بسم الله الرحمن الرحيم

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Exergy Analysis of Garri (1) Power Plant

A thesis

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Prepared by

Mudathir Mohamed Idress Shamaoon

Supervised by

Dr. Mohamed Musadag El-Awad

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قال الله تعالى:

(اللَّهُ نُورُ السَّمَاوَاتِ وَالْأَرْضِ مَثَلُ
نُورِهِ كَمِشْكَاةٍ فِيهَا مِصْبَاحٌ الْمِصْبَاحُ
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يُوقَدُ مِن شَجَرَةٍ مُّبَارَكَةٍ زَيْتُونِةٍ لَّا
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يَهْدِي اللَّهُ لِنُورِهِ مَن يَشَاءُ وَيَضْرِبُ
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شَيْءٍ عَلِيمٌ)

سـورة النور
الآيــة 35

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Abstract

The objective of this research is to presents an engineering design and theoretical exergetic analysis of the GARRI 1 combined cycle power plant. Exergy analysis has been performed based on the first and second laws of thermodynamics for power generation system.

Expressions involving the variables for specific power-output, thermal efficiency, exergy destruction in components of the combined cycle, second-law efficiency of the gas-turbine cycle, and second law efficiency of the steam power cycle have been derived, excel has been used for calculation.

The results show the exergy analysis for a steam cycle system predicts the plant efficiency more precisely. To evaluate the energy utilization, 100% load for plant using two kinds of fuel light diesel oil and liquefied petroleum gas and 66% load light diesel oil have been analyzed, so the plant second law efficiency when using LPG is 64.35 % which is better than LDO with 64.04 %. The plant efficiency for partial load operation is 61.6% which is lower than full load operation.

مستخلص البحث

يهدف هذا البحث لتناول التصميم الهندسي و التحليـل للطاقـة التـي يمكـن الحصول عليها لمحطة الدورة المشتركة قـرى (1). تحليـل الطاقـة التـي يمكـن الحصول عليها بني علـى أسـاس القـانون الأول و القـانون الثـاني للـديناميكا الحرارية لأنظمه توليد القدرة.

تضمن هذا التحليل كل من المتغيـرات الآتيـة، القـدرة المنتجـة، الكفـاءة الحرارية، الطاقة التي يمكـن الحـصول عليهـا المبـددة فـي مكونـات الـدورة المشتركة، كفاءة القانون الثاني لـدورة الحراريـة و دوره التـوربين الغازيـة تـم حسابها بواسطة برنامج الحاسوب Excel.

عرضت نتائج التحليل للطاقـة التـي يمكـن الحـصول عليهـا فـي دورة البخار لتحديد كفاءة الدورة بوضـوح. لتقيـيم الاسـتفادة مـن الطاقـة، المحطـة بحمولة 100% و باستخدام نوعين مـن الوقـود همـا الـديزل الخفيـف و الغـاز البترولي المـسيل بالإضـافة إلـي المحطـة بحمولـة 66% باسـتخدام الـديزل الخفيف تم تحليلها.وعليه فان كفـاءة القـانون الثـاني للمحطـة باسـتخدام الغـاز البترولـي المـسيل هـي 64,35% أفـضل مـن اسـتخدام الـديزل الخفيـف 64,04%. كفاءة المحطة بالحمـل التـشغيل الجزئـي هـي 61,6% اقـل مـن الحمل الكلي في كلا الحالتين أعلاه .

List of contents

List of figures

List of tables

Abbreviation

NEC	National electricity corporation
NGS	National grid system
CCPP	Combined cycle power plant
HRSG	Heat recovery steam generation
TIT	Turbine inlet temperature
CC	Combustion chamber
LHV	Low heat value
ISO	International standards organization
ASTM	American society for testing material
NG	Natural gas
GT	Gas turbine
GTG	Gas turbine generator
RGT	Reheat gas turbine
ST	Steam turbine
STG	Steam turbine generator
AC	Air compressor
C	Condenser
CP	Condensate pump
HP	High pressure
HPFWP	High pressure feed water pump
LP	Low pressure
LPFWP	Low pressure feed water pump
LPG	Liquefied petroleum gas
LDO	Light diesel oil
MSL	Mean sea level
RH	Relative humidity
SC	Simple cycle
LMTD	Logarithm means temperature different Subscript

Subscript :

i	inlet
o	outlet

Nomenclature

P	pressure absolute (MPa)
P_0	atmospheric pressure (MPa)
T	temperature (˚C)
T_0	atmospheric temperature (˚C)
ψ	exergy (kJ/kg)
ψ_f	fuel exergy (kJ/s)
ψ_p	process heat exergy (kJ/s)
ε_f	fuel exergy factor
ε_s	steam water heat exergy factor
E_f	fuel energy (kJ/s)
H	enthalpy (kJ/s)
h	enthalpy of produced steam/water (kJ/kg)
h_c	enthalpy of feed water return to HRSG (kJ/kg)
m_a	mass flow rate of air (kg/s)
m_f	mass flow rate of fuel (kg/s)
m_e	mass flow rate of exhaust flue gas (kg/s)
m_s	mass flow rate of steam/water (kg/s)
m_{sH}	mass flow rate of high pressure steam (kg/s)
m_{sL}	mass flow rate of low pressure steam (kg/s)
Q_P	process heat = $H_o - H_i$ (kJ/s)
R	power-to-heat ratio
s	entropy (kJ/kg.K)
W_E	electrical gross power output of plant (kW)
W_{GT}	gas turbines gross power output (kW)
W_{ST}	steam turbine gross power output (kW)
η_{CC}	thermal efficiency for combined cycle plant
η_{GT}	thermal efficiency for gas turbine
η_{HRSG}	thermal efficiency for heat recovery steam generator
η_{ST}	thermal efficiency for steam cycle
$\eta_{Rankine}$	thermal efficiency for Rankine cycle
η_I	first law efficiency
η_{II}	second law efficiency

CHAPTER ONE

INTRODUCTION

1.1. ELECTRICITY SUPPLY IN SUDAN

Electricity in Sudan is mainly supplied by the National Electricity Corporation (NEC), except small private generators in main towns that are not supplied by NEC. Many industrial and large commercial operations have standby generators, among which some industries have their own continuous generation. Most of NEC's customers are supplied by the National Grid System (NGS). NEC's isolated grid systems currently provide electric supply for fourteen main towns lying in states far from main grid, these systems comprise diesel generators, and small distribution networks predominantly supplying urban consumers. The quality of supply in the grid system is much superior to that of the isolated grids, which suffer from insufficient installed capacity, lack of spare parts, inadequate maintenance, and fuel shortage. The National Grid System was however, subjected to regular load- shedding at peak times.

The National Grid was affected annually during summer and rainy seasons in which the hydro generation drops causing lack of power supply about 50% from the available power, at the same time causing instability of the system due to disturbance in frequency which is controlled from Rosseries hydro power station.

Although the new generation in Garri (1) & (2) which has added 330 MW to the system, load shedding was occurred in the summer of 2006. NEC forecast had been expected a shortage of 150 MW in the summer of 2006 and that is why NEC was planning to add 150 MW in 2006, which was not happened. An additional load is connected to the grid through the new transmission line of Elgaili Shendi/ Atbara which is now 20 MW and is expected to increase very fast due to the restriction of generation capacity before these towns are connected to the grid.

In the past within two years the demand has increased by 24% (2004-2006), NEC setup five years plan to coup with this increase four contracts are already signed to increase the system capacity:

Garri 2 upgrading	100 MW.	(Implemented).
Khartoum North	200 MW.	(Under Construction)
Kosti	500 MW.	(Under Construction)
Garri -4	100 MW.	(Under Construction)

Due to increase of demand by 24% in 2004 [1], NEC rang the bell for importance of a fast track generation to fill the gab of generation shortage expected in the summer and rainy season of 2006. In 2005, the rate of demand increase was not changed.

This fast track is important, because the implementation of the above mentioned committed projects will not start before 2007, except Garri-II upgrading to combined cycle 100 MW, which is already constructed, and it partially was participated in filling the gab by at least 100 MW in summer of 2007, in last May 2007.Thus 100 MW was added, to the generation capacity of the system.

Beside the above mentioned projects , 400 MW in El Fula,300 MW in Kilo-X, 500 MW in El Bagair, 500 MW in Garri-III and 500 MW in Port Sudan Power Stations are committed in addition to 1250 MW from Marawe Project. The White Nile Grid 220 KV Rosairess, Rank, Rabak and Khartoum with tap off from Rabak to El Obied which will serve many isolated towns from the grid. This transmission lines are under construction now.

1.2. ELECTRICAL SYSTEM IN SUDAN

Electrical System in Sudan comprises approximately 1400 km of high voltage transmission that is supplied both by thermal and hydropower stations attached table (1-1). In addition, fourteen isolated centres are served by thermal generating plants and local distribution networks see table (1-2).

TOTAL INSTALLED GENERATION IN THE NATIONAL GRID

TOTAL OF HYDRO	***343.3*** MW
TOTAL OF THERMAL	***730.7*** MW
TOTAL GENERATION	**1074** MW

Table (1-1): Existing thermal and hydro power capacity in the national grid 2005 [1].

Plant	***No units***	***Unit MW***	***Total MW***
Roseires	7	40	280
Sennar	2	7.5	15
Khashm El Girba Kaplan Tur.	2	5.5	11
PUMP Turbine	3	2.3	6.9
Jebel Aulia[1]	80	0.38	30.4
SUB-TOTAL HYDRO			***343.3***
Khartoum North Phase One	2	30	60
Khartoum North Phase Two	2	60	120
Khartoum North 1&2/GT	2	20	40
Khartoum North 3&4/GT	2	25	50
Kuku 1&2/GT	2	10	20
Gerri I(1)	6	35	210
Gerri (2)	6	35	210
Kassala/Diesel (4 + 1 sets)	5	3.5+4	7.5
Faw	2	6.6	13.2
SUB-TOTAL THERMAL			***730.7***
TOTAL Generation			**1074**

Table (1-2): Isolated power plants [1].

Plant	***No Units***	***Unit MW***	***Total MW***
Um Rwaba	3	1x0.6-2x1	2.6
Port sudan	A/B/C/D	30+6.6+15+2	53.6
Wadi Halfa	3	3X0.6	1.8
Juba	5	1	5
Waw	*2*	*0.92*	*1.84*
Malkal	2	1	2
Elgenena	2	3	6
Nyala	6	5	30
Elfasher	8	1	8
Elobied	4	3.5	14
Kadugli	2	1	2
Eldien	2	1	2
Karima	4	2.5	10
Elnuhood	2	0.8	1.6
ELDUIM	2	1.98	3.86
TOTAL			144.3

1.2.1. NEC Short (2005-2010) and Long (2005-2030) Term Plans.

NEC planned to implement the following generation power plant as described in the following table (1-3) in the five years plan (2005-2010)

Table (1-3): Generation in the five years plan (2005-2010) [2].

