5	364
kW compressor	$2.88x10^4$	$2,66x10^4$
T(C) combustion	1271	1280
MW net gas turbine	95.83	60
T(C) exit gas turbine	913	913
T(C) exit gases HRSG	594.7	571
T(C) exit steam turbine	109.3	150
MW steam turbine	21.33	30
T(C) exit pump HP	104	100

According to the results of temperatures, pressures and works, assume the thermodynamic model SRK is chosen, since it is the one that resembles the results of GARRI station process. For similarity of results that caught WILSON, but this fluid package, it is not correct, since liquid is obtained in the exit of the combustion camera. That is unthinkable in a combustion camera to a temperature of 1500^o C and a pressure of 23 bars.

During the simulation of HYSYS in stationary state mode, there are many problems to choose the fluid package, since none of the thermodynamic models resembled the results of the process of GARRI station. Most of them had the problem that we obtained liquid in the exit of the reactor, when this occurred, discarded the thermodynamic package.

3.4.5 Components of the Fluid Package:

The components that I have used for the simulation of Gas Power Plant in HYSYS® software are the following ones:

- Butane
- Butene
- Propane
- Propene
- Ethane
- CO_2
- H_2O
- Nitrogen
- Oxygen

Table(3.20) give the composition of the natural gas that feeds the power plant.

Table (3.20): Composition of the Fuel (LPG)

Component	% (Mass)
Butane	**0.265**
Butene	**0.1885**
Propane	**0.3456**
Propene	**0.1798**
Ethane	**0.0027**
Nitrogen	**0.0184**

3.4.6 Combustion Reaction:

During the definition of the fluid package, assume that also defined the reaction that takes place in the combustion camera, where it mixes the natural gas with the air that it comes from the compressor. The reaction in the reactor is the following one:

$$(C_4H_{10} + C_4H_8 + C_3H_8 + C_3H_6 + C_2H_6 + N_2) + 25.5(O2 + 3.76\ N_2) \rightarrow 16\ CO_2 + 19\ H_2O + 96.88\ N_2$$

In the HYSYS® software, the combustion reaction between the natural gas and air is defined like the next figure:

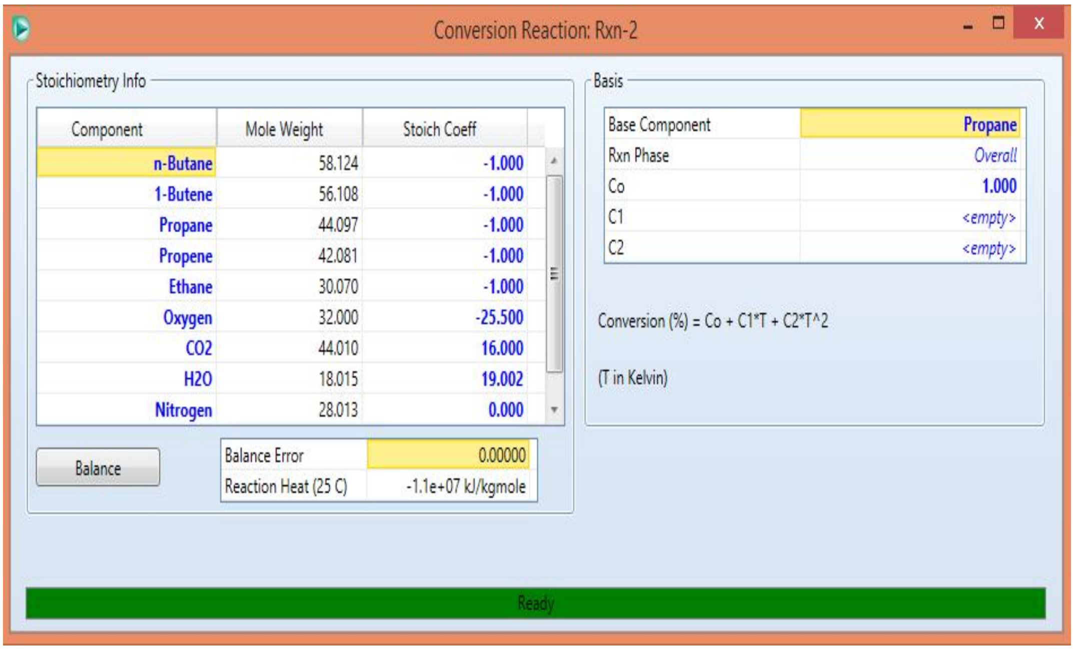


Figure (3.2): Definition of the combustion reaction in HYSYS®

3.4.7 Input Parameters:

The table (3.21) explains the Material streams of worksheet that entered to HYSYS simulation. Data was taken from GARRI combined cycle power plant.

Table (3.21): Material streams

	Unit	AIR IN	AIR OUT	GAS IN	COMBUSTION	LIQUID OUT	HRSG IN	EXIT GAS	HRSG WATER	HRSG STEAM
Vapour Fraction		1	1	1	1	0	1	1	0	1
Temperature	C	38	307.840905	45	1271	1271	855.589639	524.636495	104.531017	465
Pressure	kPa	96.6	880	880	880	880	221.5	221.5	4300	4300
Molar Flow	kgmole/h	30988.60022	30988.60022	385.92527	31374.52549	0	31374.52549	31374.52549	6777.714071	6777.714071
Mass Flow	kg/s	249.2	897120	5.16	915696	0	915696	915696	33.917	122101.2
Liquid Volume Flow	m3/h	1019.948323	1019.948323	33.984031	1053.932354	0	1053.932354	1053.932354	122.347605	122.347605
Heat Flow	MW	3.14975	71.703134	-6.710529	361.578123	0	228.743543	129.440499	-527.034275	-427.731231
	Unit	STEAM OUT	Cool Water	Water In	Water Out	Vapor Out	Steam in	Steam Extraction	Hot Water	......
Vapour Fraction		1	0	0	0	1	1	1	0.931534	....
Temperature	C	193.158101	41	41.042054	104	104	193.158101	193.158101	46.005717	....
Pressure	kPa	380	10	380	380	380	380	380	10	......
Molar Flow	kgmole/h	6777.714071	6178.216998	6178.216998	6777.714071	0	6178.216998	599.497073	6178.216998	....
Mass Flow	kg/s	122101.2	111301.2	111301.2	122101.2	0	111301.2	3.713	111301.2	......
Liquid Volume Flow	m3/h	122.347605	111.52581	111.52581	122.347605	0	111.52581	10.821795	111.52581	......
Heat Flow	MW	-444.798005	-489.071405	-489.056078	-527.221938	0	-405.45508	-39.342926	-418.904387	......

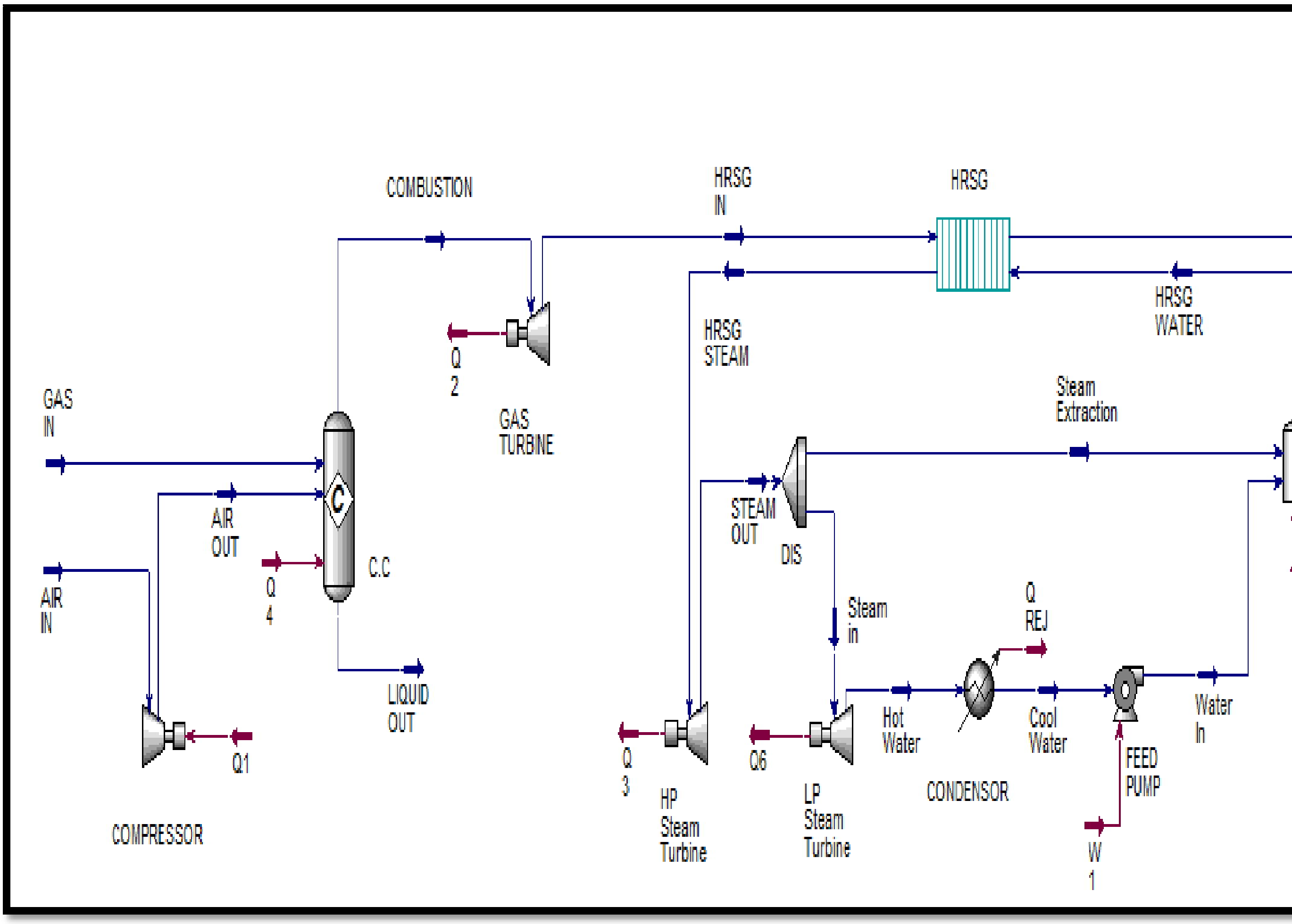


Figure (3.3): HYSYS simulation review

3.4.8 Results of the Steady-State Simulation with HYSYS:

For the calculation of the efficiency of the combined cycle are needed the net work produced by the cycle. The net work corresponds to the one generated by the turbines less the one consumed by the bomb and the compressor.

The results of the simulated cycle are shown in the following tables, where the efficiencies of compressors, turbines and bomb are included; the produced energy and the one consumed for the total calculation of combined cycle. The global efficiency of the plant is obtained relating the Wnet obtained with the heat that we put in the system through the combustion camera.

Table (3.22): Energy Streams result

	Unit	**GT**	**HP ST**	**LP ST**	**COMP**	**Q add**
Heat Flow	MW	132.8346	17.06678	13.44931	68.55338	296.5855
	Unit	Q rej	Q Dearator	W HPP	W FP	
Heat Flow	MW	70.16702	1.177066	0.187663	0.015327	

Table (3.23): Efficiencies of turbines, compressors and the pump

Efficiency	**(%)**
Compressor	83 %
Gas Turbine	83 %
Steam Turbine	83 %
Pump	75 %

Table(3.24): Net work of the combined cycle

Work	**(MW)**
Compressor	68.55338
Gas Turbine	132.8346
HP Steam Turbine	17.06678
LP Steam Turbine	13.44931
Pumps	0.20299
TOTAL	94.59432

Table (3.25): Global efficiency of the simulated plant

Global efficiency	
Net Work (MW)	94.59432
Heat (combustion) (MW)	296.5855
Combined Cycle efficiency	31.89 %

3.5 Simulation of Combined Cycle Power Plant with CyclePad Simulator:

3.5.1 Starting Cyclepad

When we use Cyclepad, to create a new design dialog, that tell Cyclepad the title of the design, whether it is an open (steady-state, steady-flow) cycle or a closed (control mass, control volume) cycle, and whether we want Cyclepad to calculate efficiencies for a heat engine, heat pump, or a refrigeration cycle.

Let's title our problem "combined cycle" and tell Cyclepad we are dealing with a closed cycle. Then click OK.

3.5.2 Adding Devices to the Design:

After the Open New Design dialog disappears, Cyclepad shows us the Device Palette. The Device Palette contains the following devices:

- Heater
- Cooler
- heat exchanger
- pump
- compressor
- turbine
- throttle
- splitter
- mixer
- source

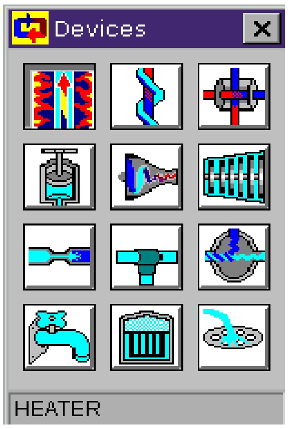


Figure (3.4): devices on Cyclepad

- reactor
- sink

To add a device to our design, a click on the device on the Device Palette is needed, and then click on the spot on the design where we want that device to be. Devices can be moved by dragging, and right-clicking on a device brings up a menu allowing us to change its orientation or delete it.

Our first task, is to add all of the devices we need to our design. Which consist of all component of combined cycle power plant such as (compressors, turbines, heaters, condenser ……. etc).After completes the addition of all necessary components to the design. We prepared the position of each component to connect all of them together correctly depend of the design of combined cycle that we need to make simulation for it.

The design below explains the fully connected build mode and components of combined cycle power plant that we make it for this research:

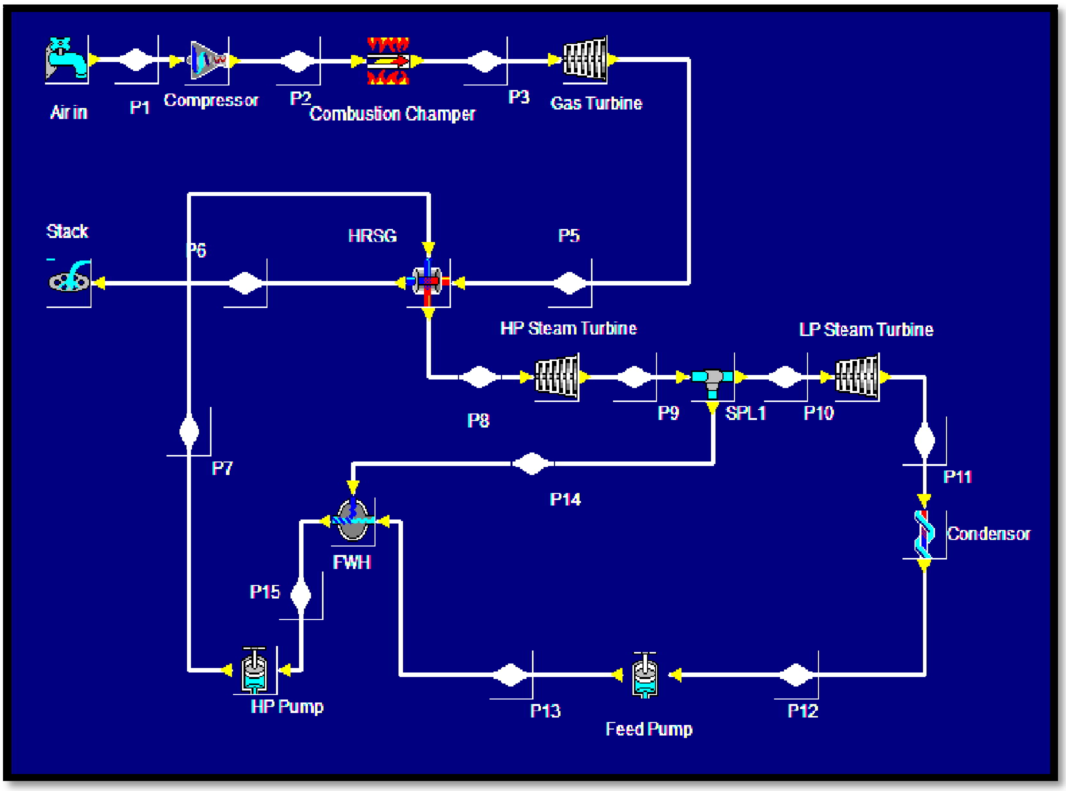


Figure (3.5): Build mode of combined cycle with cyclepad

After all inlets and outlets are connected, Cyclepad tells us that we have a complete design and asks if we wish to switch to Analyze Mode, which is what we want right now. If we wanted to stay in Build Mode, we could add and delete more components, change the label names, or move things around to suit our tastes better.