Year	Additional Generation Plan (2006-2010)	MW
2006	35 MW CCGT Garri (2)	35
2007	70 MW CCGT Garri (2)	70
	2x35 MW Fast Track Kilo (x)	70
	1×55 MW- 1st Unit Garri (4)	55
	2×125 MW Morewe Hydro. Main Nile	250
	2×25 Kassala Diesel	50
2008	1x100 MW Steam Kh. N (Phase 3)	100
	2x125 MW Steam Kosti	250
	2×135 MW Steam Generation at EL Fula	270
	1×55 MW 2nd Unit Garri (4)	55
2009	2×135 MW Garri(3)	270
	1x100 MW Steam Kh. N (Phase 3)	100
	2x135 MW Steam ELBagair	270
	2X 135 MW - Coal/generation Port Sudan	270
	3×125 MW Morewe Hydro	375
	1×135 MW Steam Generation at EL Fula	135
	1x125 MW Steam Kosti	125
	2X 135 MW - Coal/generation Port Sudan	270
2010	2×135 MW Garri(3)	270
	1×135 MW Steam Generation at EL Fula	135
	3×125 MW Morewe Hydro	375
	2x135 MW Steam ELBagair	270
	1x125 MW Steam Kosti	125

The load forecast for the long term plan is shown in figure (1-1) and table (1-4), which represents that during the next three years there is gab between the generation and demand.

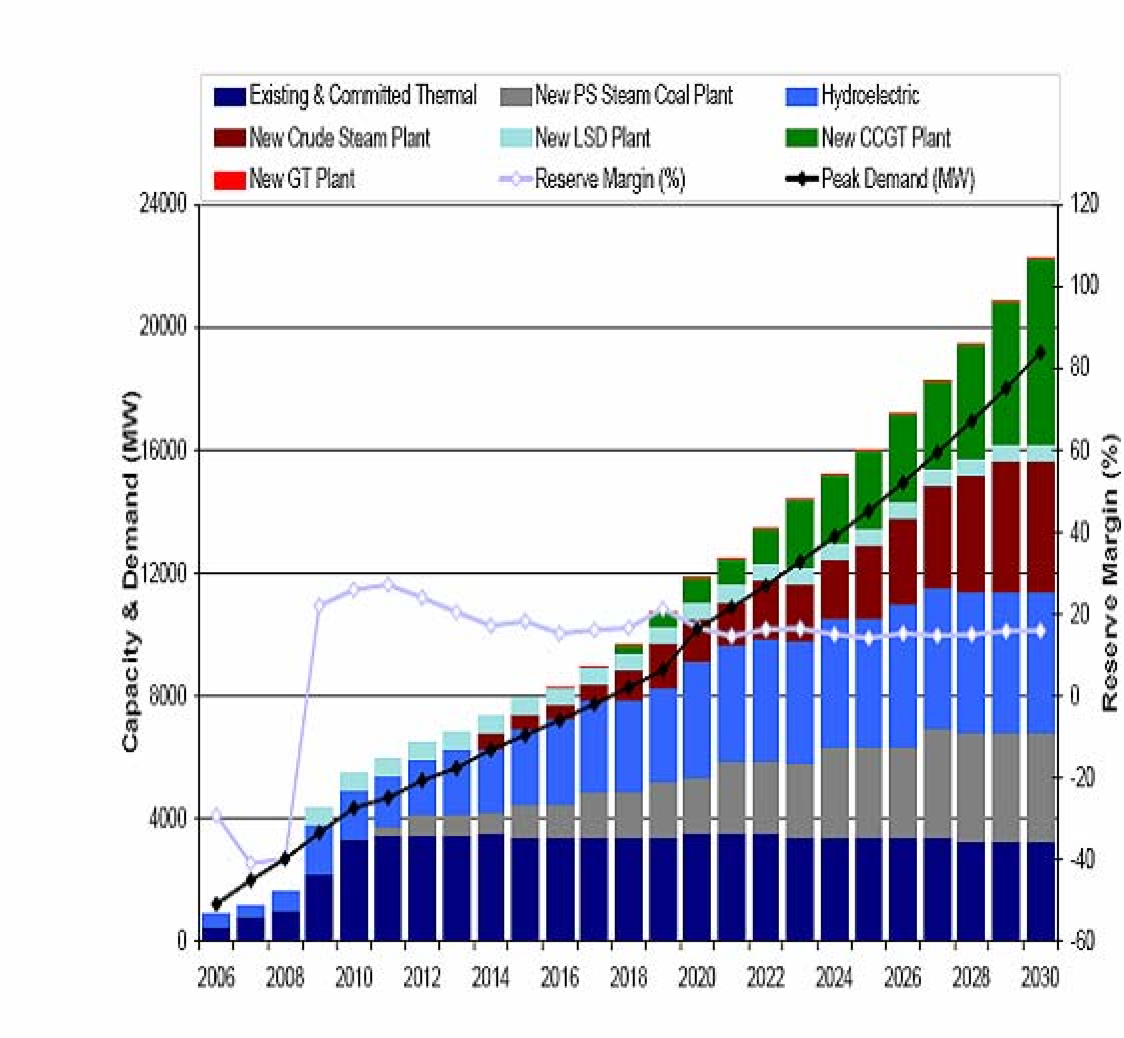

Figure (1-1): High load forecast planting programme [3].

Table (1-4): High load forecast planting programme [3].

YEAR	TOTAL CAPACITY (MW)	DEMAND (MW)	RESERVE MARGINE (MW)	RESEVE MARGINE (%)
2006	**871**	**1228**	**-357**	**-29**
2007	**1187**	**2007**	**-820**	**-41**
2008	**1627**	**2699**	**-1072**	**-40**
2009	**4333**	**3549**	**784**	**22**
2010	**5486**	**4354**	**1132**	**26**

1.3. SECOND-LAW OF THERMODYNAMIC ASPECTS OF DAILY LIFE

Thermodynamics is a fundamental natural science that deals with various aspects of energy, and even non-technical people have a basis understanding of energy and the first law of thermodynamics since there is hardly any aspect of life that does not involve the transfer or transformation of energy in different forms. All the dieters for example, base there lifestyle on the conservation of energy principle. Although the first law aspects of thermodynamics are readily understood and easily accepted by most people there is not a public awareness about the second law of thermodynamics, and the second law aspects are not fully appreciated even by people with technical backgrounds. These cause some students to view the second law as something that is of theoretical interest rather than an important and practical engineering tool. As a result, students show little interest in a detailed study of the second law of thermodynamic. This is unfortunate because the students end up with a one-sided view of thermodynamic and miss the balanced, complete picture.

Many ordinary events that go unnoticed can serve as excellent vehicles to convey important concepts of thermodynamics. Below we will attempt to demonstrate the relevance of the second low concepts such as exergy, reversible work, irreversibility, and the second law efficiency to various aspects of daily life using examples with which even nontechnical people can identify. Hopefully, this will enhance our understanding an appreciation of the second law of thermodynamics and encourage us to use it more often in technical and non-technical areas. The critical reader is reminded that the concepts presented below are soft and difficult to quantize, and that they are offered to stimulate interest in the study of the second law of thermodynamics and to enhance our understanding and appreciation of it.

The second concepts are implicitly used in various aspects of daily life. Many successful people seem to make extensive use of them without even realizing it. There is growing awareness that quality plays as important a role as quantity in even ordinary daily activities.

In thermodynamics, reversible work for a process is defined the maximum useful work out put (or minimum work input) for that process. It is the useful work that a system would deliver (or consume) during a process between two specified states if that process is executed in a reversible (perfect) manner. The difference between the reversible work and the actual useful work is due to imperfections and is called irreversibility (the wasted work potential). For the special case of the final state being the dead state or the state of the surrounding, the reversible work becomes a maximum and is called the exergy of the system at the initial state. The irreversibility for a eversible or perfect process is zero.

The exergy of a person in daily life can be viewed as the best job that person can do under the most favourable conditions. The reversible work in daily life, on the other hand can be viewed as the best job a person can do under some specified conditions then the difference between the reversible work and the actual work under those conditions can be viewed as irreversibility or the exergy destroyed. In engineering system, we try to identify the major sources of irreversibility's and minimize them in order to maximize performance. In daily life, a person should do just that to maximize his or her performance.

The exergy of a person at a given time and place can be viewed as the maximum amount of work he or she can do at that time and place. Exergy is certainly difficult to quantify because of the interdependence of physical and intellectual capabilities of a person. The ability to perform physical and intellectual tasks simultaneously complicates things even further. Schooling and training obviously increase the exergy of a person. Aging decreases the physical exergy. Unlike most mechanical things, the exergy of human beings is a function of time, and the physical and/or intellectual exergy of a person goes to waste if it is not utilize at the time. A barrel of oil loses nothing from its exergy if left unattended for 40 years. However, a person will lose much of his or her entire exergy during that time period if he or she just sits back.

The increased awareness that the world's energy resources are limited has caused some governments to re-examine their energy policies and take drastic measures in eliminating waste. It has also sparked interest in the scientific community to a closer look at the energy conversion devices and to develop new techniques to better utilized the existing limited resources. The first law of thermodynamic deals with the quantity of energy and asserts that energy can not be created or destroyed. This law serves as a necessary tool for the book keeping of energy during process and offers no challenges to the engineer. The second law however deals with the quality of the energy. More specifically, it is concerned with the degradation of energy during a process, the entropy generation, and the lost opportunities to do work, and it is offers plenty of room for improvement [4].

1.4. OBJECTIVES OF STUDY

The objective of this research is to:

1. Examine the performance of Garri power station in light of the first law and second law of thermodynamics.
2. Take drastic measures in eliminating waste.
3. Pointing for new techniques to better utilize the existing limited resources.

CHAPTER TWO
LITERATURE REVIEW

Studies of engineering designs and exergy analyses for power generation systems are of scientific interest and also essential for the efficient utilization of energy resources. For this reason, the exergy analysis has drawn much attention by scientists and system designers in recent years. Some devoted their studies [5,6] to component exergy analyses [7] and efficiency improvement; others concentrate on systems design and analyses[8-12].

A computerized thermodynamic analyses of a gas turbine based combined cycle power plant (CCPP) fitted with a triple pressure HRSG and hot reheat foresees the efficiency reaching a value of 62% [8]. Huang [9] shows that the performance evaluation of a CCPP based only on the first law of thermodynamics is not adequate, but the second law of thermodynamics must be taken into consideration to get a better evaluation. Verkhivker and Kosoy [10] pointed out the principal processes which cause the destruction of exergy in a power generation cycle are the combustion process, the subsequent heating of the working fluid and the heat transfer in the heat exchangers. For a combined triple (Brayton/Rankine/Rankine)/ (gas/steam/ ammonia) power cycle, Marrero et al [11]. also confirms that the largest irreversibility is produced in the combustion process. It decreases with the increase of the gas turbine inlet temperature.

Parametric studies performed by Bilgen [12] show that the first law and second law efficiencies are decreased with the increase of the power-to-heat ratio. The first law efficiency is strongly related to the power-to-heat ratio in a cogeneration plant. The efficiency is reduced around 40% when the power-to-heat ratio increases from 1 to 20. On the other hand, the second law efficiency is degraded only about 2% when the power-to-heat ratio increases from 1 to 20. It complies with the second law of thermodynamics—work is the valuable commodity of a power plant.

Work can be completely and continuously converted to heat. Heat is not completely converted to work in a cycle.

Pak and Suzuki [13] studied exergy in the improvement of gas turbine co generation based on T_0 = 15˚C and P_0 = 101 kPa as the dead states using 3 methods:

1. Increase temperature of the hot gas entering the gas turbine (TIT) about 200 K.
2. Use a regenerator between gas-turbine outlet and compressor outlet gas before entering the combustion chamber (CC).
3. Use superheated steam from waste heat boiler for the combustion chamber (Assuming that natural gas is composed mainly of methane).

Before improvement, the maximum exergy losses are found at the combustion chamber. Total exergy losses are around 58.7%. And total exergetic efficiency is 41.3%.

Increasing gas temperature entering the gas turbine from 1273 to 1473 K (method 1) makes a decrease in total exergy loss about 4.1% and an increase in total exergetic efficiency about 4.1%.

Installing the regenerator (method 2) makes an increase in the total exergetic efficiency to be about 43.9 %. Because the exergy loss was decreased at combustor and was occurred slightly at the regenerator.

In method 3: to maintain the outlet temperature, more fuel input is needed. The exergetic efficiency of power generation was increased about 10.2%. But the exergetic efficiency of heat generation was decreased about 14 %. Then the total exergetic efficiency was decreased about 3.8%. This method is suitable in the case of low steam generation in a waste heat boiler.

Deng-Chern Sue- Chia-Chin Chuang [14] presented a paper that presents the engineering design and theoretical exergetic analyses of the plant for combustion gas turbine based power generation systems.

Exergy analysis is performed based on the first and second laws of thermodynamics for power generation systems. The results show the exergy analyses for a steam cycle system predict the plant efficiency more precisely. The plant efficiency for partial load operation is lower than full load operation. Increasing the pinch points will decrease the combined cycle plant efficiency. The engineering design is based on inlet air-cooling and natural gas preheating for increasing the net power output and efficiency. To evaluate the energy utilization, one combined cycle unit and one cogeneration system, consisting of gas turbine generators, heat recovery steam generators, one steam turbine generator with steam extracted for process have been analyzed. The analytical results are used for engineering design and component selection.

To analyze the engineering system design and enhance gas turbine performance, two different power plants were selected to perform the experiments in this study. The first one is combined cycle Plant A; the other one is cogeneration Plant B. Fig.(2-1) illustrates the schematic flow diagram for Plant A; Fig. (2-2) shows the schematic flow diagram for Plant B. Plant B is not equipped with NG preheating. The processes for Plants A and B are similar when no steam is extracted for offsite steam users from the Plant B steam turbine. These plants are briefly described in Table 2-1.

Taiwan is located in a sub-tropical zone; the temperature difference between summer and winter is moderate. However, the power output decreases when operated in the warmer ambient air conditions due to the lower air mass flow rate.

The gas turbines information of Plants A and B are presented in Table 2-2. Normally, the gas turbine manufacturers quote performance based on ISO and LHV conditions. ISO conditions mean 15˚C ambient air temperature, 101.325 kPa barometric pressure, and 60% relative humidity. LHV means the lower heating value of the fuel being used. ASTM D3588 is the method used to determine the lower heating value of the NG fuel supplied to both Plants A and B. heating value of the NG heating value of the NG fuel supplied to both Plants A and B. heating value of the NG fuel supplied to both Plants A and B. heating value of the NG fuel supplied to both heating value of the NG fuel supplied to both Plants A and B.Plants A and B.

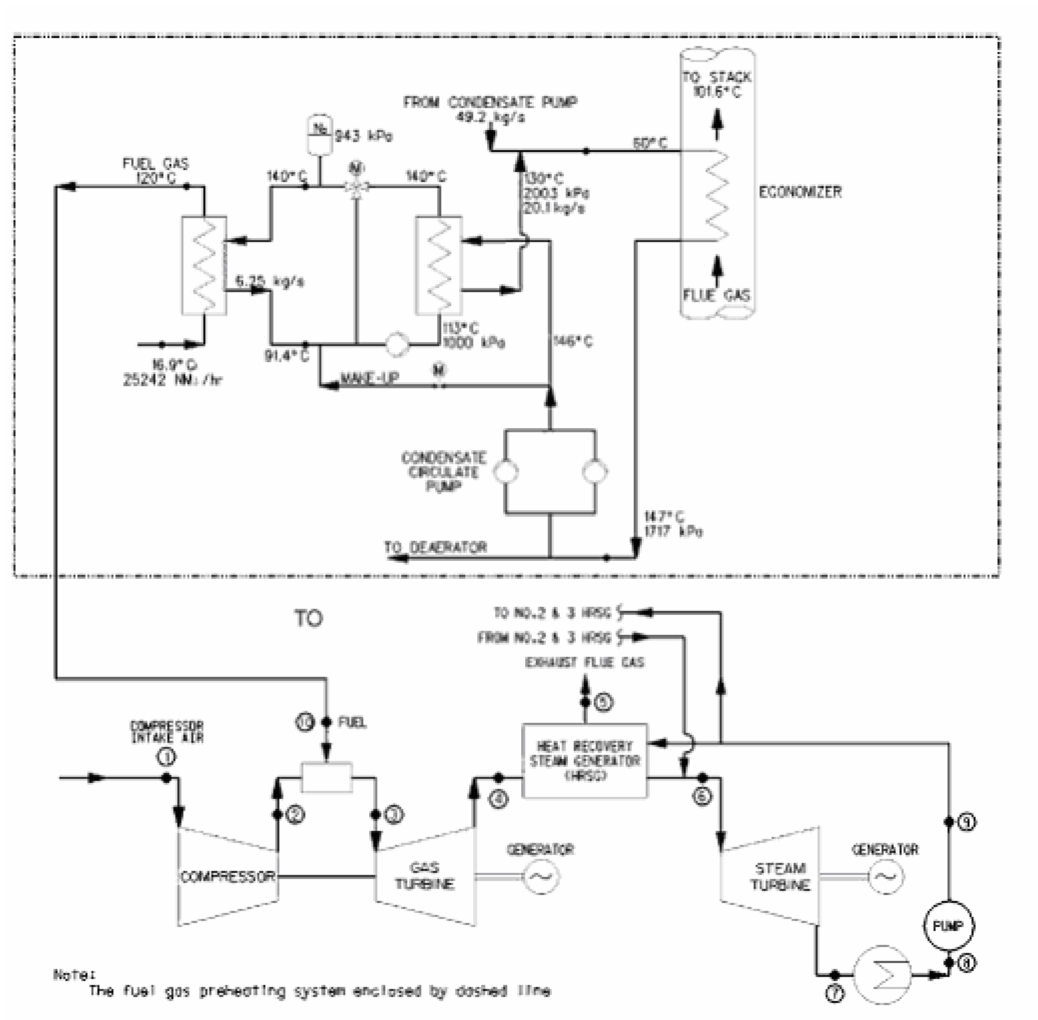

Fig. 2-1. Schematic flow diagram for Plant A.

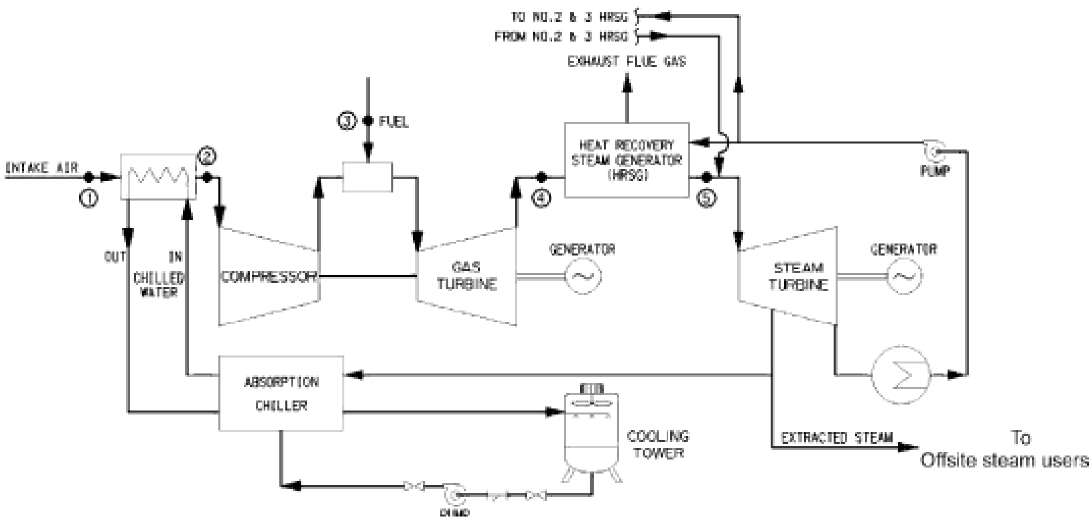

Fig. 2-2. Schematic flow diagram for Plant B.

Table 2-1: description of plants A and B

Plant type	Plant A	Plant B
	5 combined cycle units	1 cogeneration cycle uni
Combination	3GTGs+3HRSGs+1STG	3 aero derivative GTGs + 3HRSG+ 1STG
Total installation capacit (MW)	2200	150+45 ton/h steam
Site elevation (meter above MSL)	4	138
Annual average RH(%)	80	78.5
Highest/lowest daily T(˚	35.4/7.4	38/6
Average annual T(˚C)	25.1	22.3
Design condition (˚C and %RH)	32 and 90	32 and 90
Preheated fuel gas T(˚C)	120	N/A
Air inlet cooling T(˚C)	N/A	5.5+0.5
Commercially operated	January 1999	January 2000

Table 2-2 gas turbine design information for plants A and B

Type	Plant A	Plant B
	Heavy duty	Aero derivative
No. of compressor stages per G	17	19 (14 HP + 5LP)
No. of turbine stages per GTG	4	7 (2HP + 5LP)
No. of shaft per GTG	1	2
Shaft rotor	Solid	Hollow concentric
Shaft speed (rpm)	3600	HP = 10000, LP = 3600
No. of combustors	2	1 annular
No. of burners per combustor	6	30
Compression ratio of GT	10.2	29.4

They conclude that based on first law (energy) and second law (exergy) analyses, the formulas for dual pressure HRSG have been developed for thermodynamic performance and engineering betterment of combustion gas turbine based power generation systems. The plant performance can be improved by improving the system design such as using dual pressure steam cycles, inlet air cooling for the gas turbine, and fuel gas preheating, etc. the efficiency difference of CCPP between second law and actual plant operation at design condition is 22.13%, which is caused by a steam cycle efficiency of only 32.74% as shown. The efficiency deviation between exergy analyses and performance test results can be improved by engineering design.