3.5.3 Analyzing Design:

When we switch to Analyze Mode, Cyclepad takes a few minutes and solve many of the equations which apply to our design. Many more equations will be added as we tell Cyclepad which assumptions apply to the various components. When the hour glass goes away, we can start making assumptions about the devices in our design and the statepoints connecting them. For this design of combined cycle, we will go around the design and add what information we know as we go, clicking on each device or statepoint to get its meter window to show up. It is particularly during this stage that our own knowledge of thermodynamics is critical to making assumptions Cyclepad will use in design solution.

The design below explains analyze mode of combined cycle power plant in cyclepad.

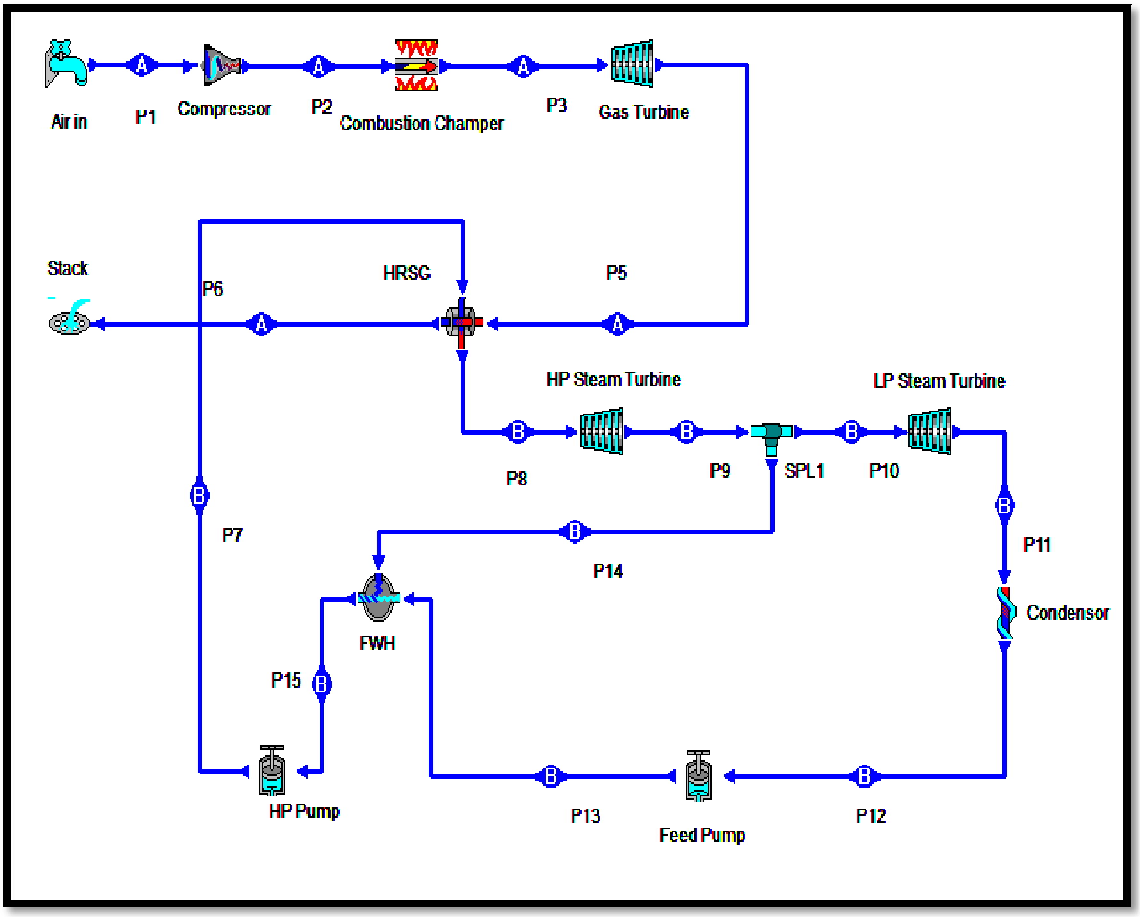


Figure (3.6): analyze mode of combined cycle with cyclepad

Entering the parameters (such as temperature, pressure, mass flow ...… etc) or material stream that known of each state point from point of start to end of cycle, The meter window shows everything about a state point or device. The Cyclepad results are shown in blue and others are shown in black.

The figures below explain the values of parameters that entered in our design for each GAS cycle and STEAM cycle

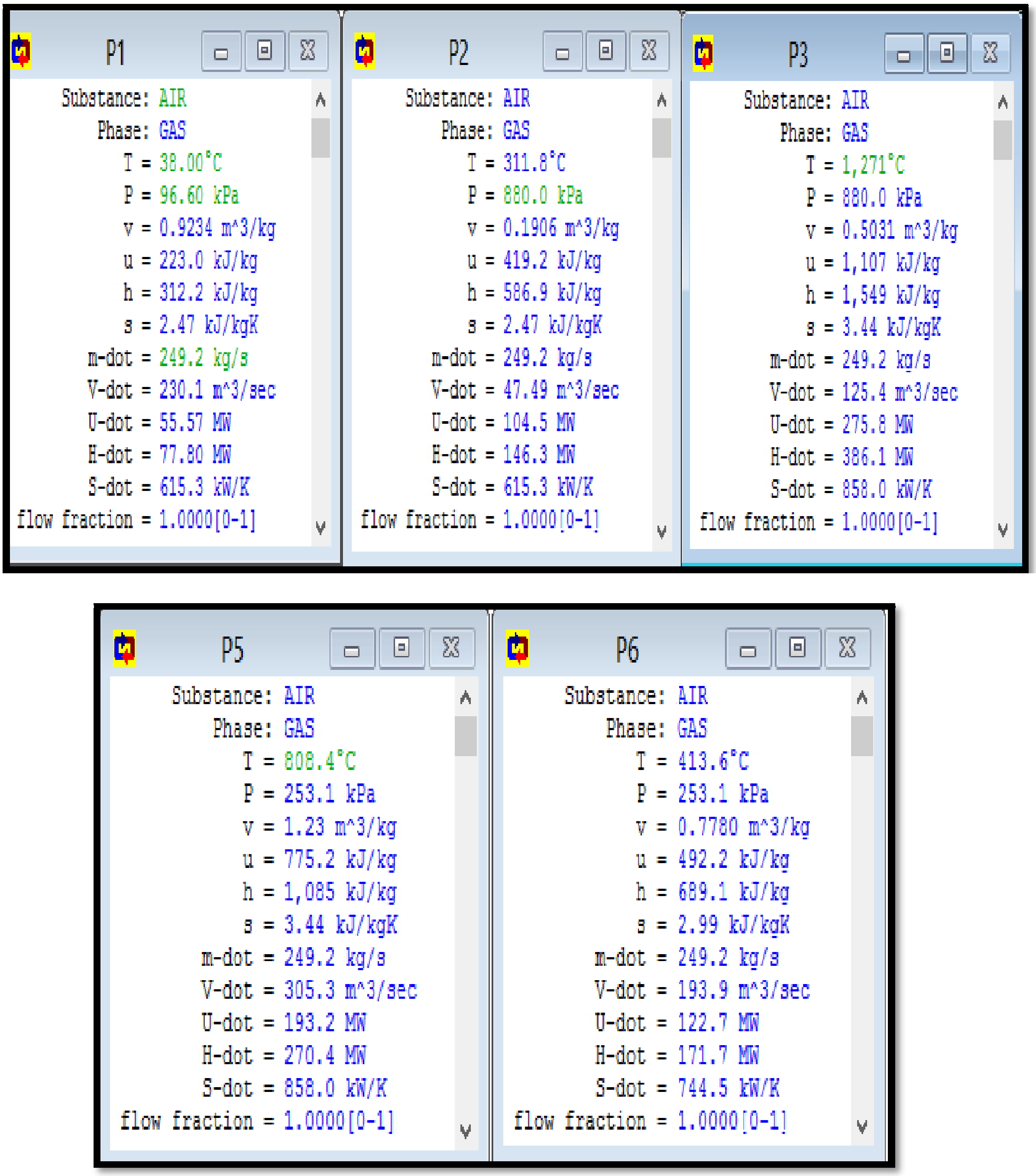
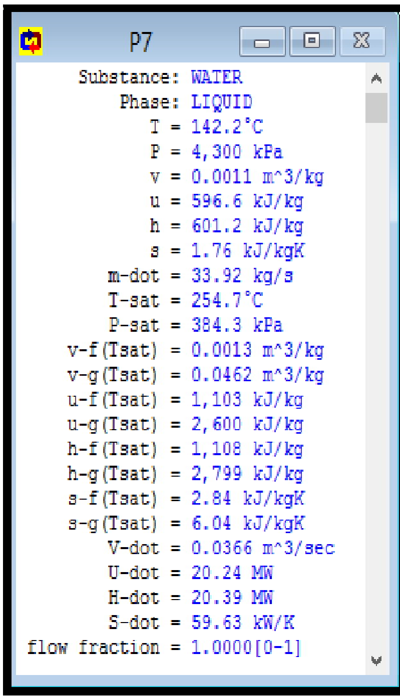
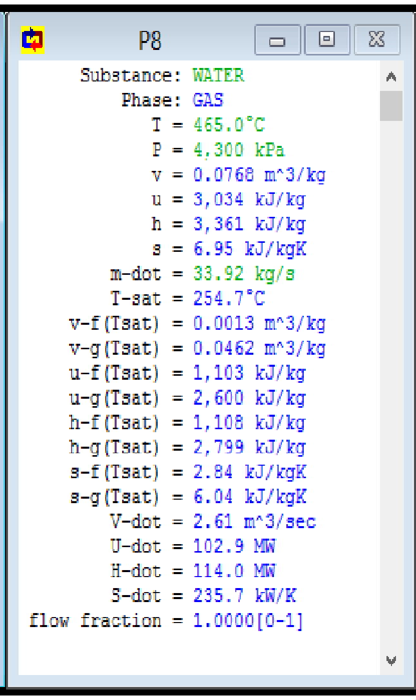
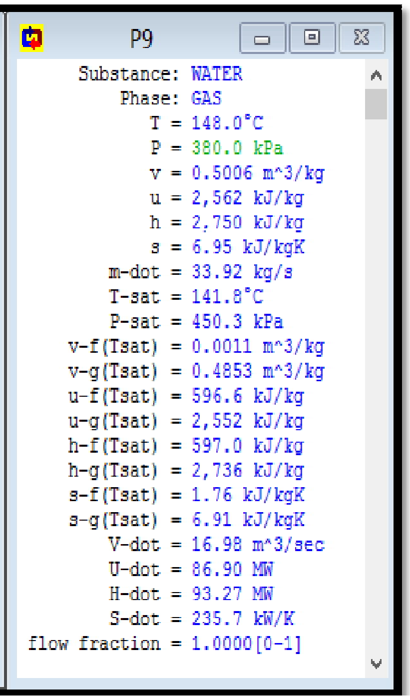
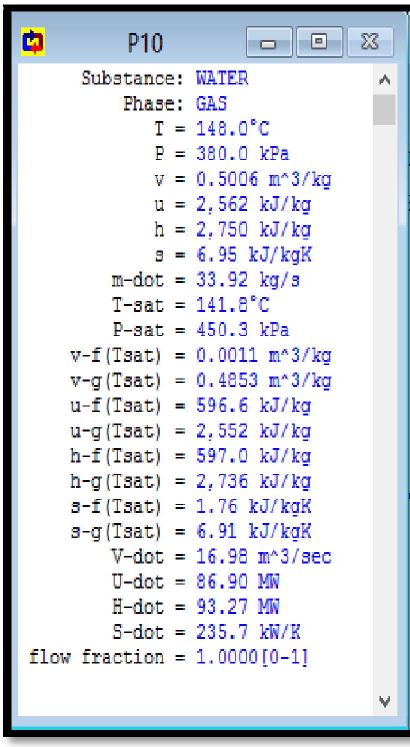
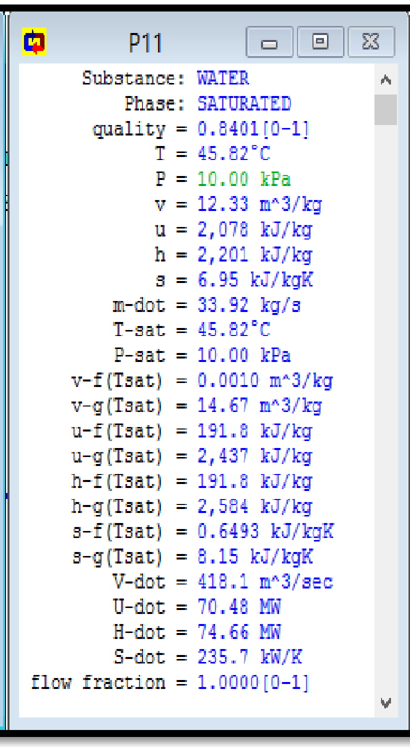
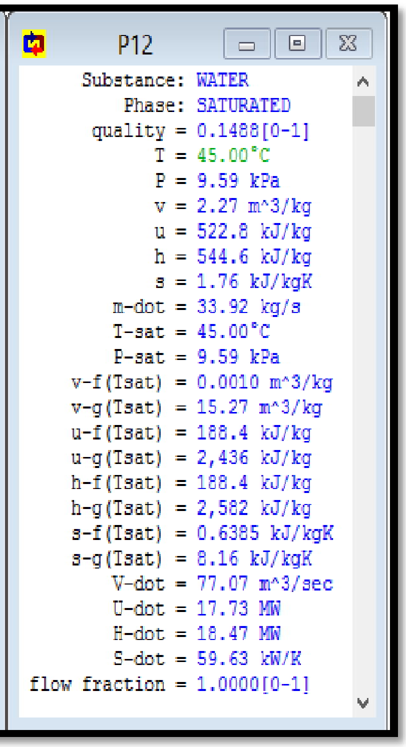


Figure (3.7): Parameters of Gas cycle


P7
Substance: WATER
Phase: LIQUID
T = 142.2°C
P = 4,300 kPa
v = 0.0011 m^3/kg
u = 596.6 kJ/kg
h = 601.2 kJ/kg
s = 1.76 kJ/kgK
m-dot = 33.92 kg/s
T-sat = 254.7°C
P-sat = 384.3 kPa
v-f(Tsat) = 0.0013 m^3/kg
v-g(Tsat) = 0.0462 m^3/kg
u-f(Tsat) = 1,103 kJ/kg
u-g(Tsat) = 2,600 kJ/kg
h-f(Tsat) = 1,108 kJ/kg
h-g(Tsat) = 2,799 kJ/kg
s-f(Tsat) = 2.84 kJ/kgK
s-g(Tsat) = 6.04 kJ/kgK
V-dot = 0.0366 m^3/sec
U-dot = 20.24 MW
H-dot = 20.39 MW
S-dot = 59.63 kW/K
flow fraction = 1.0000[0-1]

P8
Substance: WATER
Phase: GAS
T = 465.0°C
P = 4,300 kPa
v = 0.0768 m^3/kg
u = 3,034 kJ/kg
h = 3,361 kJ/kg
s = 6.95 kJ/kgK
m-dot = 33.92 kg/s
T-sat = 254.7°C
v-f(Tsat) = 0.0013 m^3/kg
v-g(Tsat) = 0.0462 m^3/kg
u-f(Tsat) = 1,103 kJ/kg
u-g(Tsat) = 2,600 kJ/kg
h-f(Tsat) = 1,108 kJ/kg
h-g(Tsat) = 2,799 kJ/kg
s-f(Tsat) = 2.84 kJ/kgK
s-g(Tsat) = 6.04 kJ/kgK
V-dot = 2.61 m^3/sec
U-dot = 102.9 MW
H-dot = 114.0 MW
S-dot = 235.7 kW/K
flow fraction = 1.0000[0-1]