The higher exergy destruction occurs because the lower pressure steam has high entropy value at same temperature level. Therefore, the CCPP operated at 50% load has an efficiency of 2.4% lower than the 100% load. The HRSG pinch point increased 10˚C only affects the overall combined cycle efficiency by 0.3%. The pinch point design value has to be carefully evaluated based on anticipated operating factors to obtain an optimum design.

Experiments conducted on the 90 MW rated gas turbines of Plant A, show that, with the installation of a fuel gas preheating system from 22.5 to 118 ˚C, less fuel consumption can be achieved under the same power output. Operating records indicate fuel savings and a slight improvement in efficiency (0.06%) for the whole plant.

The gas turbine Plant B produces 18% greater power output using 15˚C intake air than 30˚C air. In many power purchase agreements (PPA), the capacity charge is based on the peak output that can be demonstrated on a predetermined hot summer day. In such cases, the power output reduction in summer is economically unacceptable to the power producer. On the other hand, the plant operational data depict that the efficiency will be 1% lower when compressor inlet air is cooled from 10 to 5 ˚C. The trend of energy efficiency is not proportional to the power output. Therefore, the optimum operational inlet air temperature to the compressor will depend on the gas turbine design under the consideration of energy saving and contractual requirements.

From the engineering design point of view, if the offsite steam user uses process steam exclusively for absorption chillers associated with air conditioning, the power facility can install the chillers and directly provide the chilled water instead of the process steam. It will eliminate many problems such as steam traps, expansion joints, and return condensate contamination, etc.

This study reported by Somkiat Boonnasa and Pichai Namprakai[15] to assess the performance of the EGAT (Block 1), combined cycle power plant using the exergy method. Located in Samutprakran, Thailand the studied power plant consists of two unit of gas turbine and a unit of steam turbine which has been operated for 9 years. It has total capacity of 322.45 MW. Pressure ratio of the air compressor is 12. Natural gas is fuel input and the mass ratio of air to fuel is 44:1. The dead states of the environment in Thailand, which are 25 ˚C and 1.013 bar, are used in this analysis. It was found that exergy efficiency of the steam turbine, the gas turbine, heat recovery steam generator (HRSG) and the combined cycle power plant were 82.8, 20.7, 56.5 and 31 % respectively. The total exergy loss of the power plant is 718.7 MW (69% of exergy input). The benefit of this study is that the effectiveness of the performance improvement of a thermal system can be evaluated. In addition, the energy resources can be utilized efficiently.

Established in Southern part of Bangkok, the capital of Thailand, the EGAT (Block 1) combined cycle power plant in this study has been operated for 9 years since 1995. The system consists of 2 sets of gas turbine (GT), 103.93 MW each, and 1 set of steam turbine (ST), 114.526 MW, as shown in Fig.2-3. The cooling water of the condensing system will be pumped from the Chao-pra ya River. Combustion chamber of the gas turbine is fueled with a natural gas. Air to fuel ratio is 44:1. In Fig.2-3. two units of heat recovery steam generator (HRSG) which uses exhaust gas from the gas turbines to generate a steam for the steam turbine is also shown.

Due to the decline in various equipment performances and the more in energy losses, exergy efficiency analysis thus is needed especially in the combined cycle, steam turbine and HRSG. It was assumed that the two gas turbines were similar. In this study, exergy balance was then evaluated.

The benefit of this study is that the effectiveness of the performance improvement of a thermal system can be evaluated. In addition, the energy resources can be utilized efficiently

Table 2-3 conditions and parameters used in the calculation

Descriptions	Values
Ambient condition	25 ˚C / 101.3 kPa
TIT	1057 ˚C
Compressor inlet temperature	25 ˚C
Compressor pressure ratio	12
Mass flow rate of air	876.2 kg/s
Air to fuel ratio (by mass)	44:1
NCV of fuel	50.14 MJ/kg
Chemical exergy of fuel	52.15 MJ/kg
Gas turbine output (2 units)	207.86MW
Steam turbine output (1 unit)	114.53 MW

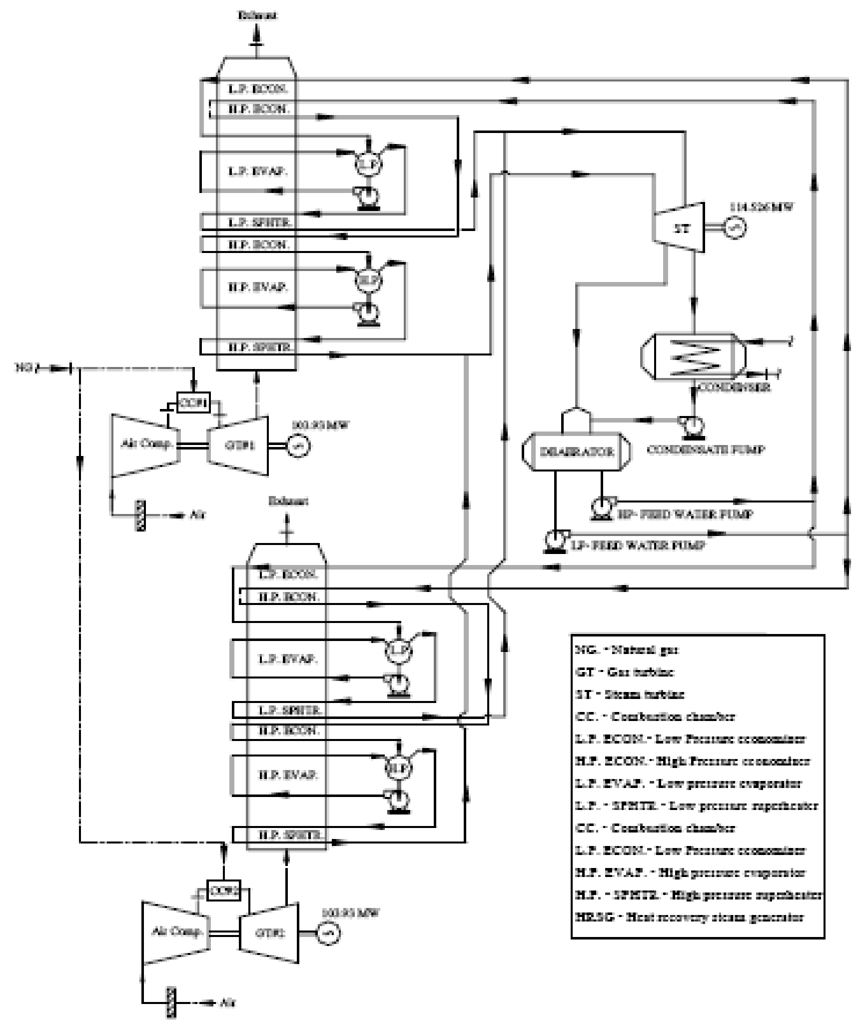

Fig. 2-3 Schematic diagram of EGAT-Block 1 (Combined cycle).

Somkiat Boonnasa and Pichai Namprakai conclude that according to an analysis, it was found that exergy efficiency of gas turbine was the lowest. Maximum exergy loss was found at the combustor (50% of input). The second important loss was at HRSG (10% of input). Exhaust gas temperature is too high which may result from the fouling on surface area of a heat exchanger thus a lower overall heat transfer coefficient in HRSG. Cleaning the heat transfer surface can solve this problem then exergy loss to environment can decrease substantially. If the exhaust temperature is deceased by 30˚C then the steam turbine output would increase 4.2 MW (3.64 %) and so does the combined cycle. It would save 73 MBaht/year. And the environment was also maintained.

To prevent the accumulation of sulfuric acid the final temperature should not be lower than the dew point. For the combustor, to improve its performance the STIG (steam-injection gas turbine) would be used. But it would also increase the capital cost.

A. Khaliq and S.C.Kaushik[16] present paper which use the second-law approach for the thermodynamic analysis of the reheat combined Brayton/Rankine power cycle. Expressions involving the variables for specific power-output, thermal efficiency, exergy destruction in components of the combined cycle, second-law efficiency of each process of the gas-turbine cycle, and second law efficiency of the steam power cycle have been derived. The standard approximation for air with constant properties is used for simplicity. The effects of pressure ratio, cycle temperature ratio, number of reheats and cycle pressure-drop on the combined cycle performance parameters have been investigated. It is found that the exergy destruction in the combustion chamber represents over 50% of the total exergy destruction in the overall cycle. The combined cycle efficiency and its power output were maximized at an intermediate pressure-ratio, and increased sharply up to two reheat-stages and more slowly thereafter.

A schematic diagram of a combined Brayton/Rankine power cycle with reheat is shown in Fig.3-4. The gas turbine is shown as a topping plant, which forms the high-temperature loop, whereas the steam plant forms the low-temperature loop. The connecting link between the two cycles is the heat-recovery steam generator (HRSG) working on the exhaust of the gas turbine. A gas-turbine cycle consists of an air compressor (AC), a combustion chamber (CC) and a reheat gas-turbine (RGT). The turbine's exhaust-gas goes to a heat-recovery steam-generator to generate superheated steam. That steam is used in a standard steam power-cycle, which consists of a turbine (ST), a condenser (C) and a pump (P). Both the gas and steam turbines drive electric generators.

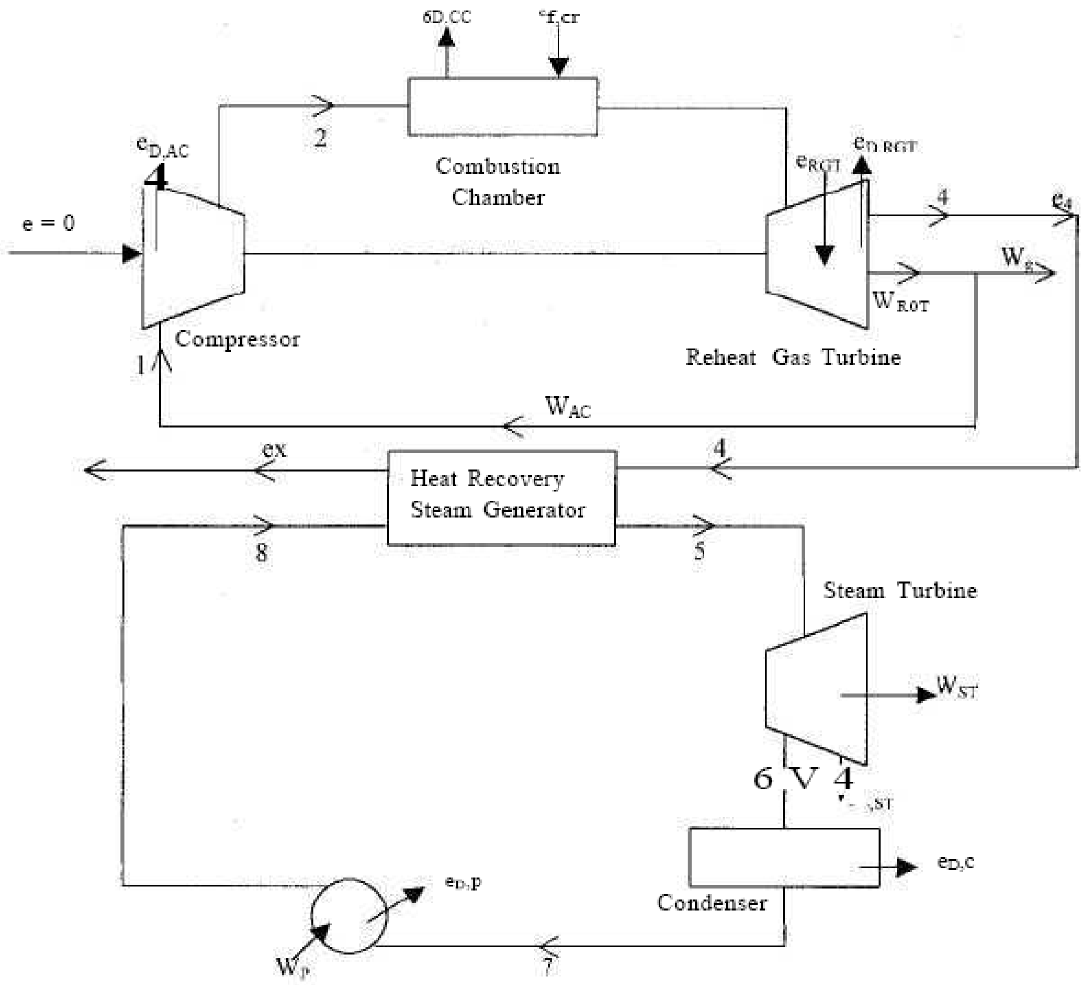

Fig.2-4 Schematic diagram of the combined Brayton/Rankine power cycle with reheat.

They conclude that an improved second-law analysis of the combined power-cycle with reheat has shown the importance of the parameters examined. The analysis has included the exergy destruction in the components of the cycle and an assessment of the effects of pressure ratio, temperature ratio and number of reheat stages on the cycle performance. The exergy balance or second-law approach presented facilitates the design and optimization of complex cycles by pinpointing and quantifying the losses. By placing reheat in the expansion process, significant increases in specific power output and efficiency were obtained. The gains are substantial for one and two reheats, but progressively smaller for subsequent stages. It is interesting to note that specific power output (per unit gas flow) increases by a factor of 2.5 for the two reheats. This may well justify the additional capital cost of the reheat system. Reheating by increasing the specific power-output reduces the sensitivity of the cycle to component losses.

CHAPTER THREE

FUNDAMENTALS OF EXERGY ANALYSIS

3.1 EXERGY: A MEASURE OF WORK POTENTIAL

The second law of thermodynamics has proved to be a very powerful tool in the optimization of complex thermodynamic systems. In this chapter which is based on [4], examination of the performance of devices in light of the second law of thermodynamics had been carried out. Discussion has started with the introduction of exergy (also called availability), which is the maximum useful work that could be obtained from the system at given state in a specific environment, and continue with the *reversible work*, which is the maximum useful work that can be obtained as a system undergoes a process between two specified states. Next discussed the *irreversibility* (also called the exergy destruction or lost work), which is the wasted work potential during a process as a result of irreversibilities, and defining a *second-law efficiency*. Then developed the exergy balance relation and apply it to closed system and control volumes.

3.2 EXERGY WORK POTENTIAL OF ENERGY

When a new energy source, such as geothermal well, is discovered the first thing the explorers do is estimate the amount of energy contained in the source, this information alone, however is of little value in deciding whether to build a power plant on that site. What we really need to know is the work potential of the source that is, the amount of energy we can extract as useful work. The rest of the energy will eventually be discarded as waste energy and is not worthy of our consideration. Thus it would be very desirable to have a property to enable us to determine the useful work potential of a given amount of energy at some specified state. This property is *exergy*, which is also called the *availability or available* energy.

The work potential of the energy contained in a system at specified state is simply the maximum useful work that can be obtained from the system. You will recall that the work done during a process depends on the initial state, the final state and the process path. That is,

Work = f (initial state, process path, final state)

In an exergy analysis, the *initial state* is specified, and thus it is not a variable. The work output is maximized when the process between two specified states is executed in a *reversible manner*; therefore all the irreversibilities are disregarded in determining the work potential. Finally, the system must be in *dead state* at the end of the process to maximize the work output.

A system is said to be in dead state when it is in thermodynamic equilibrium with the environment. At the dead state, a system is at the temperature and pressure of it is environment (in thermal and mechanical equilibrium) it has no kinetic or potential energy relative to the environment (zero velocity and zero elevation above a reference level) and it does not react with the environment (chemically inert). Also, there are no unbalances magnetic, electrical and surface tension effects between the system and it is surroundings, if these are relevant to the situation at hand. The properties of a system at the dead state are denoted by subscript zero, for example, $P_0, T_0, h_0,\ u_0$ and s_0. Unless specified otherwise, the dead state temperature and pressure are assumed to be T_0 = 40˚C and P_0 = 0.966 bar. A system has zero availability at the dead state.

Distinction should be made between the *surrounding, immediate surrounding* and the *environment*. By definition, surrounding are every thing out side the system boundaries. The immediate surrounding refer to the portion of the surrounding that is affected by the process, and the environment refer to the region beyond the immediate surrounding whose properties are not affected by the process at any point. Therefore, any irreversibility during a process occurs within the system and its immediate surrounding, and the environment is free of any irreversibilities.

The notion that a system must go to the dead state at the end of the process to maximize the work output can be explain as follows; if the system temperature at the final state is greater than (or less than) the temperature of the environment it is in, we can always produce additional work by running a heat engine between these two temperature levels. If the final pressure is greater than (or less than) the pressure of the environment we can still obtain work by letting the system expand to the pressure of the environment. If the final velocity of the system is not zero, we can catch that extra kinetic energy by a turbine and converted it to rotating shaft work, and so on. No work can be produced from a system that is initially at the dead state. The atmosphere around us contains a tremendous amount of energy. However, the atmosphere is in the dead state, and the energy it contains has no work potential.

Therefore, we conclude that a system will deliver the maximum possible work as it undergoes a reversible process from the specified initial state to the state of its environment, that is, the dead state. This represents the useful work potential of the system at the specified state and is called exergy. It is important to realize that exergy does not represent the amount of work that a work producing devise will actually deliver upon installation. Rather, it represents the *upper limit on the amount of work a devise can deliver without violating any thermodynamic laws*. There will always be a difference, large or small, between exergy and the actual work delivered by a device. This difference represents the room engineers have for improvement.

Note that the exergy of a system at a specified state depends on the conditions of the environment (the dead state) as well as the properties of the system. Therefore, exergy is a property of the system environment combination and not on the system alone. Altering the environment is another way of increasing exergy, but it is definitely not an easy alternative.

3.3 REVERSIBLE WORK AND IRREVERSIBILITY

The property exergy serves as a valuable tool in determining the quality of energy and comparing the work potentials of different energy sources or systems. The evaluation of exergy a lone, however is not sufficient for studying engineering devices operating between two fixed states. This is because when evaluating exergy, the final state is always assumed to be the dead state, which is hardly ever the case for actual engineering systems; the isentropic efficiencies are also of limited use because the exit state of the model (isentropic) process is not the same as the actual exit state.

In this section we describe two quantities that are related to the actual initial and final state of process and serve as valuable tools in the thermodynamic analysis of components or systems. These two quantities are the reversible work and irreversibility (or exergy destruction). But first we examine the surrounding work, which is the work done by or against the surroundings during a process.

The work done by work producing devices is not always entirely in a useable form. For example, when a gas in a piston- cylinder device expands, part of the work done by the gas is used to push the atmospheric air out of the way of the piston, this work which cannot be recovered and utilized for any useful purpose, is equal to the atmospheric pressure P_0 times the volume change of the system,

$$W_{surr} = P_0(V_2 - V_1) \tag{3-1}$$

The difference between the actual work W and the surrounding work W_{surr} is called the useful work W_u:

$$W_u = W - W_{Surr} = W - P_0(V_2 - V_1) \tag{3-2}$$

When a system is expanding and doing work, part of the work done is used to overcome the atmospheric pressure and thus W_{surr} represents a loss. When a system is compressed, however the atmospheric pressure helps the compression process, and thus W_{surr} represents again.

Note that the work done by or against the atmospheric pressure has significance only for systems whose volume changes during the process (systems that involve moving boundary work). It has no significance for cyclic devices and systems whose boundaries remain fixed during a process such as rigid tanks and steady flow devices (turbines, compressors, nozzles, heat exchangers, nozzles, heat exchangers, etc.)

Reversible work W_{rev} is defined as the maximum amount of useful work that can be produced (or the minimum work that needs to be supplied) as a system undergoes a process between the specified initial and final states. This is the useful work output (or input) obtained (or expended) when the process between the initial and final states is executed in a totally reversible manner. When the final state is the dead state, the reversible work equal exergy. For process that required work, reversible work represents the minimum amount of work necessary to carry out that process. For convenience in presentation, the term work is used to denote both work and power throughout this research.

Any different between the reversible work W_{rev} and the useful work W_u is due to the irreversibilities present during the process, and this difference is called irreversibility *I*. it is expressed as:

$$I = W_{rev,out} - W_{u,out} \qquad \text{or} \qquad I = W_{u,in} - W_{rev,in} \tag{3-3}$$

The irreversibility is equivalent to the exergy destroyed, discuss later for a totally reversible process, the actual and reversible work terms are identical, and thus irreversibility is zero. This is expected since totally reversible processes generate no entropy. Irreversibility is a positive quantity for all actual (irreversible) processes since $W_{rev} \geq W_u$ for work producing devices and $W_{rev} \leq W_u$ for work consuming devices.

Irreversibility can be viewed as the wasted work potential or the lost opportunity to do work. It represents the energy that could have been converted to work but was not.

The smaller the irreversibility associated with a process, the greater the work that will be produced (or the smaller the work that will be consumed). The performance of a system can be improved by minimizing the irreversibility associated with it.