P9
Substance: WATER
Phase: GAS
T = 148.0°C
P = 380.0 kPa
v = 0.5006 m^3/kg
u = 2,562 kJ/kg
h = 2,750 kJ/kg
s = 6.95 kJ/kgK
m-dot = 33.92 kg/s
T-sat = 141.8°C
P-sat = 450.3 kPa
v-f(Tsat) = 0.0011 m^3/kg
v-g(Tsat) = 0.4853 m^3/kg
u-f(Tsat) = 596.6 kJ/kg
u-g(Tsat) = 2,552 kJ/kg
h-f(Tsat) = 597.0 kJ/kg
h-g(Tsat) = 2,736 kJ/kg
s-f(Tsat) = 1.76 kJ/kgK
s-g(Tsat) = 6.91 kJ/kgK
V-dot = 16.98 m^3/sec
U-dot = 86.90 MW
H-dot = 93.27 MW
S-dot = 235.7 kW/K
flow fraction = 1.0000[0-1]

P10
Substance: WATER
Phase: GAS
T = 148.0°C
P = 380.0 kPa
v = 0.5006 m^3/kg
u = 2,562 kJ/kg
h = 2,750 kJ/kg
s = 6.95 kJ/kgK
m-dot = 33.92 kg/s
T-sat = 141.8°C
P-sat = 450.3 kPa
v-f(Tsat) = 0.0011 m^3/kg
v-g(Tsat) = 0.4853 m^3/kg
u-f(Tsat) = 596.6 kJ/kg
u-g(Tsat) = 2,552 kJ/kg
h-f(Tsat) = 597.0 kJ/kg
h-g(Tsat) = 2,736 kJ/kg
s-f(Tsat) = 1.76 kJ/kgK
s-g(Tsat) = 6.91 kJ/kgK
V-dot = 16.98 m^3/sec
U-dot = 86.90 MW
H-dot = 93.27 MW
S-dot = 235.7 kW/K
flow fraction = 1.0000[0-1]

P11
Substance: WATER
Phase: SATURATED
quality = 0.8401[0-1]
T = 45.82°C
P = 10.00 kPa
v = 12.33 m^3/kg
u = 2,078 kJ/kg
h = 2,201 kJ/kg
s = 6.95 kJ/kgK
m-dot = 33.92 kg/s
T-sat = 45.82°C
P-sat = 10.00 kPa
v-f(Tsat) = 0.0010 m^3/kg
v-g(Tsat) = 14.67 m^3/kg
u-f(Tsat) = 191.8 kJ/kg
u-g(Tsat) = 2,437 kJ/kg
h-f(Tsat) = 191.8 kJ/kg
h-g(Tsat) = 2,584 kJ/kg
s-f(Tsat) = 0.6493 kJ/kgK
s-g(Tsat) = 8.15 kJ/kgK
V-dot = 418.1 m^3/sec
U-dot = 70.48 MW
H-dot = 74.66 MW
S-dot = 235.7 kW/K
flow fraction = 1.0000[0-1]

P12
Substance: WATER
Phase: SATURATED
quality = 0.1488[0-1]
T = 45.00°C
P = 9.59 kPa
v = 2.27 m^3/kg
u = 522.8 kJ/kg
h = 544.6 kJ/kg
s = 1.76 kJ/kgK
m-dot = 33.92 kg/s
T-sat = 45.00°C
P-sat = 9.59 kPa
v-f(Tsat) = 0.0010 m^3/kg
v-g(Tsat) = 15.27 m^3/kg
u-f(Tsat) = 188.4 kJ/kg
u-g(Tsat) = 2,436 kJ/kg
h-f(Tsat) = 188.4 kJ/kg
h-g(Tsat) = 2,582 kJ/kg
s-f(Tsat) = 0.6385 kJ/kgK
s-g(Tsat) = 8.16 kJ/kgK
V-dot = 77.07 m^3/sec
U-dot = 17.73 MW
H-dot = 18.47 MW
S-dot = 59.63 kW/K
flow fraction = 1.0000[0-1]

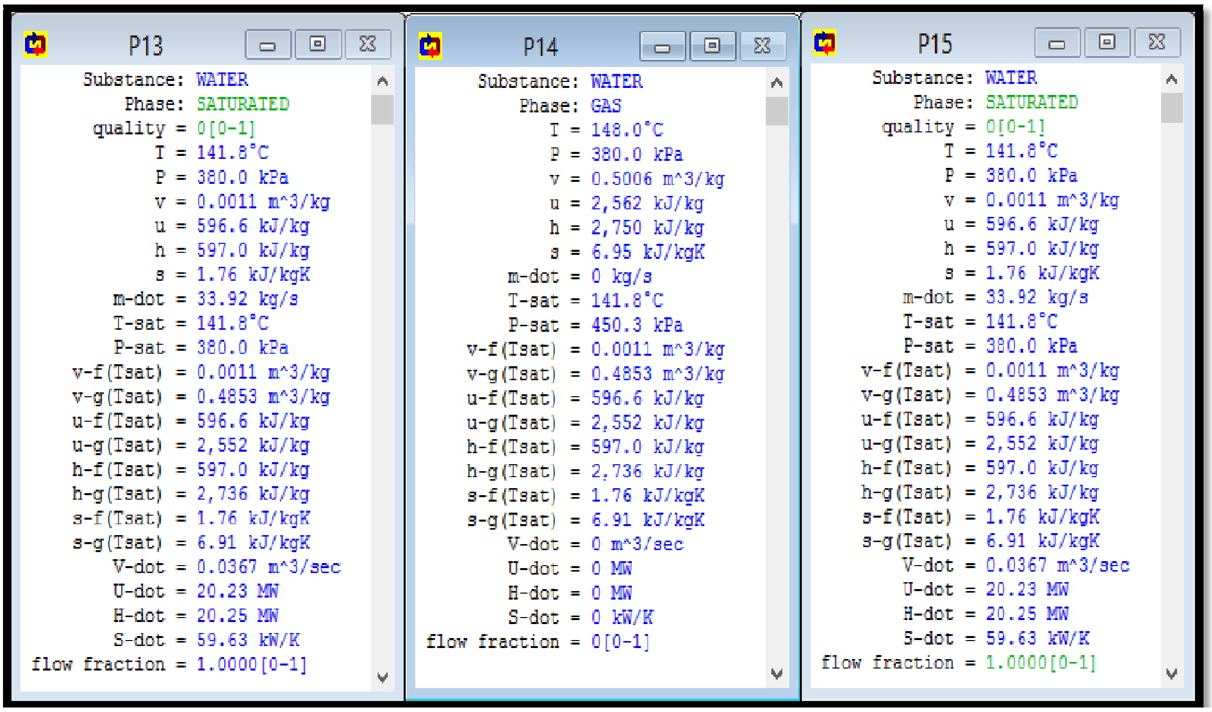


Figure (3.8): Parameters of Steam cycle

3.5.4 Result of Simulation of Combined Cycle Power Plant By Using Cyclepad Software:

The figure below explain the results of simulation of combined cycle power plant by using Cyclepad software which is consist thermal efficiency, net power out, net energy in, pumps work,……etc.

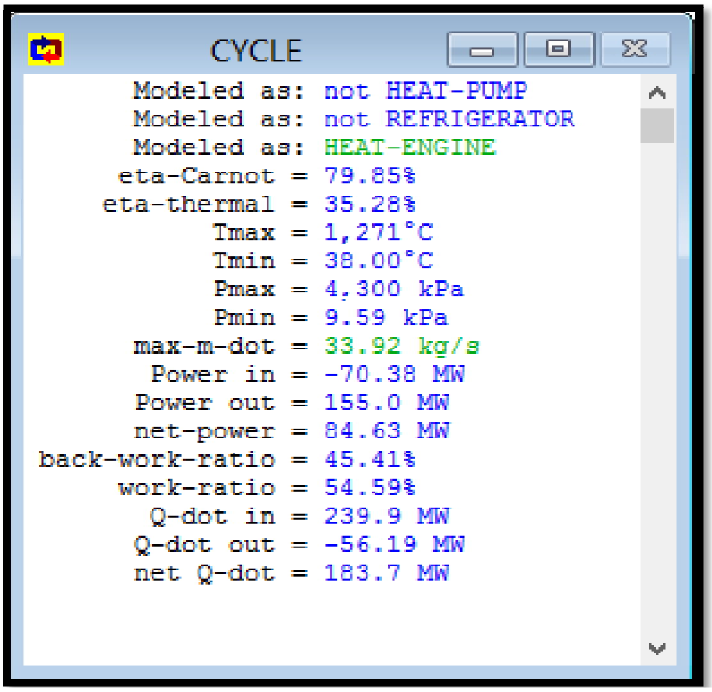


Figure (3.9): Result of Cyclepad simulation

From figure above we see all important result of cyclepad simulation but we can summarize the result in tables as shown as below:

Table (3.26): Net work of Cyclepad combined cycle simulation

Works	**(MW)**
Compressor	68.46
Gas Turbine	115.7
HP Steam Turbine	20.73
LP Steam Turbine	18.61
Pumps	0.1462
TOTAL	84.63

Table (3.27): Global efficiency of the simulated plant by Cyclepad

Global efficiency	
Net Work (MW)	84.63
Net Heat (combustion) (MW)	239.9
Combined Cycle th-efficiency	35.28%

3.6 Comparison Results of the Two Simulated Plants:

Table (3.28) represents comparison between HYSYS simulation analysis, Cyclepad analysis and the real data of GARRI power plant.

Table (3.28): comparison between simulation results and GARRI data

	GARRI plant	**HYSYS simulated plant**	**Cyclepad simulated plant**
Net Work (MW)	58.220	94.59432	84.63
Heat (combustion) (MW)	212.482	296.5855	239.9
Combined Cycle efficiency	27.4 %	31.89 %	35.28%

From table above found that HYSYS simulated plant efficiency is near to the actual efficiency of GARRI plant more than that efficiency which calculated on Cyclepad simulated plant. So we arrive to that HYSYS simulation software is better than cyclepad simulation software cause its results and efficiency is nearly to that results and efficiency of real plant we make simulation of it.

So we will use HYSYS simulator for our next study which is optimization of combined cycle by making different scenario to calculate the optimum value of parameters that give maximum efficiency of combined cycle power plant.

3.7 Matlab Code for combined Cycle:

3.7.1 Checking result of Simulations by Matlab Code:

The parameter which affects the performances of Garri combined power cycle plant is the ambient temperature due to variation of the climate that varies from 15°C to 48°C .The study is to find the optimal inlet temperature of the air to compressor to give maximum efficiency by using Matlab code. Program flow chart in figure (3.19) and the M-file program Appendix (N)

3.7.2 Mat-lab program input:

Ambient temperature (compressor inlet temperature), air & fuel mass flow rate, isentropic efficiency of compressor and turbine, mechanical efficiency of compressor, turbine inlet temperature, the specific heat ratio, compressor ratio, gas turbine inlet temperature, power output of steam turbine ,heat caloric value.

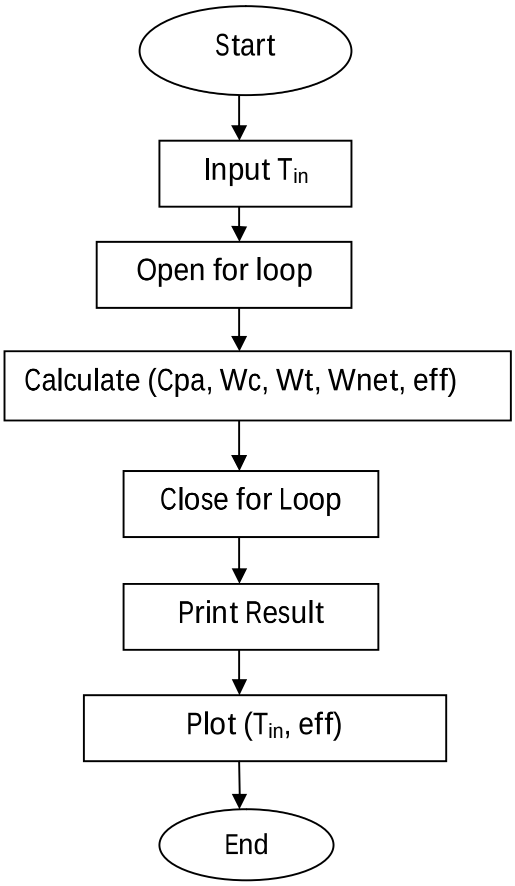


Figure (3.10): Mat-lab program flow chart

3.7.3Mat-lab program Output:

The results and the output of the Mat-lab program are in the table (3.29) &figure (3.11)

Table (3.29): Matlab Program output

T_{in}	15°c	20°c	25°c	30°c	35°c	38°c	40°c	45°c	48°c
Comp out temp(°K)	457.23	465.17	473.11	481.05	488.99	493.75	496.92	504.86	509.62
C_{pa} (kJ/kg °K)	1.0032	1.0035	1.0037	1.0040	1.0043	1.0045	1.0046	1.0049	1.0051
Wc (MW)	42.27	43.02	43.76	44.51	45.26	45.71	46.01	46.76	47.21
Wnet (MW)	66.63	65.89	65.15	64.40	63.65	63.20	62.90	62.15	61.70
Efficiency (%)	31.42	31.07	30.72	30.37	30.01	29.80	29.66	29.31	29.09

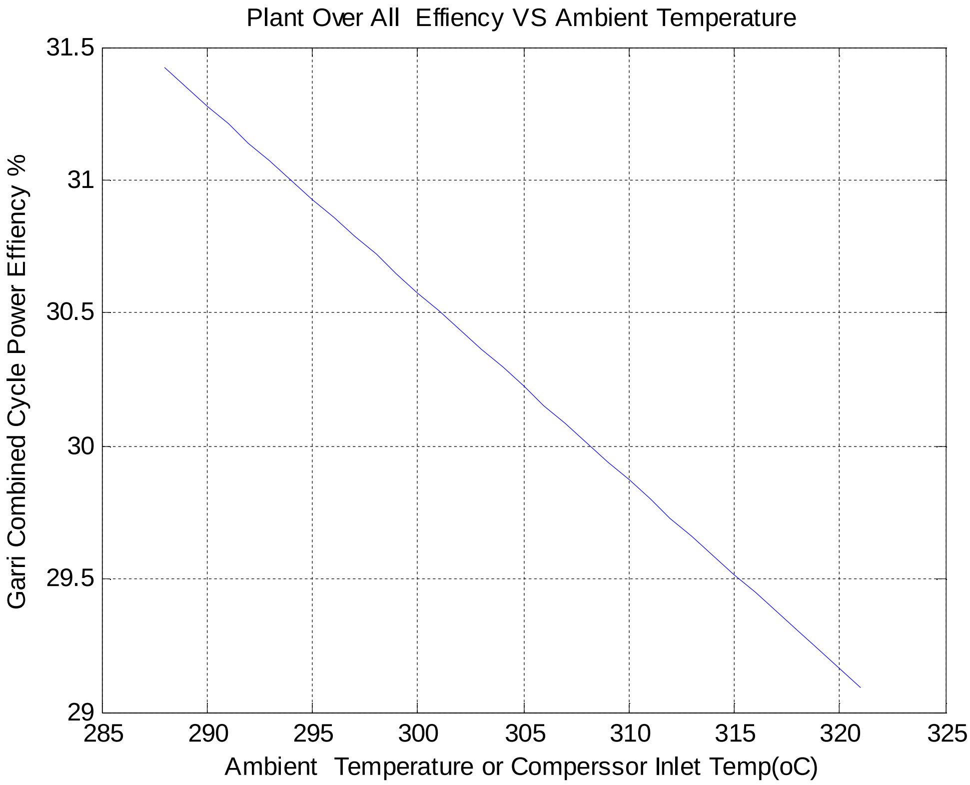


Figure (3.11): results and the output of the Mat-lab program

3.8 Economic analyses for the study:

According to the Generation and type of fuel used at each station, analysis of different generation power plants price of kwhr, experiments shows that the cost of operation and maintenance in most types of thermal and hydraulic power stations close to the cost of fuel that effect on the electrical system generation.

In this study the cost of generating the various and using different types of fuel from crude oil, coal, and natural gas according to Appendix (O) the economic gain of the study in overall electricity generation in Sudan due to Garri combined cycle efficiency improvement as in table (3.30).