3.4 SECOND-LAW EFFICIENCY, η_{II}

The thermal efficiency and the coefficient of performance for devices had been defined as a measure of performance. They were defined on the basis of the first law only, and they are sometimes referred to as the first-law efficiencies. The first law efficiency, however, make no reference to the best possible performance, and thus it may be misleading. Consider two heat engines, both having a thermal efficiency of 30 percent, one of the engines (engine A) is supplied with the heat from a source at 600 K, and the other one (engine B) from a source at 1000K. Both engines reject heat to a medium at 300K. At first glance, both engines seem to convert to work the same fraction of heat that they receive; thus they are performing equally well. When we take a second look at these engines in light of the second law of thermodynamics, however we see totally different picture. These engines, at best, can perform as reversible engine, in which case their efficiencies would be

$$\eta_{rev,A} = (1 - \frac{T_L}{T_H})_A = 1 - \frac{300K}{600K} = 50\%$$

$$\eta_{rev,B} = (1 - \frac{T_L}{T_H})_B = 1 - \frac{300K}{1000K} = 70\%$$

Now it is becoming apparent that engine B has a greater work potential available to it (70 percent of the heat supplied as compared to 50 percent fore engine A), and thus should do a lot better than engine A. therefore, we can say that engine B is performing poorly relative to engine A even though both have the same thermal efficiency.

It is obvious from this example that the first law efficiency alone is not a realistic measure of performance of engineering devices. To overcome this efficiency, we define a ***second law efficiency*** η_{II} as the ratio of the actual thermal efficiency to the maximum possible (reversible) thermal efficiency under the same conditions:

$$\eta_{II} = \eta_{th} / \eta_{th,rev} \quad \text{(heat engines)} \qquad (3\text{-}4)$$

Based on this definition, the second law efficiencies of the two heat engines discussed above are

$$\eta_{II} = 0.30 \big/ 0.50 = 0.60 \quad \text{and} \qquad \eta_{II} = 0.30 \big/ 0.70 = 0.43$$

That is engine A is converting 60 percent of the available work potential to useful work. This ratio is only 43 percent for engine B.

The second law efficiency can also be expressed as the ratio of the useful work output and the maximum possible (reversible) work output:

$$\eta_{II} = W_u / W_{rev} \text{ (work producing devices)} \tag{3-5}$$

This definition is more general since it can be applied to processes (in turbines, piston-cylinder devices, etc.) as well as to cycles. Note that the second law efficiency cannot exceed 100 percent.

We can also define a second law efficiency for work consuming noncyclic (such as compressors) and cyclic (such as refrigerators) devices as the ratio of the minimum (reversible) work input to the useful work input:

$$\eta_{II} = W_{rev} / W_u \text{ (work consuming devices)} \tag{3-6}$$

For cyclic devices such as refrigerators and heat pumps, it can also be expressed in terms of the coefficients of performance as

$$\eta_{II} = COP / COP_{rev} \text{ (refrigerators and heat pumps)} \tag{3-7}$$

Again, because of the way we defined the second law efficiency, its value can not exceed 100 percent. In the above relations, the reversible work W_{rev} should be determined by using the same initial and final states as in the actual process.

The definitions above for the second law efficiency do not apply to devices that are not intended to produce or consume work. Therefore, we need more general definition. However, there is no agreement on a general definition of the second law efficiency, and thus a person may encounter different definitions for the same device. The second law efficiency is intended to serve as the measure of approximation to reversible operation, and thus its value should range from zero in the worst case (complete destruction of exergy) to one in the best case (no destruction of exergy). With this in mined, we defined the second law efficiency of a system during a process as

η_{II} = exergy recovered/ exergy supplied

=1-exergydestroyed/exergysupplied (3-8)

Therefore, when determining the second law efficiency, the first thing we need to do is determining how much exergy or work potential is consumed during a process. In reversible operation, we should be able to recover entirely the exergy supplied during the process, and the irreversibility in this case should be zero. The second law efficiency will be zero when we recover none of the exergy supplied to the system. Note that exergy can be supplied or recovered at various amount in various forms such as heat, work, kinetic energy, potential energy, internal energy and enthalpy. Some times there are differing (though valid) opinions on what constitutes supplied exergy, and this causing differing definitions for second law efficiency. At all times, however, the exergy recovered and the exergy destroyed (the irreversibility) must add up to the exergy supplied. Also we need to define the system precisely in order to identify correctly any inter action between the system and its surroundings.

For a heat engine, the exergy supplied is the decrease in the exergy of the heat transfer to the engine, which is the difference between the exergy of the heat supplied and the exergy of the heat rejected. (The exergy of the heat rejected at the temperature of the surrounding is zero). The net work output is the recovered exergy.

For refrigerator or heat pump, the exergy supplied is the work input *W* since the work supplied to a cyclic device is entirely available. The recovered exergy is the exergy of the heat transferred to the high temperature medium (which is the reversible work) for a heat pump, and the exergy of the heat transferred from the law temperature medium for a refrigerator.

For a heat exchanger with two unmixed fluid streams, normally the exergy supplied is the decrease in the exergy of the high temperature fluid stream, and the exergy recovered is the increase in the exergy of the lower temperature fluid stream.

3.5 EXERGY CHANGE OF A SYSTEM

The property exergy is the work potential of a system in a specified environment and represent the maximum amount of useful work that can be obtained as the system is brought to equilibrium with the environment. Unlike energy, the value of exergy depends on the state of the environment as well as the state of the system. Therefore, exergy is combination property. The exergy of a system that is in equilibrium with its environment is zero. The state of the environment is refer to as the (dead state) since the system is practically "dead" (can not do any work) from a thermodynamic point of view when it reaches that state.

In this section we will limit the discussion to *thermo-mechanical exergy,* and thus disregard any mixing and chemical reaction. Therefore, a system at this "restricted dead state" will be at the temperature and pressure of the environment and it will have on kinetic or potential energies relative to the environment.

Below we develop relations for the exergies and exergy changes for a fixed mass and a flow stream.

3.5.1 Exergy of a Fixed Mass:

3.5.1.1 Non flow (or closed system) exergy

In general, internal energy consists of *sensible, latent, chemical* and *nuclear* energies. However, in the absence of any chemical or nuclear reaction, the chemical and nuclear energies can be disregarded and the internal energy can be considered to consist of only sensible and latent energies that can be transferred to or from a system as heat whenever there is a temperature difference a cross the system boundary. The second law of thermodynamics states that heat cannot be converted to work entirely, and thus the work potential of internal energy must be less than the internal energy itself. But how much less?

To answer that question, we need to consider a stationary closed system at a specified state that undergoes a reversible process to the state of the environment (that is, the final temperature and pressure of the system should be T_0 and P_0, respectively). The useful work driven during this process is exergy of the system at its initial state.

Consider a piston-cylinder device that contains a fluid of mass *m* at temperature *T* and pressure *P*. the system (the mass inside the cylinder) has a volume V, internal energy *U*, and entropy *S*. the system is now allowed to undergo a differential change of state during which the volume change by a differential amount *dV* and heat is transferred in the differential amount of *dQ*. Taking the direction of heat and work transfers to be from the system (heat and work output), the energy balance for the system during this differential process can be expressed as

$$\partial E_{in} - \partial E_{out} = \partial E_{system} \qquad (3\text{-}9)$$

Net energy transfer *changes in internal. Kinetic*

by heat work and mass *potential ,etc. energies*

Since the only form of energy the system contains is the internal energy, and the only forms of energy transfer a fixed mass can involve are heat and work. Also, the only form of work a simple compressible system can involve during a reversible process is the boundary work, which is given to be $\partial W = P\,dV$.

When the direction of work is taken to be from the system (otherwise it would be - PdV). The pressure P in the PdV expression is the absolute pressure, which is measured from the absolute zero. Any useful work delivered by a piston-cylinder device is due to the pressure above the atmospheric level. Therefore,

$$\partial W = PdV = (P - P_0)\partial V + P_0 \partial V = \partial W_{b,useful} + P_0 \partial V \qquad (3\text{-}10)$$

A reversible process cannot involve any heat transfer through a finite temperature difference, and thus any heat transfer between the system at temperature T and its surrounding at T_0 must be occur through a reversible heat engine. Noting that $\partial S = \partial Q/T$ for a reversible process, and the thermal efficiency of a reversible heat engine operating between the temperatures of T and T_0 is $\eta_{th} = 1 - T_0/T$, the differential work produced by the engine as a result of this heat transfer is

$$\partial W_{HE} = (1 - T_0/T)\partial Q = \partial Q - (T_0/T)\partial Q$$

$$= \partial Q - (-T_0 \partial S) \longrightarrow \partial Q = \partial W_{HE} - T_0 \partial S \qquad (3\text{-}11)$$

Substituting the ∂W and ∂Q *expression* in Eqs. 3-10 and 3-11 into the energy balance relation Eqs. 3-11 gives after rearranging,

$$\partial W_{total,useful} = \partial W_{HE} + \partial W_{b,useful} = -dU - P_0 dV + T_0 dS$$

Integrating from the given state (no subscript) to the dead state (0 subscript) we obtain

$$W_{totaluseful} = (U - U_0) + P_0(V - V_0) - T_0(S - S_0) \qquad (3\text{-}12)$$

Where $W_{total\ useful}$ is the total useful work delivered as the system undergoes a reversible process from the given state to the dead state. Which is exergy by definition.

A closed system, in general, may possess kinetic and potential energies, and the total energy of a closed system is equal to the sum of its internal, kinetic, and potential energies. Noting that kinetic and potential energies themselves are form of exergy, the exergy of a closed system of mass *m* is

$$X = (U - U_0) + P_0(V - V_0) - T_0(S - S_0) + mC^2/2 + mgz \qquad (3\text{-}13)$$

On a unit mass basis, the closed system (or nonflow) exergy Φ is expressed as

$$\Phi = (u - u_0) + P_0(v - v_0) - T_0(s - s_0) + C^2/2 + gz \qquad (3\text{-}14)$$

$$= (e - e_0) + P(v - v_0) - T_0(s - s_0)$$

Where u_0, v_0 and s_0 are the properties of the system evaluated at the dead state. Note that the exergy of a system is zero at the dead state since $e = e_0$, $v = v_0$, and $s = s_0$ at that state.

The exergy change of a closed system during a process is simply the difference between the final and initial exergies of the system,

$$\Delta X = X_2 - X_1 = m(\Phi_2 - \Phi_1) = (E_2 - E_1) + P_0(V_2 - V_1) - T_0(S_2 - S_1) \qquad (3\text{-}15)$$
$$= (U_2 - U_1) + P_0(V_2 - V_1) - T_0(S_2 - S_1) + m(C_2^2 - C_1^2)/2 + mg(Z_2 - Z_1)$$

Or, on a unit mass basis,

$$\Delta\Phi = \Phi_2 - \Phi_1 \qquad (3\text{-}16)$$
$$= (u_2 - u_1) + P_0(v_2 - v_1) - T_0(s_2 - s_1) + (C_2^2 - C_1^2)/2 + g(z_2 - z_1)$$
$$= (e_2 - e_1) + P_0(v_2 - v_1) - T_0(s_2 - s_1)$$

For stationary closed systems, the kinetic and potential energy terms drop out. When the properties of a system are not uniform, the exergy of the system can be determined by integration from

$$X_{system} = \int \Phi \partial m = \int v\Phi\rho dV \qquad (3\text{-}17)$$

Where V is the volume of the system and ρ is density.

Note that exergy is a property, and the value of a property does not change unless the state changes. Therefore, the exergy change of a system is zero if the state of the system or the environment does not change during the process. For example the exergy change of steady flow devices such as nozzles, compressors, turbines, pumps and heat exchangers in a given environment is zero during steady operation.

The exergy of a closed system is either positive or zero. It is never negative. Even a medium at low temperature ($T < T_0$) and/or low pressure ($P < P_0$) contains exergy since a cold medium can serve as the heat sink to a heat engine that absorbs heat from the environment at T_0 , and an evaluated space makes it possible for the atmospheric pressure to move a piston and do useful work.

3.5.1.2 Exergy of a Flow Stream: Flow (or Stream Exergy)

We know that a flowing fluid has an additional form of energy, called the *flow energy,* which is the energy needed to maintain flow in a pipe or duct, and is expressed as $w_{flow} = Pv$ where v is the specific volume of the fluid, which is equivalent to the volume change of a unit mass of the fluid as it is displaced during flow. The flow work is essentially the boundary work done by a fluid on the fluid downstream, and thus the exergy of flow work is equivalent to the exergy of the boundary work, which is the boundary work in excess of the work done against the atmospheric air at P_0 to displace it by a volume v. noting that the flow work is Pv and the work done against the atmospheric is P_0v, the exergy of flow energy can be expressed as

$$x_{flow} = Pv - P_0 v = (P - P_0)v \tag{3-18}$$

Therefore, the exergy associated with flow energy is obtained by replacing the pressure P in the flow work relation by the pressure in excess of the atmospheric pressure, $P - P_0$. Then the exergy of a flow stream is determined simply adding the flow exergy relation above to the exergy relation in equation (3-16) for a nonflow fluid

$$x_{flowingfluid} = x_{nonflowingfluid} + x_{flow} \tag{3-19}$$

$$= (u - u_0) + P_0(v - v_0) - T_0(s - s_0) + C^2/2 + gz + (P - P_0)v$$

$$= (u + Pv) - (u_0 + Pv_0) - T_0(s - s_0) + C^2/2 + gz$$

$$= (h - h_0) - T_0(s - s_0) + C^2/2 + gz$$

The final expression is called flow (or stream) exergy and is denoted by ψ

$$\textit{Flowexergy:} \quad \psi = (h - h_0) - T_0(s - s_0) + C^2/2 + gz \tag{3-20}$$

Then the exergy change of a fluid stream as it undergoes a process from state 1 to state 2 becomes

$$\Delta\psi = \psi_2 - \psi_1 = (h_2 - h_1) - T_0(s_2 - s_1) + (C_2^2 - C_1^2)/2 + g(z_2 - z_1) \tag{3-21}$$

For fluid stream with negligible kinetic and potential energies, the kinetic and potential energy terms drop out.

Note that the exergy change of a closed system or a fluid stream represents the maximum amount of useful work that can be done (or the minimum amount of useful work that needs to be supplied if it is negative) as the system changes from state 1 to state 2 in a specified environment, and represent the reversible work W_{rev} . It is independent of the type of process executed, the kind of system used and the nature of energy interaction with the surrounding. Also note that the exergy of a closed system cannot be negative, but the exergy of a flow stream can at pressures below the environment pressure P_0.

3.6 THE DECREASE OF EXERGY PRINCIPLE AND EXERGY DESRUCTION

The *conservation of energy principle* indicates that energy cannot be created or destroyed. Also the *increase of entropy principle*, which can be regarded as one of the statements of the second law, indicate that entropy can be created but cannot be destroyed. That is entropy generation S_{gen} must be positive (actual processes) or zero (reversible processes), but it cannot be negative. Now we are about to establish an alternative statement of the second law of thermodynamics, called the *decrease of exergy principle*, which is the counterpart of the increase of entropy principle.

Consider an isolated system shown in figure (3-1). By definition, no heat, work, or mass can cross the boundaries of an isolated system, and thus there is no energy and entropy transfer. Then the energy and entropy balances for an isolated system can be expressed as

Energy balance: $\overset{0}{E_{in}} - \overset{0}{E_{out}} = \Delta E_{system} \longrightarrow 0 = E_2 - E_1$

Entropy balance: $\overset{0}{S_{in}} - \overset{0}{S_{out}} + S_{gen} = \Delta S_{system} \longrightarrow S_{gen} = S_2 - S_1$

Multiplying the second relation by T_0 and subtracting it from the first one gives

$$-T_0 S_{gen} = E_2 - E_1 - T_0(S_2 - S_1) \tag{3-22}$$

From equation 3-15 we have

$$X_2 - X_1 = (E_2 - E_1) + \overset{0}{P_0(V_2 - V_1)} - T_0(S_2 - S_1) \tag{3-23}$$

$$= (E_2 - E_1) - T_0(S_2 - S_1)$$

Since $V_2 = V_1$ for an isolated system (it cannot involve any moving boundary and thus any boundary work). Combining Eqs. 3-22 and 3-23 gives

$$T_0 S_{gen} = X_2 - X_1 \tag{3-24}$$

Since T_0 is the absolute temperature of the environment and thus a positive quantity, $S_{gen} \geq 0$, and thus $T_0 S_{gen} \geq 0$. Then we conclude that

$$\Delta X_{isolated} = (X_2 - X_1)_{isolated} \leq 0 \tag{3-25}$$

This equation can be expressed as the exergy of an isolated system during a process always decreases or, in the limiting case of a reversible process, remains constant. In other words, it never increases and exergy is destroyed during an actual process. This is known as the **decrease of exergy principle**. For an isolated system, the decrease in exergy equal the exergy destroyed.

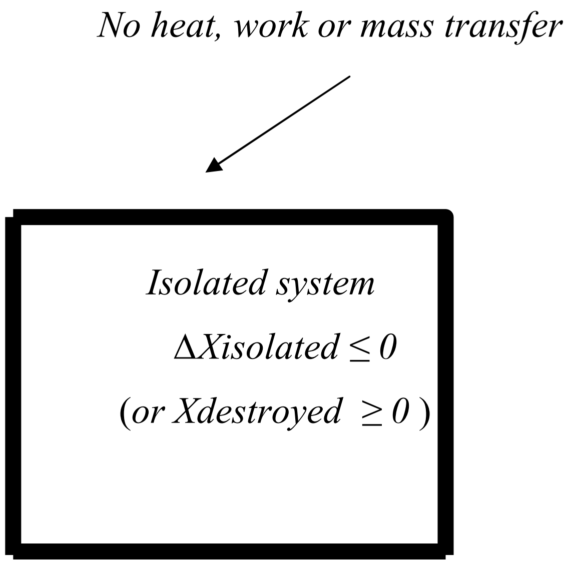

Figure (3-1): The isolated system considered in the development of the decrease of exergy principle.

3.6.1 Exergy Destruction

Irreversibilities such as friction, mixing, chemical reaction, heat transfer through a finite temperature difference, unrestrained expansion, non-quasi-equilibrium compression or expansion always *generate entropy*, and any thing that generates entropy always *destroys exergy*. The exergy destroyed is proportional to the entropy generated, as can be seen from Eq. 3-24, and expressed as

$$X_{destroyed} = T_0 S_{gen} \geq 0 \qquad (3\text{-}26)$$

Note that exergy destroyed is a positive quantity for any actual process and becomes zero for a reversible process. Exergy destroyed represents the lost work potential and is also called the *irreversibility* or *lost work*.

Equation 2-35 and 2-26 for the decrease of exergy and the exergy destruction are applicable to any kind of system undergoing any kind of process since any system and its surrounding can be enclosed by a sufficiently large arbitrary boundary cross which there is no heat, work, and mass transfer and thus any system and its surroundings constitute an isolated system.

No actual process is truly reversible, and thus some exergy is destroyed during a process. Therefore, the exergy of the universe, which can be considered to be an isolated system, is continuously decreasing. The more irreversible a process is the larger the exergy destruction during that process. No exergy is destroyed during a reversible process ($X_{destroyed,rev} = 0$).

The decrease of exergy principle does not imply that the exergy of a system cannot increase. The exergy change of a system can be positive or negative during a process, but exergy destroyed cannot be negative. The decrease of exergy principle can be summarized as follows:

$$X_{destroyed} = \begin{cases} > 0 & \textit{Irreversible process} \\ = 0 & \text{Re}\textit{versible process} \\ < 0 & \text{Im}\textit{possible process} \end{cases} \tag{3-27}$$

This relation serves as an alternative criterion to determine whether a process is reversible, irreversible or impossible.

3.7 EXERGY BALANCE: CLOSED SYSTEMS

The nature of exergy is opposite to that of entropy in that exergy can be destroyed, but it cannot be created. Therefore, the exergy change of a system during a process is less than the exergy transfer by an amount equal to the exergy destroyed during the process within the system boundaries. Then the decrease of exergy principle can be expressed as

$$\begin{pmatrix} Total \\ exergy \\ entring \end{pmatrix} - \begin{pmatrix} Total \\ exergy \\ leaving \end{pmatrix} - \begin{pmatrix} Total \\ exergy \\ destroyed \end{pmatrix} = \begin{pmatrix} Change\ in\ the \\ total\ exergy \\ of\ the\ system \end{pmatrix}$$

Or

$$X_{in} - X_{out} - X_{destroyed} = \Delta X_{system} \qquad (3\text{-}28)$$

This relation is referred to as the **exergy balance** and can be stated as *the exergy change of a system during a process is equal to the difference between the net exergy transfer through the system boundary and the exergy destroyed within the system boundaries as a result of irreversibilities*.