Table (3.30) economic analyses according to plant overall efficiency variation

Type of Efficiency result	Efficiency value%	S.F.C (Kg/kwh)	Annular fuel cost$	Fuel only (SDG/kwh)	Price (SDG/kwh)
Actual plant (reference)	27.40	0.328	434.4	1.215	0.3428
ASPENHYSYS simulation	31.89	0.282	373.2	1.044	0.3113
CYCLEPADE simulation	35.33	0.251	331.6	0.928	0.2898
MATLAB CODE	31.42	0.286	378.8	1.060	0.3141
Optimum	33.88	0.266	351.3	0.983	0.3000

CHAPTER IV

RESULT & DISCUSSIONS

CHAPTER IV

Result & Discussions

4.1 Optimization 0f Combined Cycle Power Plant:

Thermodynamic analysis and optimization of combined cycle power plant depending on the operating parameters on the combined cycle with different losses which occur in different components of plant are.:

Operating parameters which influence the combined cycle performance are;

1) Air inlet temperature (ambient temperature)
2) Air mass flow rate
3) Fuel mass flow rate
4) Air/fuel ratio
5) Compressor pressure ratio
6) Gas turbine inlet temperature
7) Live steam pressure
8) Live steam temperature
9) Condenser pressure
10) Mass flow rate of steam
11) Extraction mass flow rate
12) Pinch point temperature difference

Using HYSYS software with Microsoft excel the effect of each parameters are presented in the following figures

4.2 Effect of different parameters on plant thermal efficiency:

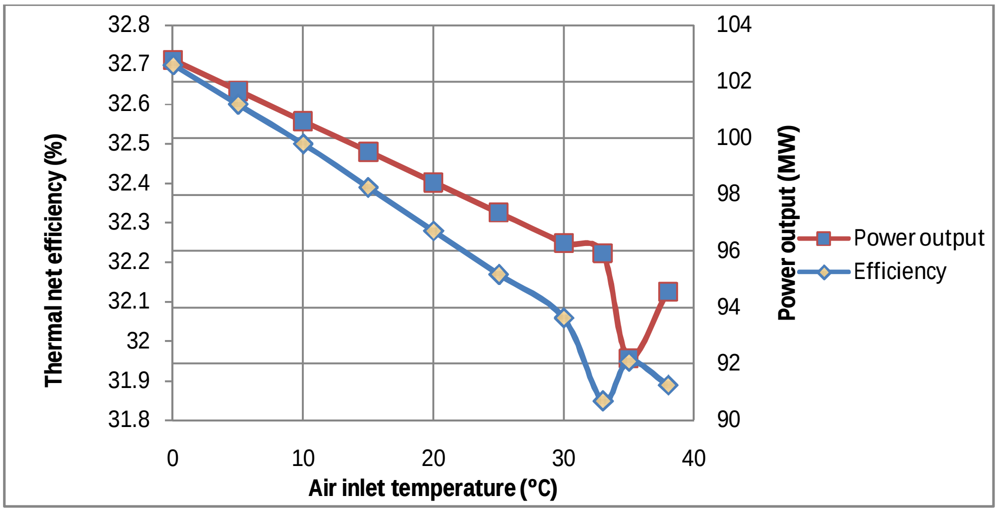


Figure (4.1): Effect of Air inlet temperature on plant efficiency and output power

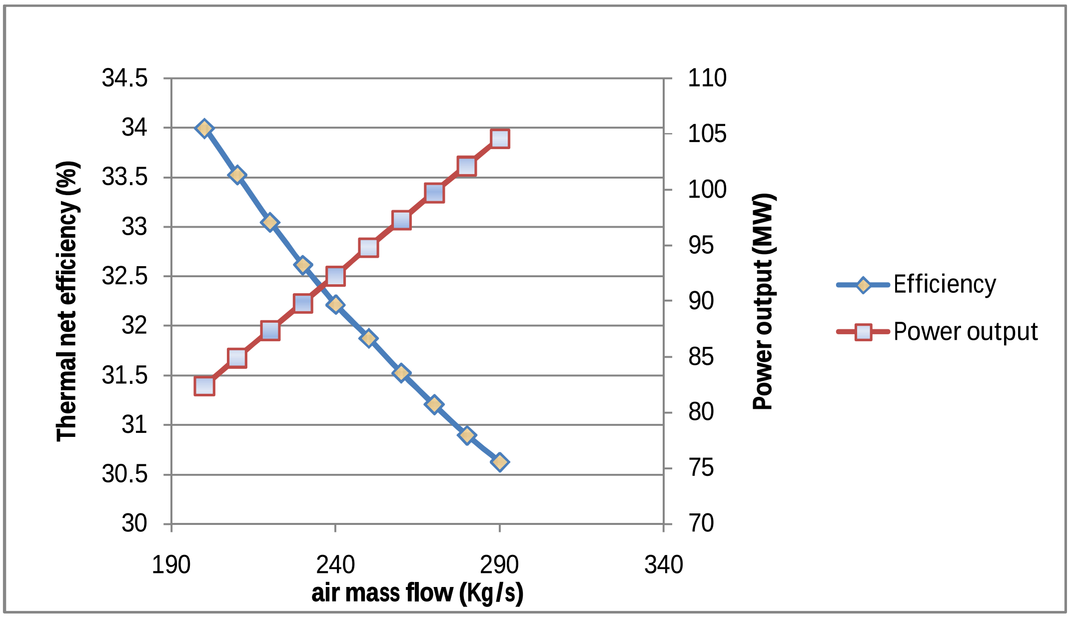


Figure (4.2): Effect of air mass flow on plant efficiency and output power

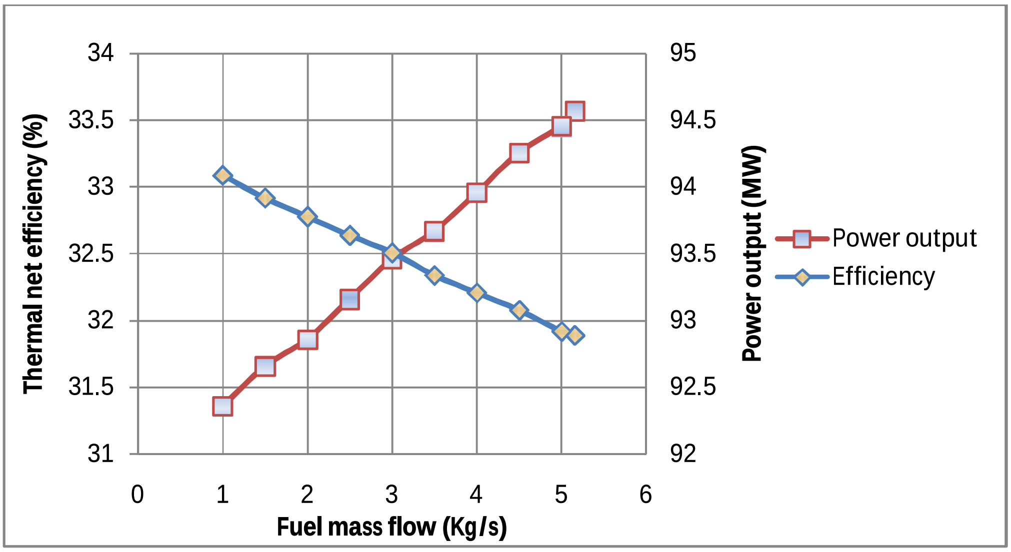


Figure (4.3): Effect of Fuel mass flow on plant efficiency and output power

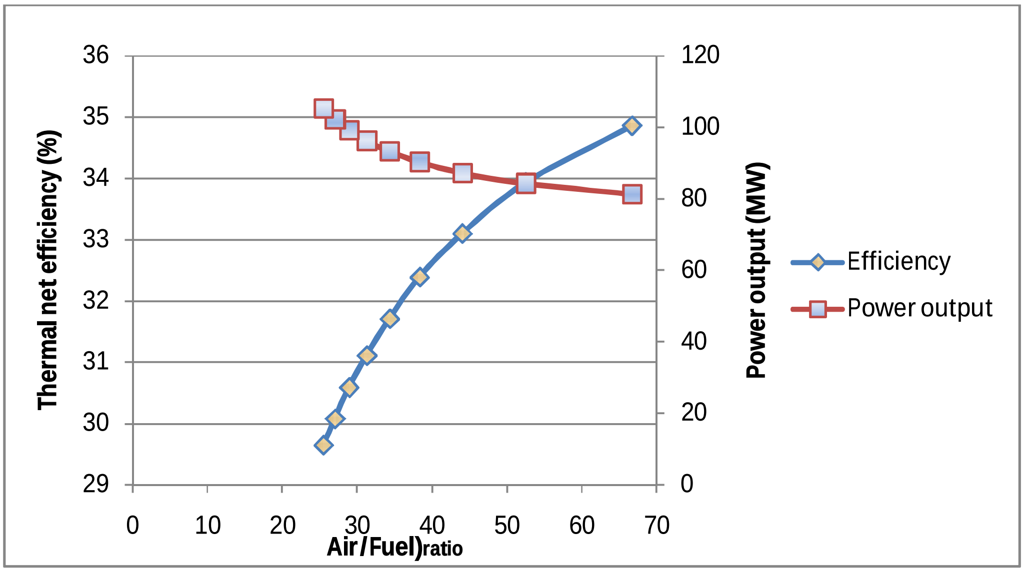


Figure (4.4): Effect of Air/ Fuel ratio on plant efficiency and output power

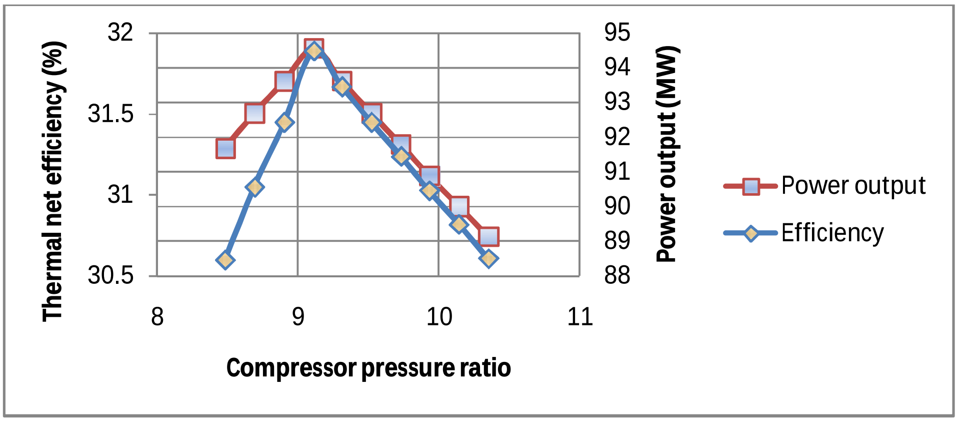


Figure (4.5): Effect of compressor pressure ratio on plant efficiency and output power

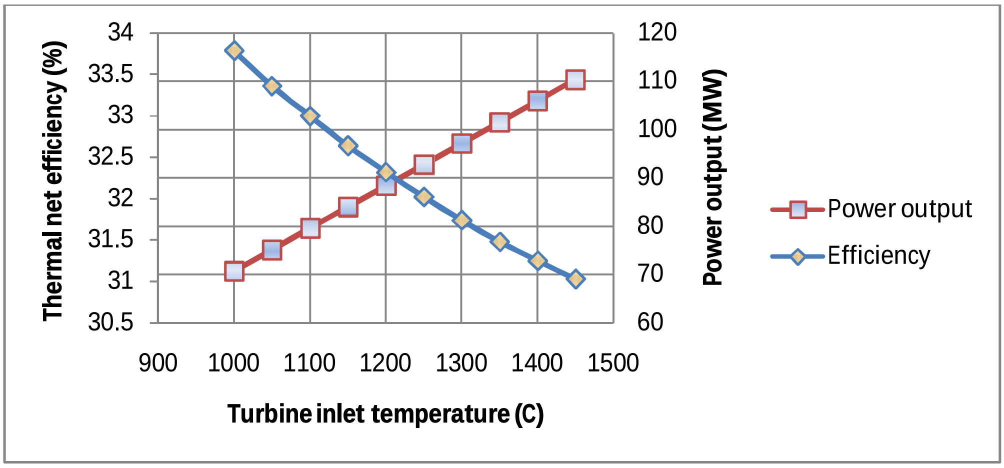


Figure (4.6): Effect of turbine inlet temperature on plant efficiency and output power

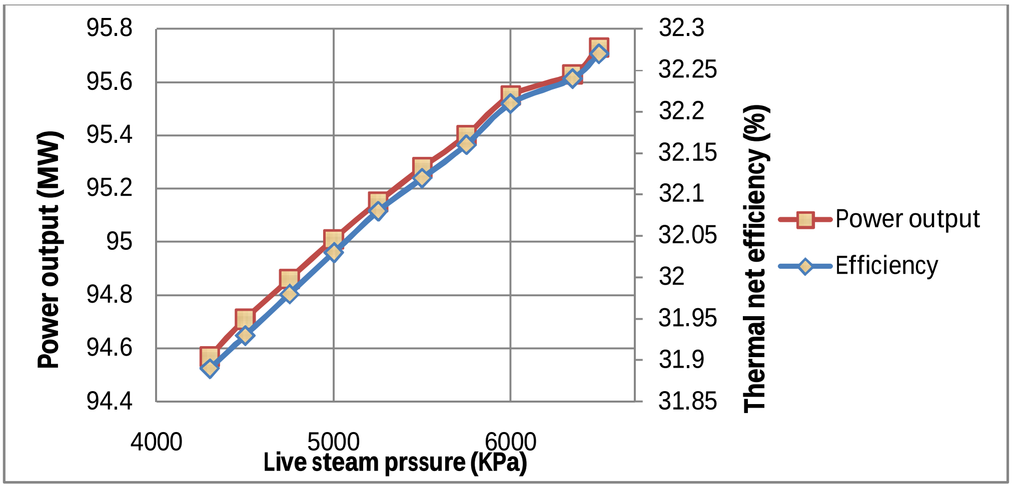


Figure (4.7): Effect of Live steam pressure on plant efficiency and output power

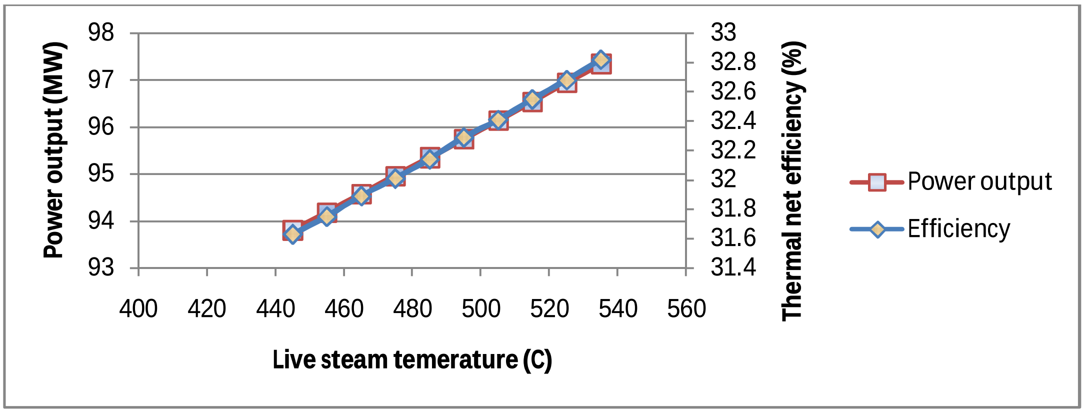


Figure (4.8): Effect of Live steam temperature on plant efficiency and output power

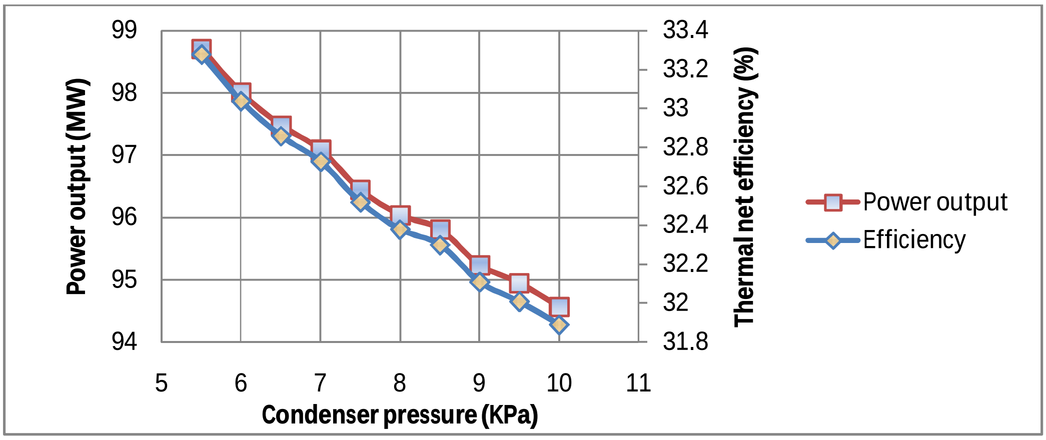


Figure (4.9): Effect of Condenser pressure on plant efficiency and output power

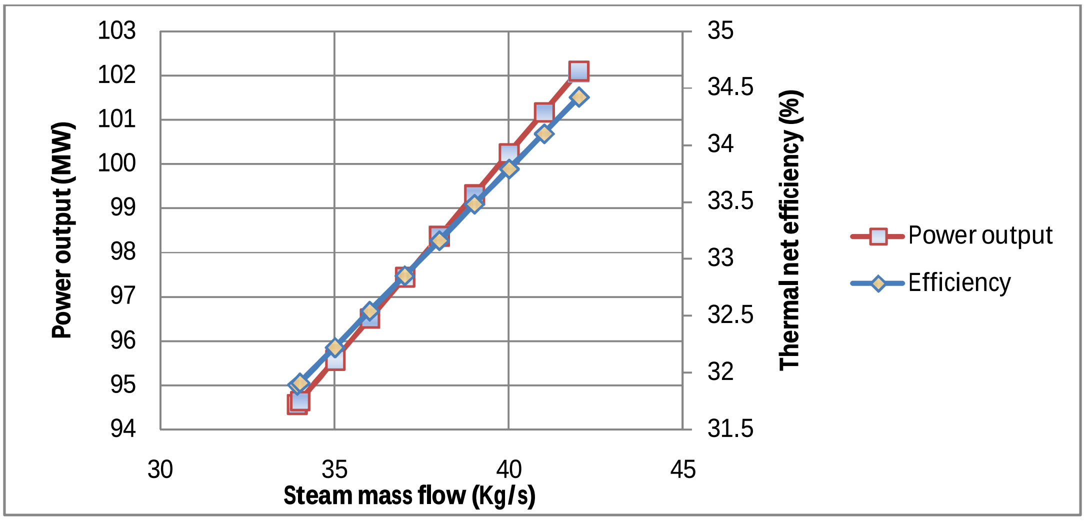


Figure (4.10): Effect of Steam mass flow on plant efficiency and output power

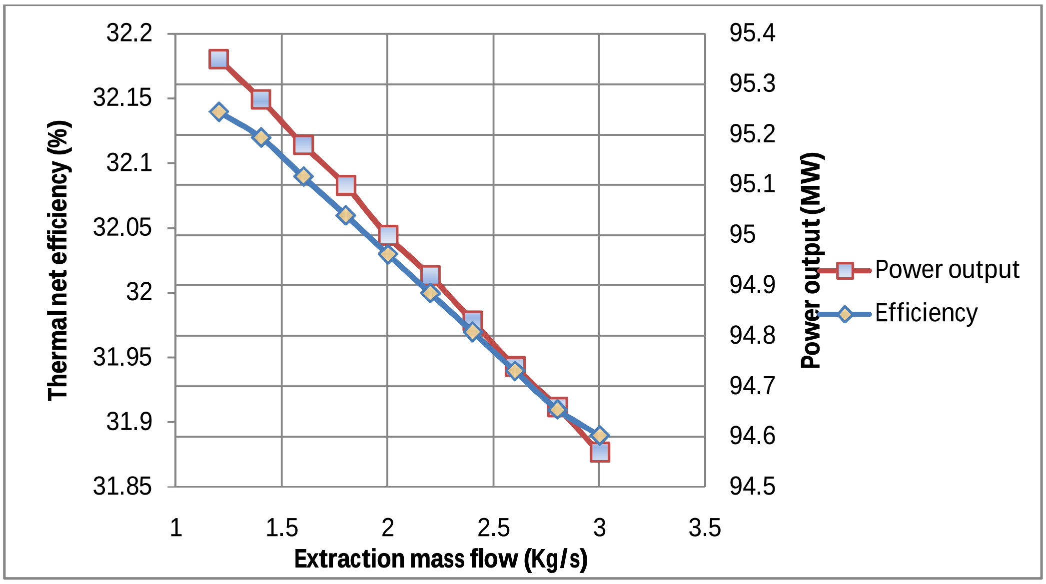
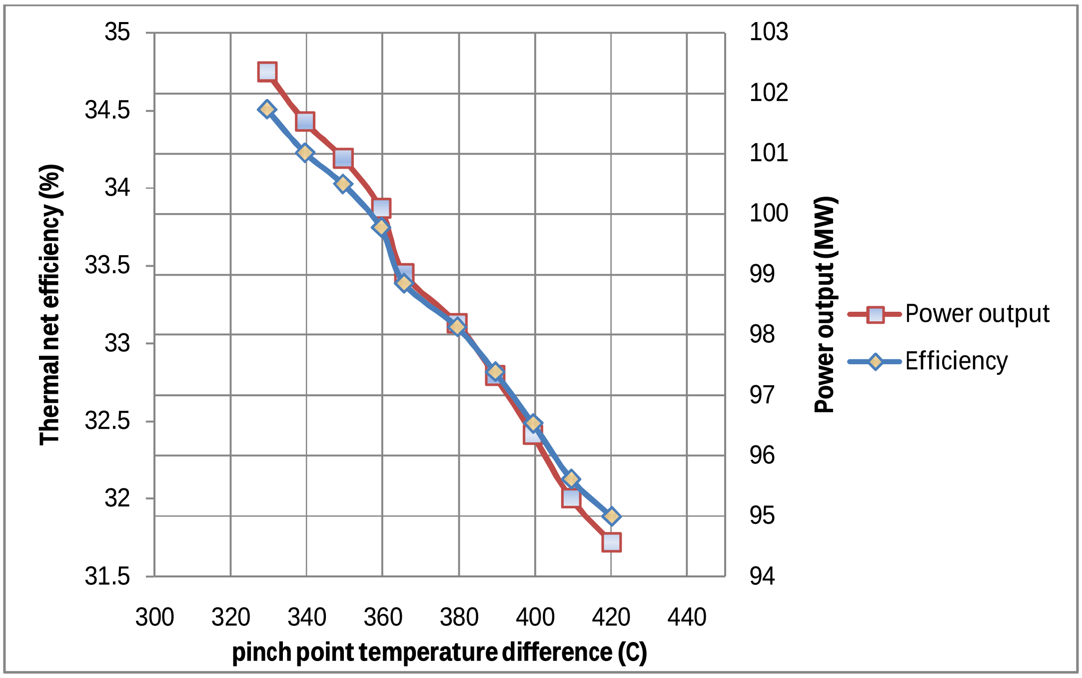


Figure (4.11): Effect of Extraction steam mass flow on plant efficiency and output power



(4.12): Effect of pinch point temperature difference on efficiency and output power

For optimization we make different scenario of working plant that we change in different parameters and focus on its effect on efficiency. Different scenarios represented in table that shown below.

4.3 Result of Optimization:

The figure below explains the curve between scenarios and efficiency which explain the maximum efficiency is equal 33.88%.

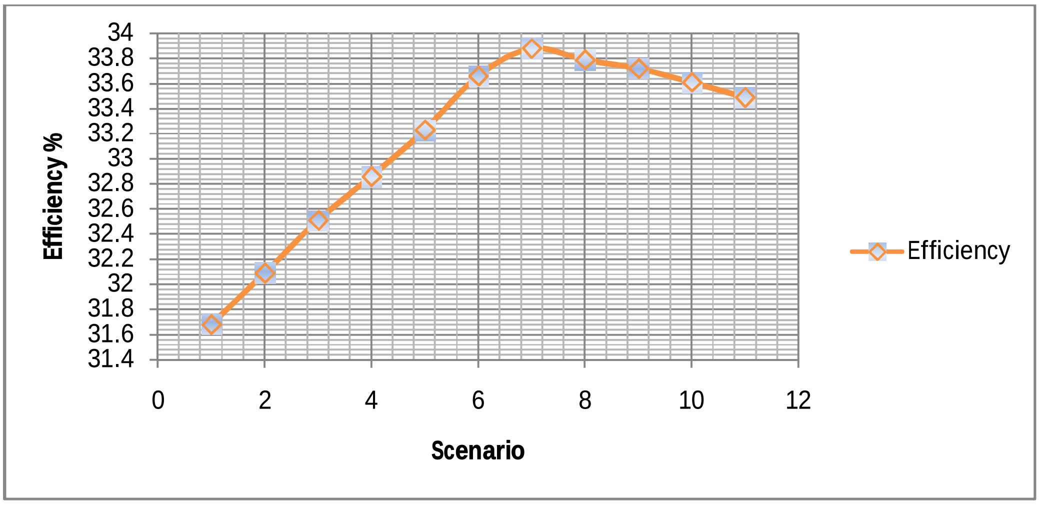


Figure (4.13): efficiency curve with numbers of scenario

Table (4.1): Comparison of optimized solution to the study

Parameters	Project work	Optimized solution	Absolute difference	Relative difference
Air inlet temperature) (C)	38	28	-10	-26.31%
Fuel mass flow rate (Kg/s)	5.16	5.5	+0.35	+6.78%
Air mass flow rate (Kg/s)	249.2	260	+10.8	+4.33%
Gas turbine inlet temperature (C)	1271	1300	+29	+2.28%
steam pressure (KPa)	4300	5500	+1200	+27.9%
steam temperature (C)	465	520	+55	+11.83%
Condenser pressure (KPa)	10	11	+1	+0.1%
Mass flow rate of steam (Kg/s)	33.917	32	-1.917	-5.65%
Extraction mass flow rate (Kg/s)	3	2.8	-0.2	-6.66%
Compressor pressure ratio	9.11	9.52	+0.41	+4.5%

The table above explains the optimum values of parameters which give maximum efficiency equal **33.88%.**

Reference to the results the Parameters that have a significant impact on the overall efficiency and the power output of the plant and it has and economic effect on the results are the Fuel mass flow rate and compressor pressure ratio, the study verified and choose the best values to give optimum efficiency and power output under a minimum cost.

4.4 Economic Analyses &Discussion:

For economic analyses assume that:

1 Ton = 600SDG

1Kw = 2SDG

Fuel required (ton/MWhr) = fuel consumption / Gross generation

The money required for generation SDG/ Kwhr= Fuel required*Price*1000

The results of economic for energy are as table (4.2):

Table (4.2): economic analysis for the study result

Parameters	Actual plant	Study result	Different between
Gross generation(MW)	95.120	117.641	22.521
Fuel consumption(Ton/hr)	392.400	418.256	25.856
Ton/MWhr	4.125	3.555	-.57
SDG/Kwhr	8250.0	7110.00	1140
Fuel cost/hr	235440	250955	15515
Electric generation payback/hr	190240	235282	45042

From the table (4.2) we find that:

*the fuel required for MWhr in the study equals to 3.555(ton/MWhr) and that reguired for the actual plant is (3.125Ton/MWhr) that means the study save 0.57 ton of fuel for every 1 MWhr.

*Also we find in the study the payback of energy more than the actual plant by 45042 SDG.

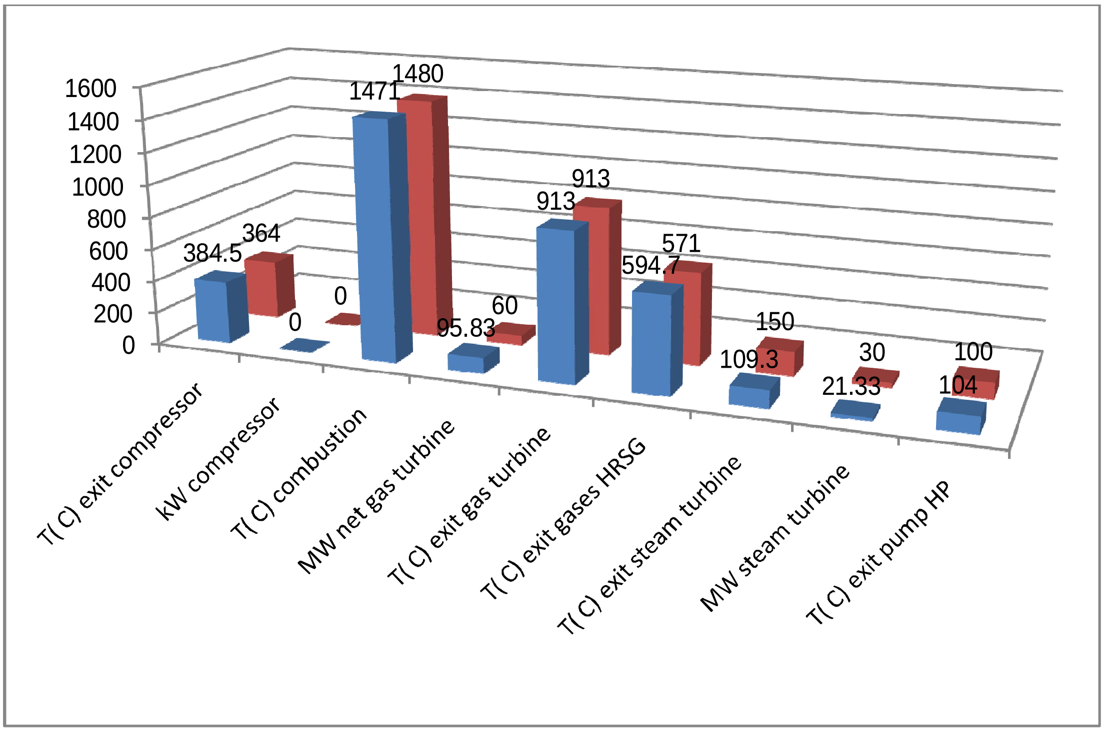


Figure (4.14): different between actual and optimum parameters

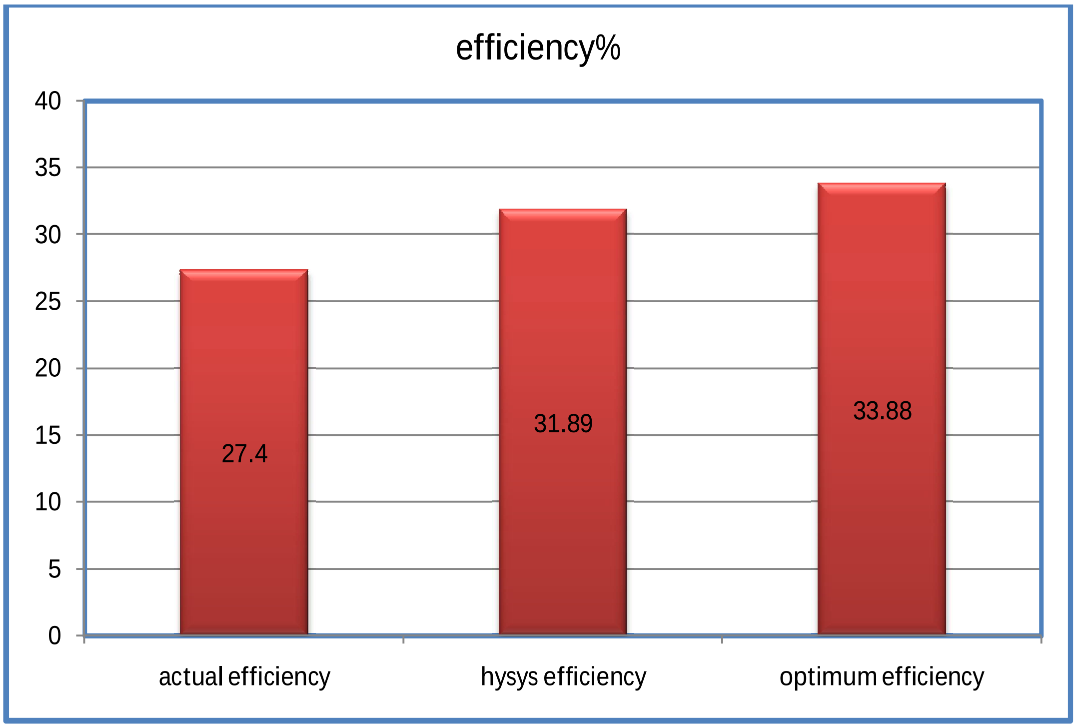


Figure (4.15): Thermal Efficiency (actual, Hysys and Optimum)

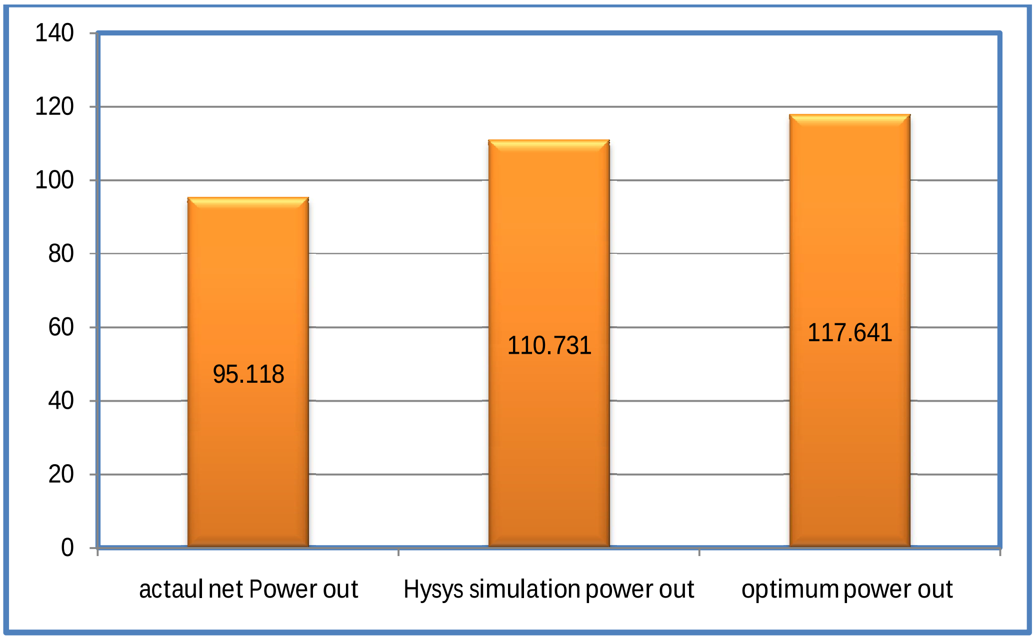


Figure (4.16): net power output (MW) (actual, Hysys and Optimum)

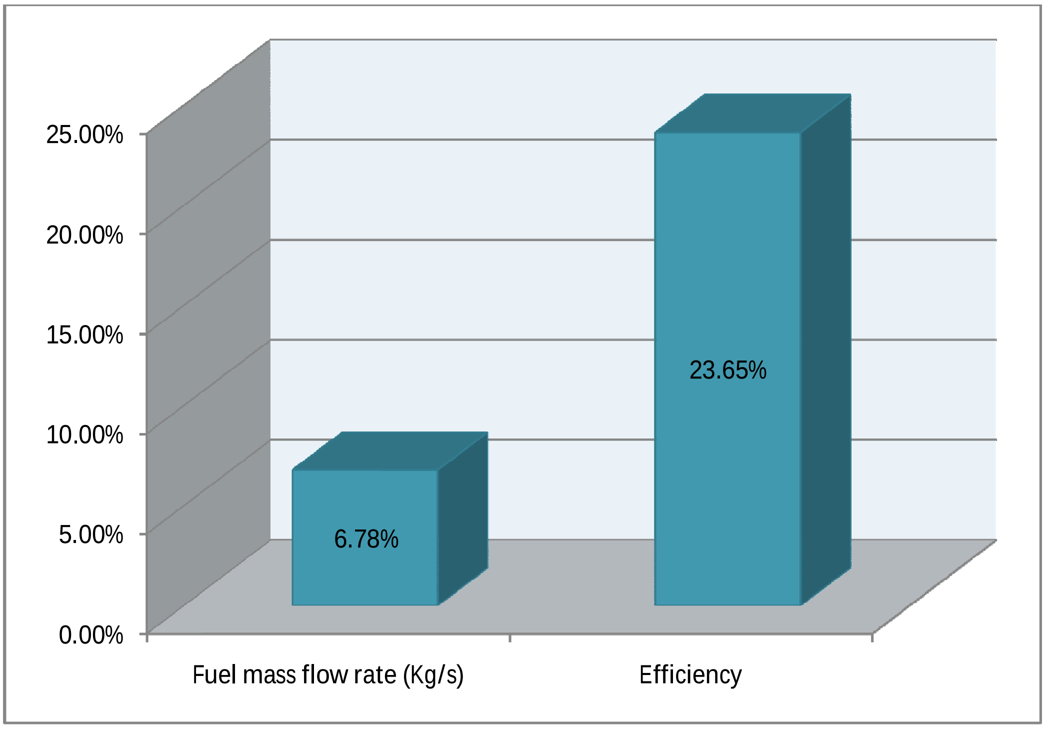


Figure (4.17): percentage increase in actual & optimum value

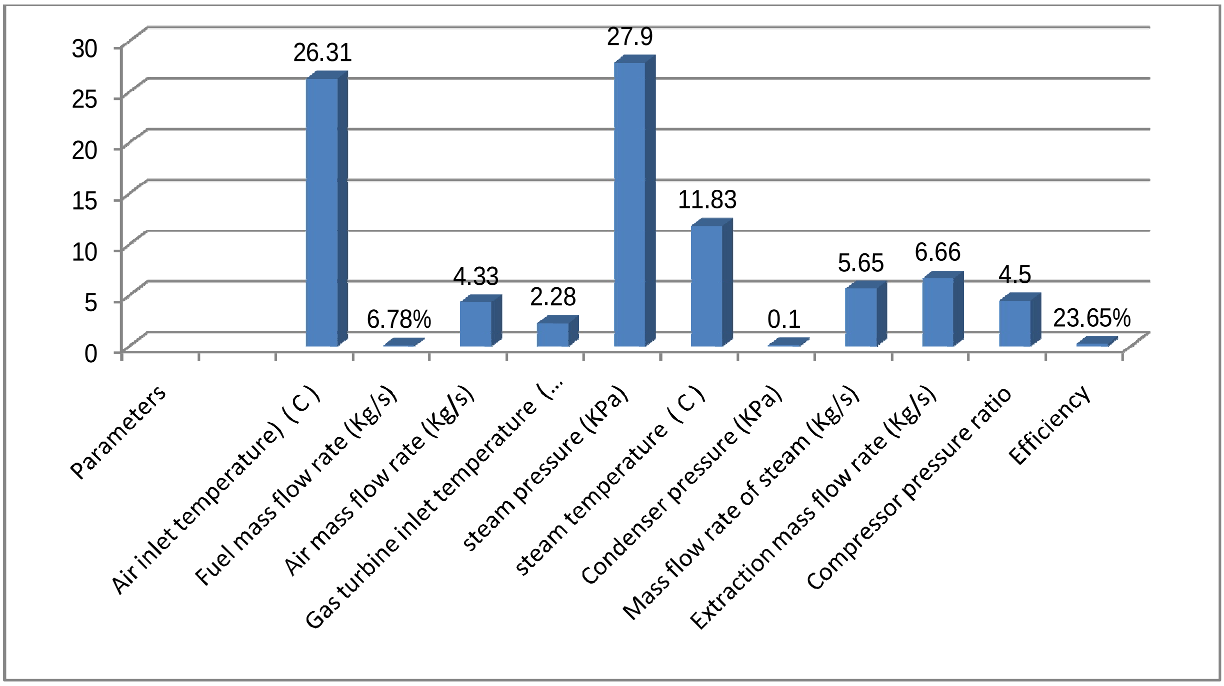


Figure (4.18): percentage increase between actual and optimum values of the parameters

CHAPTER IV

CONCLUSION & RECOMMENDATIONS

CHAPTER IV

Conclusion & Recommendations

5.1Conclusion:

A simulation of the operating system in GARRI1 combined cycle station was done by using different simulators which are ASPEN HYSYS through a numerical analysis to obtain the closest design to GARRI1 combined cycle station, that show the simulation, is the closest to the actual efficiency for GARRI1 combined cycle power plant. The comparison between the designed cycles based on the thermal net efficiency produced, the thermal net efficiencies calculated were:

- GARRI (1) combined cycle station net efficiency is **27.4%.**
- The efficiency obtained from ASPEN HYSYS simulator is **31.89%.**

The effects of major operating parameters can be summarized as follows:

1. The decrease in air inlet temperature (ambient temperature) will make an increase in efficiency and power output.
2. The decrease in fuel mass flow rate will make a decrease in efficiency and increase in power output.
3. The compressor pressure ratio should be optimum for maximum performance of combined cycle.
4. The turbine inlet temperature should be kept on higher side for maximizing power output, but in other side it minimizes the thermal net efficiency.
5. The increase in live steam pressure will make an increase in efficiency and power output.
6. The increase in live steam temperature will make an increase in efficiency and power output.

7. The decrease in condenser pressure will make an increase in efficiency and power output.
8. The increase in steam mass flow will make an increase in efficiency and power output.
9. The decrease in extraction steam mass flow will make an increase in efficiency and power output.
10. The decrease in pinch point temperature improves the combined cycle performance by increasing the efficiency and power output.

The maximum efficiency of GARRI (1) was calculated. And by changing those operating parameters, the efficiencies through assuming different scenario's under different operating parameters was calculated. As maximum efficiency equals **33.88%**.

From calculating the maximum efficiency, the optimum operating parameters were derived, which are:

Table (5.1): optimum operating parameters

Parameter	**value**
Air inlet temperature (ambient temperature)	28 ºC
Mass flow rate of fuel (LPG)	5.5 Kg/s.
Air mass flow rate	260 Kg/s
Compressor pressure is	920 KPa
Turbine inlet temperature	1300 ºC
Live steam pressure	5500 KPa
Live steam temperature	520º C
Mass flow rate of steam	32 Kg/s
Extraction mass flow rate	2.8 Kg/s
Condenser pressure	11 KPa

5.2 Recommendation:

1. Designing simulation software by using a specific programming language for studying thermal power plants.
2. Applying simulation by using MATLAB program for its precise numerical analysis.
3. Applying different operation research methods for calculating the optimum operating parameters due to its accuracy in extracting the optimum values for the operating parameters which leads to the highest efficiency.
4. Extending the research domain to include all GARRI (1) combined cycle power plant blocks.
5. Possibility of redesigning GARRI (1) combined cycle power plant to appropriate the different results of this study to increase its efficiency.

CHAPTER VI

REFERENCES & APPENDICES

CHAPTER VI

References & Appendices

References:

1. Standard handbook of powerplant engineering / [edited by] Thomas C. Elliott, Kao Chen, Robert C. Swanekamp (1998).
2. Sanjay Kumar and Onkar Single, Department of Mechanical Engineering, Harcourt Butler Technological Institute, Kanpuer-208002, U.P., INDIA Performance Evaluation of Gas-steam Combined cycle Having Transpiration Cooled Gas Turbine 2013.
3. Mc Govern, J., Cullen, B., Feidt, M., Petrescu, S. (2010). Validation of a simulation model for a combined Otto and Stirling Cycle power plant. ES2010: ASME 4th International Conference on Energy Sustainability. Phoenix, Arizona, USA. May 17-22.
4. Eleni T. Bonataki_ and K.C. Giannakoglou (2005), Preliminary Design Of Optimal Combined Cycle Power Plants Through Evolutionary algorithms, Mechanical Engineer, PhD, Public Power Corporation, Thermal Projects Engineering and Construction Department, Aristotelous 30-32, Athens 104 33, GREECE.
5. Boonnasaa S, Namprakaia P, Muangnapoh T (2006). Performance improvement of the combined cycle power plant by intake air cooling using an absorption chiller. Ener., 31: 2036-2046.
6. Bonataki E.T. and K.C. Giannakoglou, An Automated Tool for Single– and Multi–Objective Optimization for Redesigning Combined Cycle Gas Turbine Power Plants, Proceedings of the Fourth GRACM Conference on Computational Mechanics, Patras,Greece, June 2002.
7. Bonataki, E.T. A.P. Giotis and K.C. Giannakoglou, Multi–Objective Design of Optimal Combined Cycle Gas Turbine Power Plants with Supplementary Firing, EUROGEN 2003, Barcelona, Spain, 2005.
8. Bonataki E.T., L.S. Georgoulis, H. Georgopoulou and K.C. Giannakoglou, Optimal Design of Combined Cycle Gas Turbine Power Plants based on Gas Turbine

Performance Data,ERCOFTAC Design Optimization: Methods & Applications, Athens,Greece, March 2004.

9. Yokoyada. R., S. Takeuchi, K. Ito, “Thermoeconomic Analysis and Optimization of a Gas Turbine Cogeneration Unit by a Systems Approach”, Proceedings of ASME Turbo Expo 2005, Power of Land, Sea, and Air, June 6-9, 2005, Reno, Nevada, USA.
10. Mohagheghi M, Shayegan J. Thermodynamic optimization of design variables and heat exchangers layout in HRSGs for CCGT, using genetic algorithm. Appl Therm Eng 2009.
11. The third paper Scientific Conference - Energy and Development - the electric power crisis: causes and solutions (D. Yassin Haj Abdin, Director General of the National Electricity Corporation in April 1997 m).
12. P-K Raja power plant engineering published by New Age International (p) Ltd, 2006.
13. Sudan EIGAILI combined cycle power station project technical specification, by manufacturing company HARBIN power engineering company 18-02-2001
14. Aspen hysys plus simulation program.
15. Cyclepad Simulstion Program.
16. Solar thermal power 2020 exploiting the heat from the sun to combat climate change, published by European Solar Thermal Power Industry Association (ESTIA).
17. Auxiliary Steam-Plant Equipment, Engineering Liberary@McGraw-Hill
18. Downloaded from Digital Engineering Library @McGraw-Hill (www.digitalengineeringlibrary.com)
19. copyright©2004 The McGraw-Hill Companies. All rights reserved.
20. Any use is subject to the Terms of Use as given at website.
21. Gas Turbine and Optimization of Combined Cycle Power Plant Prof. Mohamed Abdelhalim.
22. http://www.brighthubengineering.com/power-plant/42019-what-are-the-types-of-hrsgs/
23. http://steamofboiler.blogspot.com/2011/08/combined-cycle-power-plant.html
24. http://en.wikipedia.org/wiki/heat_recovery_steam_generator#mediaviewer/File:HRSGPoster.jpg
25. http://www.powertechnology.com/contractors/boilers/aalborg/aalborg3.html.
26. http://www.victoryenergy.com/ImgUL/images/HRSG_FLOW-DIAGRAM.jpg.
27. http://www.victoryenergy.com/heat_recovery_steam_generator/
28. Solar thermal collectors and applications, Soteris A. Kalogirou Department of Mechanical Engineering, Higher Technical Institute.
29. http://www.energybc.ca/profiles/solarthermal.html.

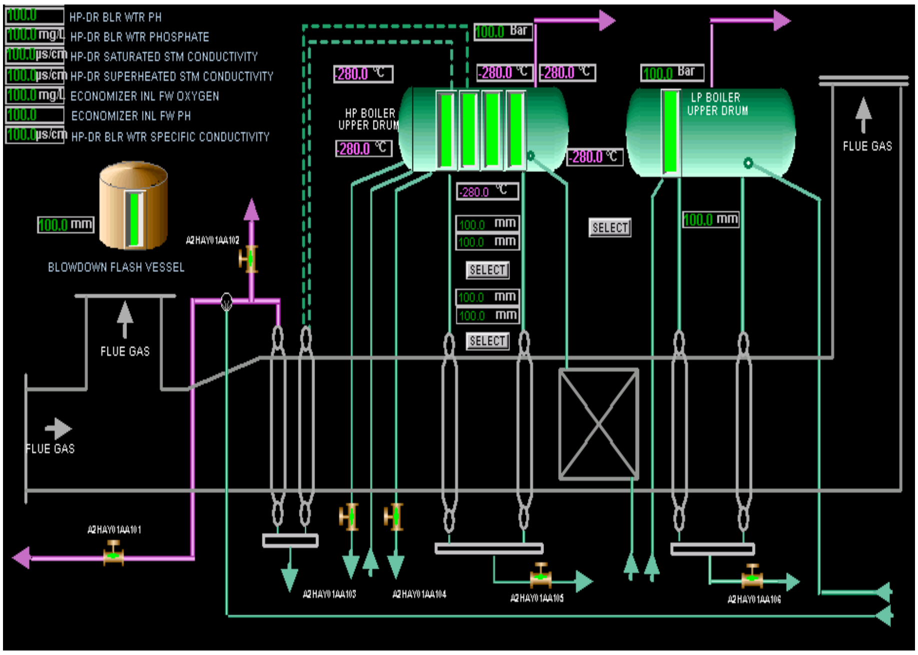
30. http://www.sajidarl.blogspot.com/2012/02/deaerator-is-device-that-is-widely-used.html.
31. http://www.zeroco2.no/capture/source-of-co2/combined-cycle-power-plant
32. http://www.mpoweruk.com/heat_engines.htm.
33. http://wwwairlimo.biz/gallery-of-activities.html.
34. GT log sheet and HRSG & ST log sheet plant-1 2013 & 2014 (GARRI power station).
35. http://www.archimedesolarenergy.it/enea_archimede_project.htm

Appendices

Appendix No: (A) Research Program Timetable

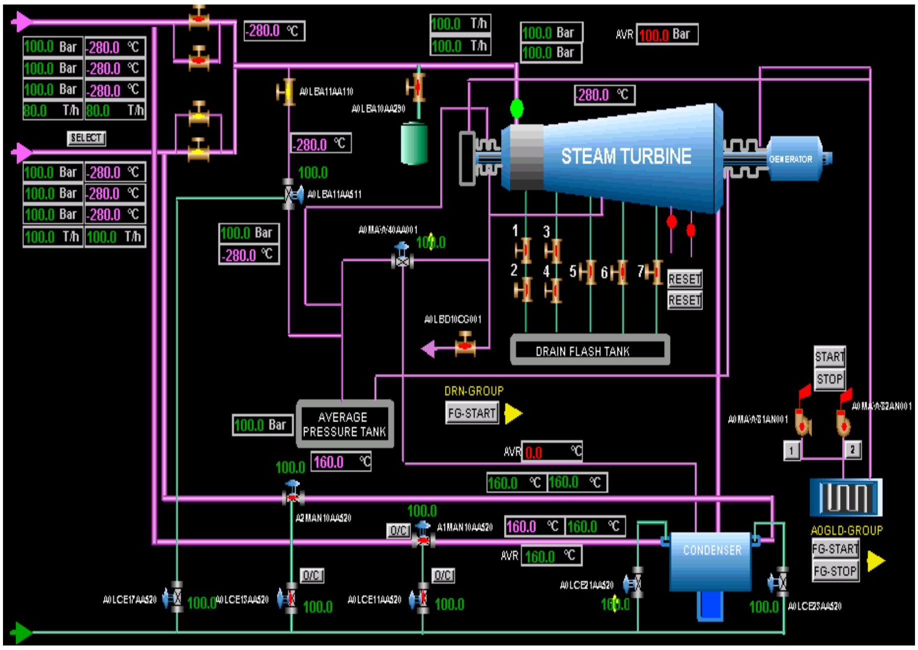
Activity	Year 2013				Year 2014				Year2015			
	25 %	50%	75%	100%	25%	50%	75%	100%	25%	50%	75%	100%
Literature review												
Combined cycle course work												
Combined cycle Simulation program												
Combined cycle simulation												
Preparing presentation												
Combined cycle economic sensitivity analysis and performance												
Experimental work												
Revising thesis												
proofing thesis												
Binding thesis												

Appendix No: (B)



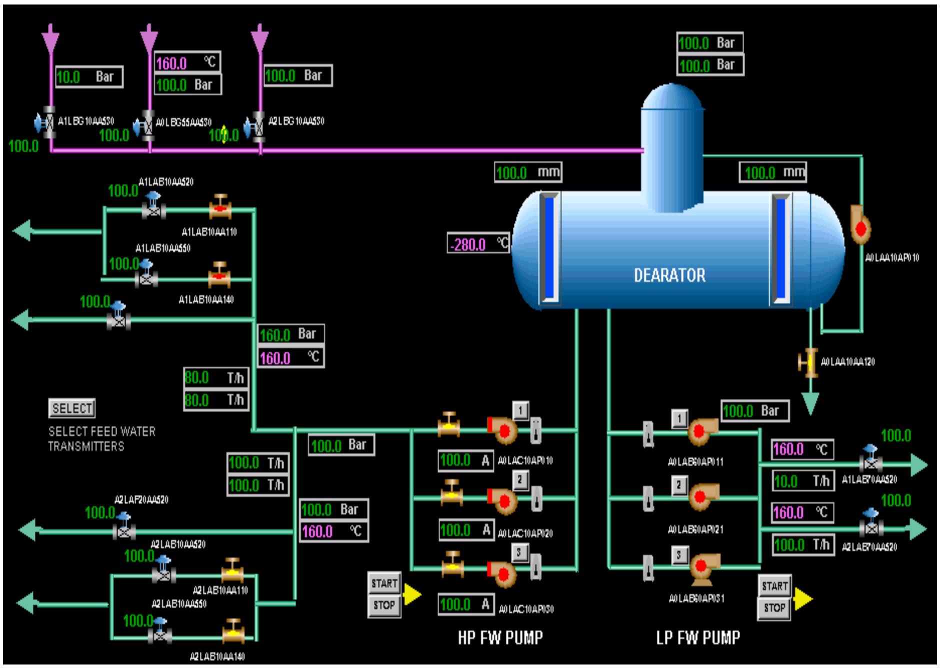
Water/ Steam flow diagram

Appendix No: (C)



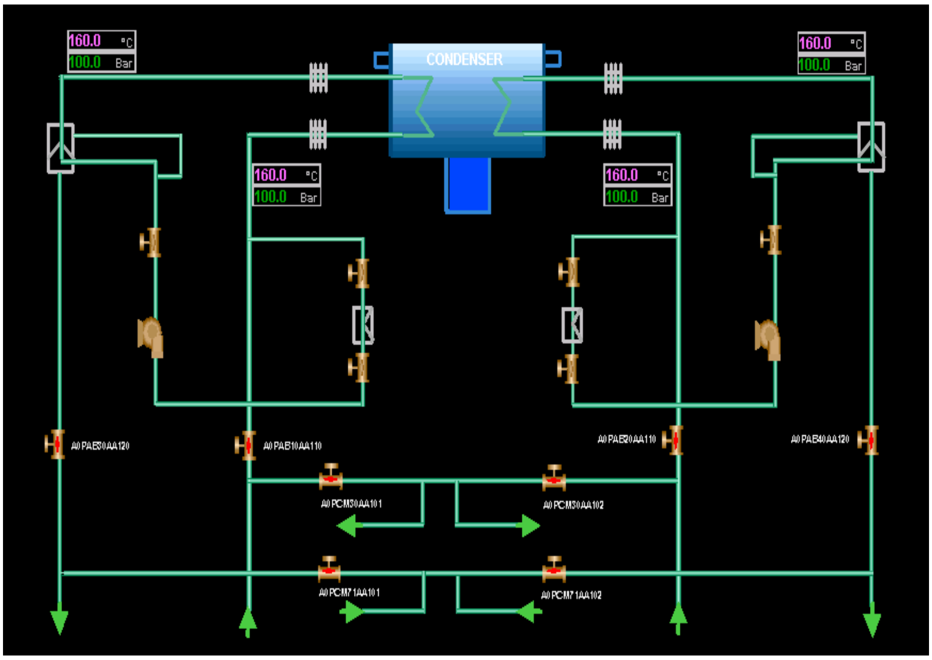
Main steam and bypass system diagram

Appendix No: (D)



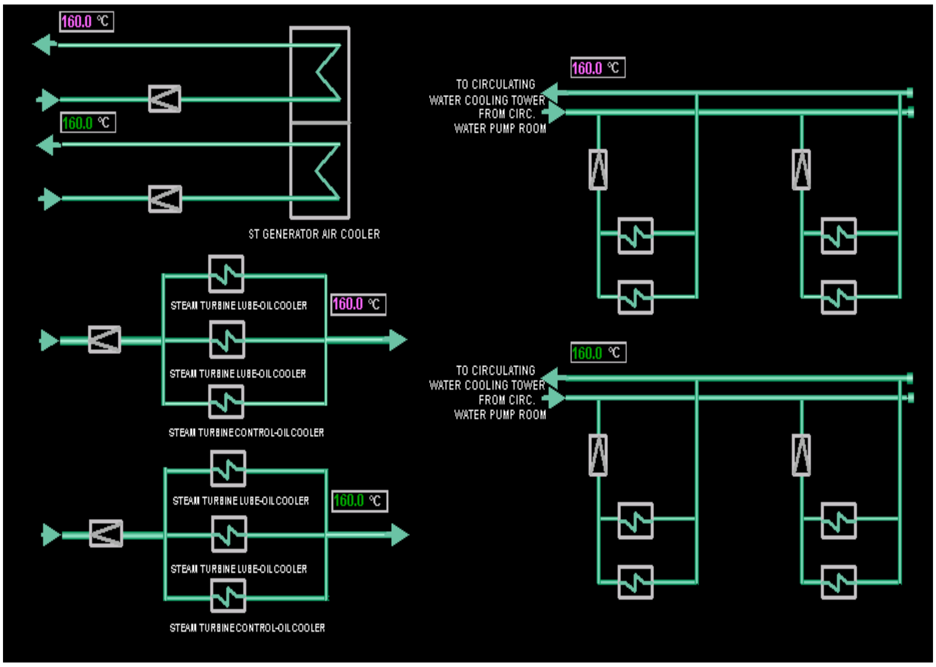
Deaerator heating steam system Diagram

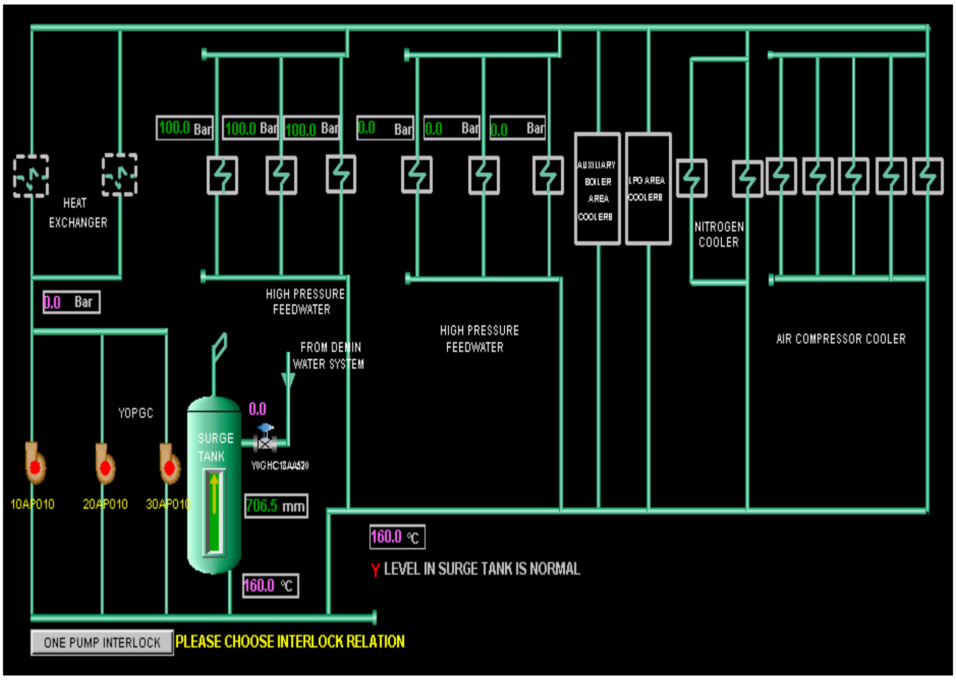
Appendix No: (E)

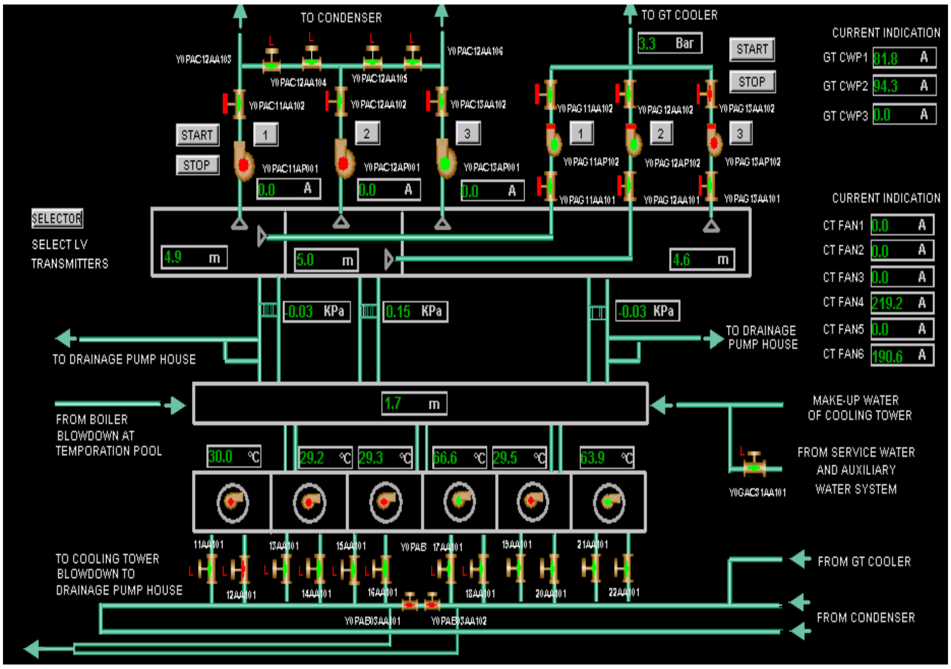


Condensate System Diagram

Appendix No: (F)

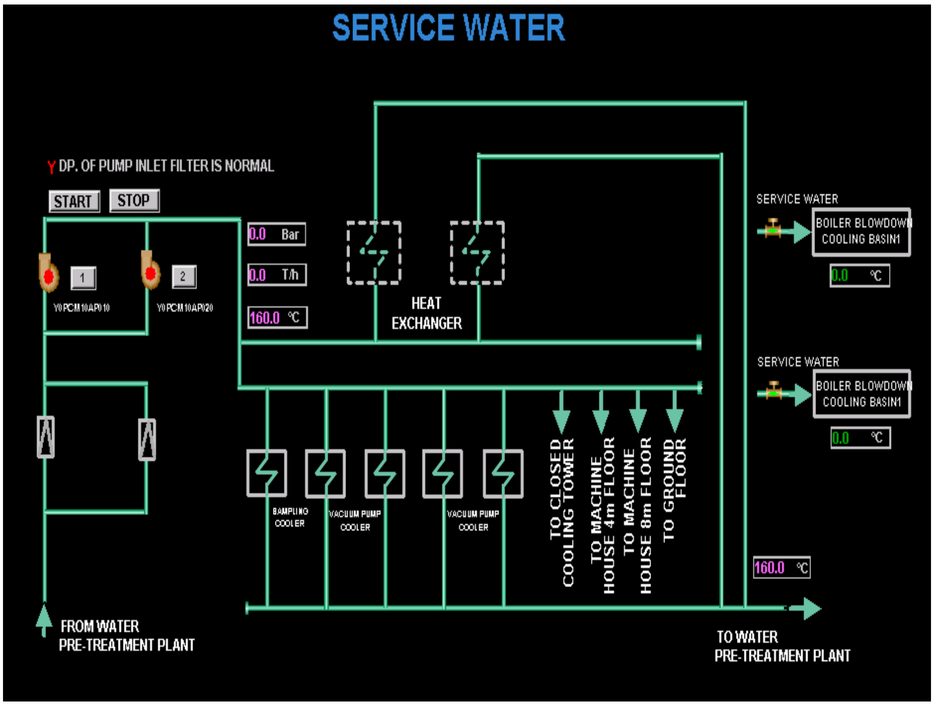






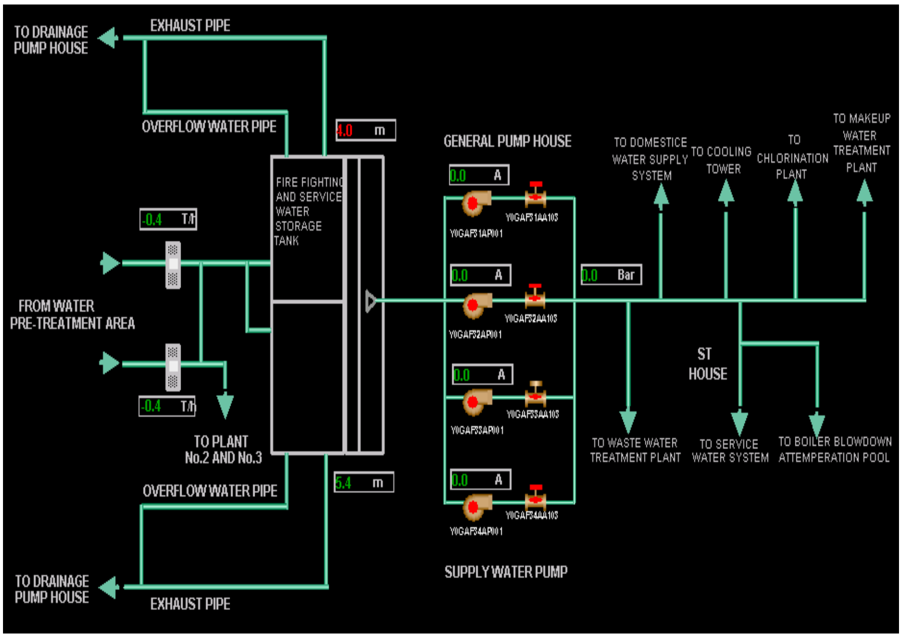
Circulating Cooling Water System Diagram

Appendix No: (G)



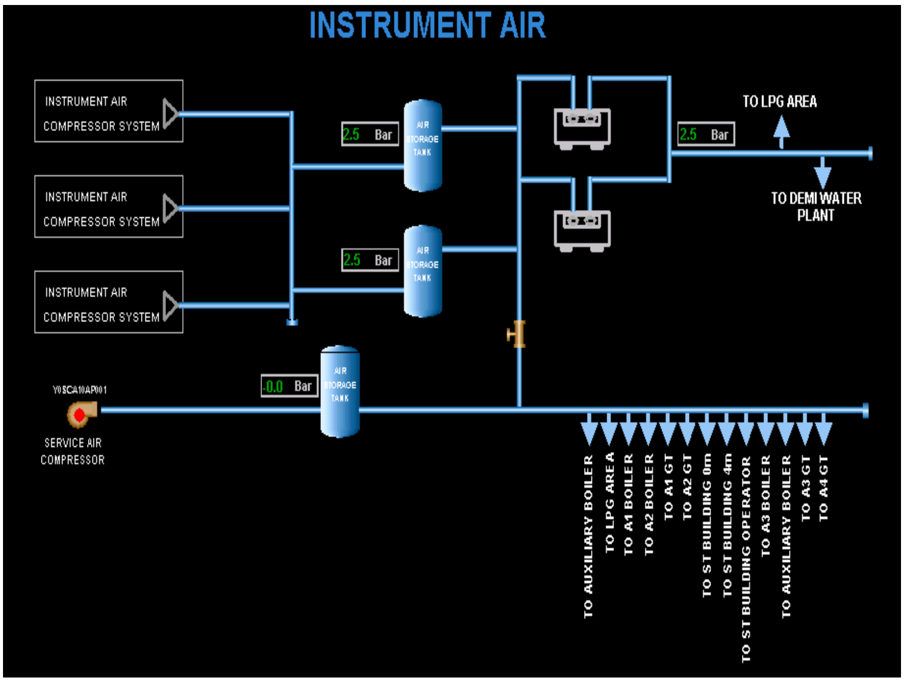
Service and Cooling Water System Diagram

Appendix No: (H)



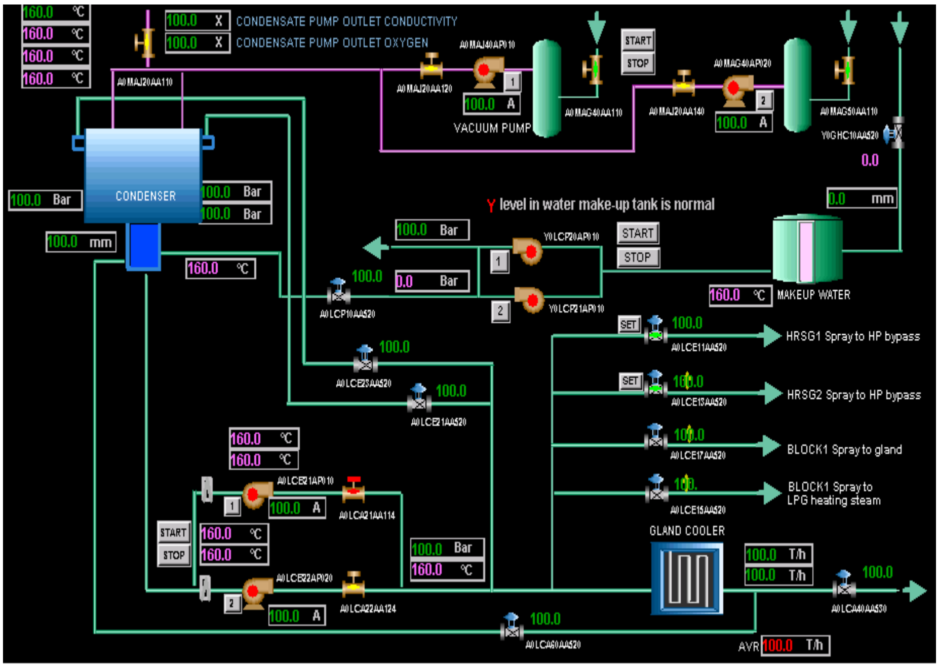
Make – up Water and filling Water System Diagram

Appendix No: (I)



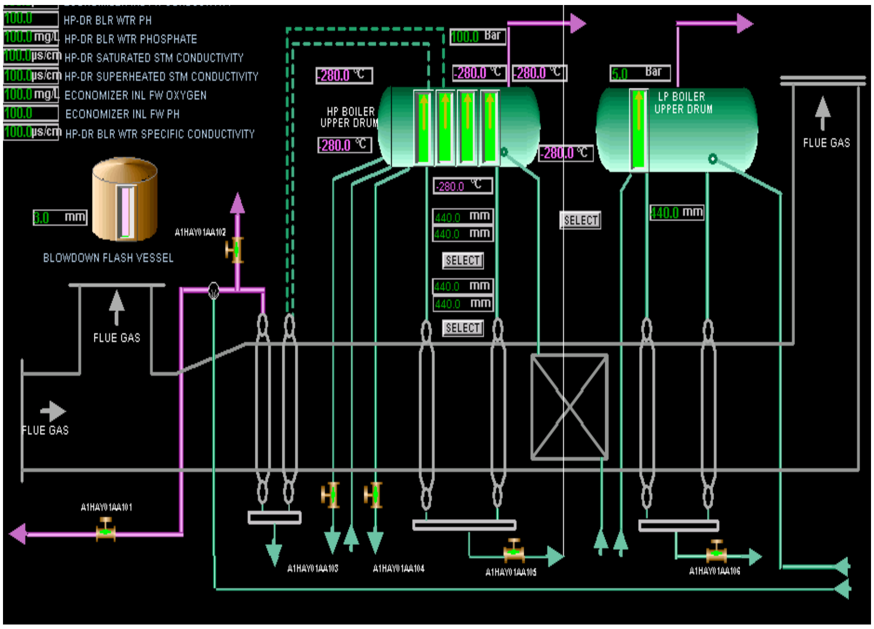
Instrument and service compressed Air system Diagram

Appendix No: (J)



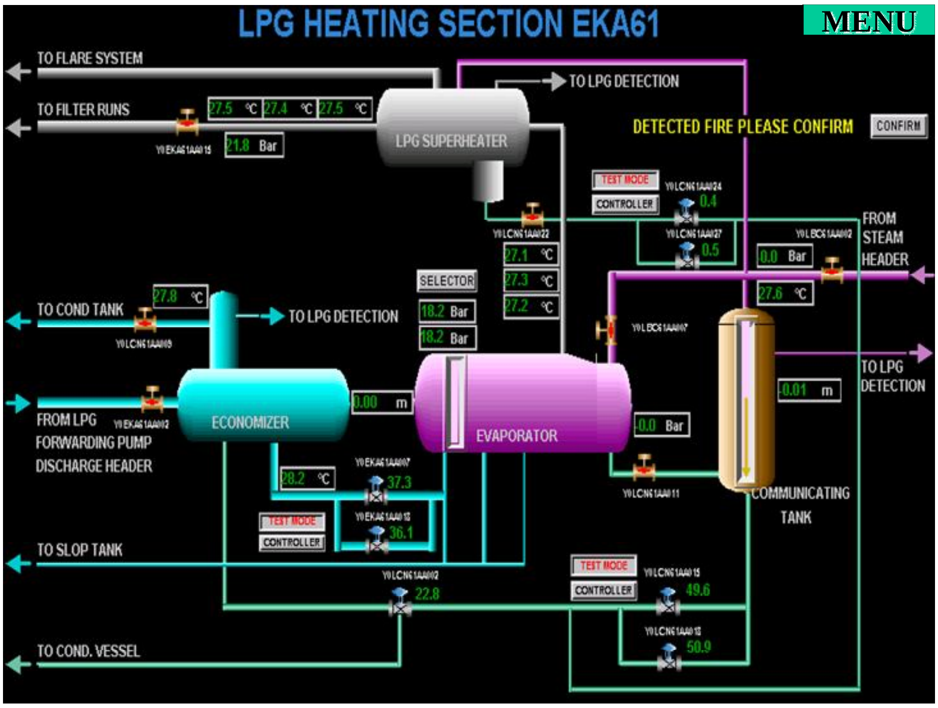
Condenser Air Evacuation system Diagram refers

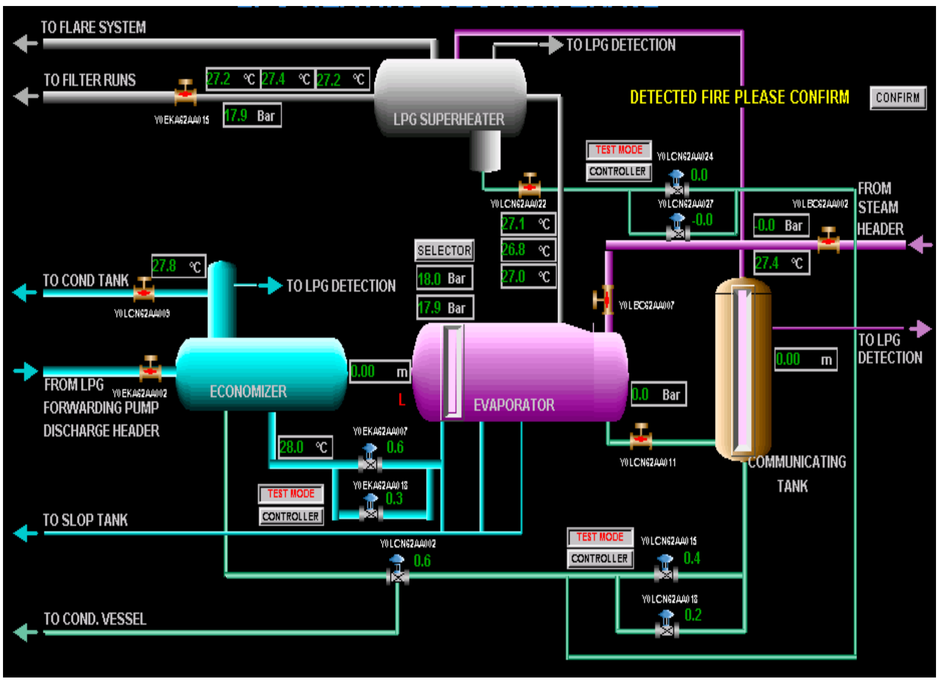
Appendix No: (K)

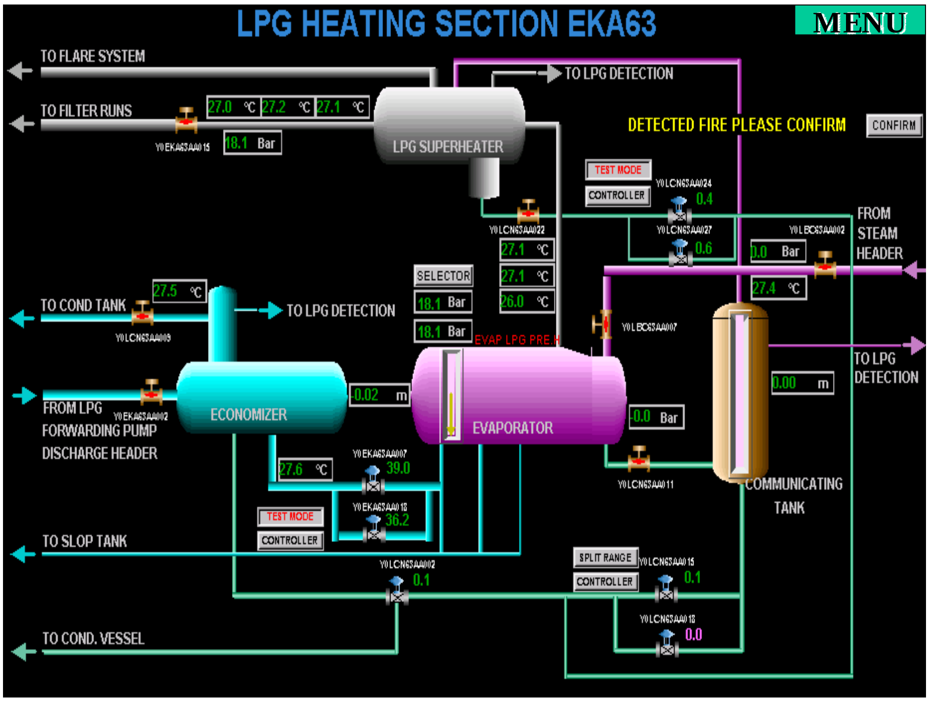


HRSG low pressure circulating system diagram

Appendix No: (L)







LPG heating steam system diagram

Appendix (N) mat-lab M-file program

```
function [V]= GTloop1(rp)
Tin1=input('lower_temperature          =');
Tin2=input('upper_temperature          =');
Tin=Tin1:Tin2;
for i=1:length(Tin);
clc
Comp_isen_eff=0.95;
Turbine_isen_eff=0.878;
Comp_mech_eff=0.90;
T1(i)= Tin(i);
T1(i)= Tin(i);
T3= 1544;
Gg=1.11;
Ga=1.33;
rp=9.10;
mf=5.16;
HCV=41100;
ma=249;
ws=32000;
%Compressor
T2s(i)=T1(i)*rp^(Ga-1)/Ga;
T2a(i)= ((T2s(i)-T1(i))/(Comp_isen_eff))+T1(i)
Cpa(i)=((1.0189*10^3)-(0.1378*T1(i))+((T1(i).^2)*1.9843*10^-
4)+((T1(i).^3)*4.2399*10^-7)-((T1(i).^4)*3.7632*10^-10))*10^-3
 %WC(i)=(Cpa(i)*(T2a(i)-T1(i)))
 WC(i)= ma*(Cpa(i)*(T2a(i)-T1(i)))
%WC(i)= ma*(Cpa(i)*(T2a(i)-T1(i)))/Comp_mech_eff;
%WC(i)= ma*Cpa(i)*(T2s(i)-T1(i))
%Combustion Chamber
%Qadd(i)=cpg*(T3-T2a(i));
%Qadd(i)=(mf+ma)*1.4*(T3-T2s(i))
Qadd=mf*HCV;
% Turbine
T4s=T3/((rp)^(Gg-1)/Gg);
T4a=T3-((T3-T4s)/(Turbine_isen_eff));
%WT=(cpg)*(T3-T4s);
%WT=(mf+ma)*(1.44)*(T3-T4s)
WT=(mf+ma)*(Ga)*(T3-T4a);
%Thermal efficiency calculation
Wnet(i)=(WT+ws)-WC(i)
eff(i)= (Wnet(i)/Qadd)*100.00
%eff = Wnet(i)/Qadd(i)
 %V(1)= eff;
% V(2)= Wnet;
end
plot(Tin,eff)
grid on
xlabel('Ambient  Temperature or Comperssor Inlet Temp(oC)')
ylabel('Garri Combined Cycle Power Effiency %')
title('Plant Over All  Effiency VS Ambient Temperature')
 end
```