We mentioned earlier that exergy can be transferred to or from a system by heat, work, and mass transfer. Then the exergy balance for any system undergoing any process can be expressed more explicitly as

General:

$$\underbrace{(X_{in} - X_{out})}_{\substack{\text{Net exergy transfer} \\ \text{by heat, work, and mass}}} - \underbrace{X_{destroyed}}_{\substack{\text{exergy} \\ \text{destruction}}} = \underbrace{\Delta X_{system}}_{\substack{\text{change} \\ \text{in exergy}}} \quad \text{(kJ)} \qquad (3\text{-}29)$$

Or in the rate form, as

General, rate form:

$$(\dot{X}_{in} - \dot{X}_{out}) - \dot{X}_{destroyed} = \Delta\dot{X}_{system} \text{ (kW)} \qquad (3\text{-}30)$$

Rate of net exergy transfer by heat, work, and mass | *rate of exergy destruction* | *rate of change of exergy*

Where the rates of exergy transfer by heat, work, and mass are expressed as

$\dot{X}_{heat} = (1 - T_0/T)\dot{Q}$, $\dot{X}_{work} = \dot{W}_{useful}$ and $\dot{X}_{mass} = \dot{m}\psi$, respectively, and $\Delta X_{system} = dX_{system}/dt$. The exergy balance can also be expressed per unit mass as

General, unit-mass basis:

$$(x_{in} - x_{out}) - x_{destroyed} = \Delta x_{system} \qquad \text{(kJ/kg)} \qquad (3\text{-}31)$$

Where all the quantities are expressed per unit mass of the system. Note that for a reversible process, the exergy destruction term $X_{destroyed}$ drops out from all of the relations above. Also, it is usually more convenient to find the entropy generation S_{gen} first, and then to evaluate the exergy destroyed directly from Eq. 3-26. That is,

$$X_{destroyed} = T_0 S_{gen} \quad \text{or} \quad \dot{X}_{destroyed} = T_0 \dot{S}_{gen} \qquad (3\text{-}32)$$

When the environment conditions P_0 and T_0 and the end states of the system are specified, the exergy change of the system $\Delta X_{system} = X_2 - X_1$ can be determined directly from Eq. 3-15 regardless of how the process is executed. However, the determination of the exergy transfers by heat, work, and mass requires knowledge of these interactions.

A closed system does not involve any mass flow and thus any exergy transfer by it. Taking the positive direction of heat transfer to be to the system and the positive direction of work transfer to be from the system, the exergy balance for a closed system can be expressed more explicitly as

$$\textit{Closed system: } X_{heat} - X_{work} - X_{destroyed} = \Delta X_{system} \tag{3-33}$$

Or

$$\textit{Closed system: } \sum (1 - \frac{T_0}{T_K}) Q_K - \left[W - P_0 (V_2 - V_1) \right] - T_0 S_{gen} = X_2 - X_1 \tag{3-34}$$

Where Q_K is the heat transfer through the boundary at temperature T_K at location *k.* divide the equation above by the time interval Δt and taking the limit as $\Delta t \longrightarrow 0$ gives the rate form of the exergy balance for a closed system,

Rate form:

$$\sum (1 - \frac{T_0}{T_K}) \dot{Q}_K - \left[\dot{W} - P_0 dV_{system} / dt \right] - T_0 \dot{S}_{gen} = dX_{system} / dt \tag{3-35}$$

Note that the relations above for a closed system are developed by taking the heat transfer to a system and work done by the system to be positive quantities. Therefore, heat transfer from the system and work done on the system will be taken to be negative quantities when using those relations.

The exergy balance relations presented above can be used to determine the *reversible work* W_{rev} by setting the exergy destruction term equal to zero. The work *W* in that case becomes the reversible work. That is, $W = W_{rev}$ when $X_{destroyed} = T_0 S_{gen} = 0$.

Note that $X_{destroyed}$ represents the exergy destroyed within the system boundary only, and not the exergy destruction that may occur outside the system boundary during the process as a result of external irreversibilities. Therefore, a process for which $X_{destroyed} = 0$ is internally reversible but not necessarily totally reversible. The total exergy destroyed during a process can be determined by applying the exergy balance to an *extended system* that includes the system itself and its immediate surrounding where external irreversibilities might be occurring. Also, the exergy change in this case is equal to the sum of exergy changes of the system and the *exergy change* of the immediate surroundings. Note that under steady conditions, the state and thus the exergy of the immediate surroundings (the "buffer zone") at any point will not change during the process, and thus the exergy change of the immediate surroundings will be zero. When evaluating the exergy transfer between an extended system and the environment, the boundary temperature of the extended system is simply taken to be the environment temperature *T0*

For a *reversible process*, the *entropy generation* and thus the *exergy destruction* are *zero*, and the exergy balance relation in this case becomes analogous to the energy balance relation. That is, the exergy change of the system becomes equal to the exergy transfer.

Note that the *energy change* of a system equal the *energy transfer* for any process, but the *exergy change* of a system equals the *exergy transfer* only for *a reversible process*. The *quantity* of energy is always preserved during an actual process (the first law), but the quality is bound to decrease (the second law). This decrease in *quality* is always accompanied by an increase in entropy and a decrease in exergy. When 10 kJ of heat is transferred from a hot medium to a cold one, for example, we will still have 10 kJ of energy at the end of the process, but at a lower temperature, and thus at a lower quality and at a lower potential to do work.

3.8 EXERGY BALANCE: CONTROL VOLUMES

The exergy balance relation for control volumes differ from those for closed systems in that they involve one more mechanism of exergy transfer: *mass flow a cross the boundaries*. As mentioned earlier, mass possesses exergy as well as energy and entropy, and the amounts of these three extensive properties are proportional to the amount of mass (figure 3-2). Again taking the positive direction of heat transfer to be to the system and the positive direction of work transfer to be from the system, the general exergy balance relations (Eqs,3-29 and 3-30) can be expressed for a control volume more explicitly as

$$X_{heat} - X_{work} + X_{mass,in} - X_{mass,out} - X_{destroyed} = (X_2 - X_1)_{CV} \tag{3-36}$$

Or

$$\sum(1 - \frac{T_0}{T_K})Q_K - [W - P_0(V_2 - V_1)] + \sum m_i\psi_i - \sum m_e\psi_e - X_{destroyed} \tag{3-37}$$

$$= (X_2 - X_1)_{CV}$$

Where the subscripts are i = inlet, e = exit, 1 = initial state, and 2 = final state of the control volume. It can also be expressed in the rate form as

$$\sum(1 - \frac{T_0}{T_K})\dot{Q}_K - [\dot{W} - P_0 dV_{cv}/dt] + \sum \dot{m}_i\psi_i - \sum \dot{m}_e\psi_e - \dot{X}_{destroyed} = dX_{cv}/dt \tag{3-38}$$

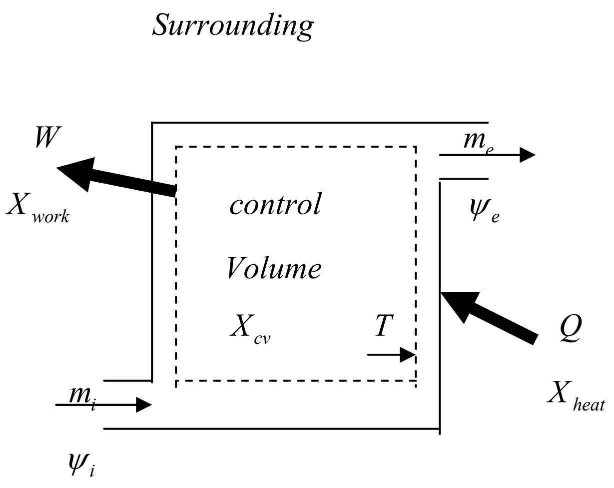

Figure (3-2): Exergy is transferred into or out of a control volume by mass as well as heat and work transfer.

The exergy balance relation above can be stated as *the rate of exergy change within the control volume during a process is equal to the rate of the net exergy transfer through the control volume boundary by heat, work, and mass flow minus the rate of exergy destruction within the boundaries of the control volume*.

When the initial and final states of the control volume are specified, the exergy change of the control volume is $X_2 - X_1 = m_2\Phi_2 - m_1\Phi_1$.

3.8.1 Exergy Balance for Steady-Flow System

Most control volumes encountered in practice such turbines, compressors, nozzles, diffusers, heat exchangers, pipes, and ducts operate steadily, and thus they experience no changes in their mass, energy, entropy, and exergy contents as well as their volumes. Therefore, $dV_{CV}/dt = 0$ and $dX_{CV}/dt = 0$ for such systems, and the amount of exergy entering a steady-flow system in all forms (heat, work, mass transfer) must be equal to the amount of exergy leaving plus the exergy destroyed. Then the rate form of the general exergy balance (Eq.3-38) reduces for a steady-flow process to

Steady-flow:

$$\sum(1-\frac{T_0}{T_K})\dot{Q}_K - \dot{W} + \sum \dot{m}_i\psi_i - \sum \dot{m}_e\psi_e - \dot{X}_{destroyed} = 0 \qquad (3\text{-}39)$$

For a single-stream (one-inlet, one-exit) steady-flow device, the relation above further reduce to

Single-stream:

$$\sum(1-\frac{T_0}{T_K})\dot{Q}_K - \dot{W} + \dot{m}(\psi_1 - \psi_2) - \dot{X}_{destroyed} = 0 \qquad (3\text{-}40)$$

Where $\dot{m}$ is the mass flow rate and the change in the flow exergy is given by Eq.3-21 as

$$\psi_1 - \psi_2 = (h_1 - h_2) - T_0(s_1 - s_2) + (C_1^2 - C_2^2)/2 + g(z_1 - z_2)$$

Dividing Eq. 3-40 by $\dot{m}$ gives the exergy balance on a unit-mass basis as

Per-unit mass:

$$\sum(1-\frac{T_0}{T_K})q_k - w + (\psi_1 - \psi_2) - x_{destroyed} = 0 \text{ (kJ/kg)} \qquad (3\text{-}41)$$

Where $q = \dot{Q}/\dot{m}$ and $w = \dot{W}/\dot{m}$ are the heat transfer and work done per unit mass of the working fluid, respectively.

For the case of an adiabatic single-stream device with no work interactions, the exergy balance reaction further simplifies to $\dot{X}_{desttroyed} = \dot{m}(\psi_1 - \psi_2)$, which indicate that the specific exergy of the fluid must decrease as it flows through a work-free adiabatic device or remain the same $(\psi_1 = \psi_2)$ in the limiting case of a reversible process regardless of the changes in other properties of the fluid.

3.8.2 Reversible Work, W_{rev}

The exergy balance relation presented above can be used to determine the reversible work W_{rev} by setting the exergy destroyed equal to zero. The work W in that case becomes the reversible work. That is, *General:* $W = W_{rev}$ when $X_{destroyed} = 0$ (3-42)

For example, the reversible power for a single-stream steady-flow device is, from Eq. 3-40

Single-stream: $W_{rev} = \dot{m}(\psi_1 - \psi_2) + \sum(1 - \frac{T_0}{T_K})\dot{Q}_K$ (kW) (3-43)

Which reduces for an adiabatic device to

Adiabatic, single stream: $\dot{W}_{rev} = \dot{m}(\psi_1 - \psi_2)$ (3-44)

Note that the exergy destroyed is zero only for a reversible process, and reversible work represents the maximum work output for work producing devices such as turbines and the minimum work input for work consuming devices such as compressors.

3.8.3 Second-Law Efficiency of Steady-Flow Devices, η_{II}

The second-law efficiency of various steady-flow devices can be determined from its general definition, η_{II} = (Exergy recovered) / (Exergy supplied). When the changes in kinetic and potential energies are negligible, the second-law efficiency of an *adiabatic turbine* can be determined from

$$\eta_{11,turb} = w \Big/ w_{rev} = \frac{h_1 - h_2}{\psi_1 - \psi_2} \quad \text{or} \quad \eta_{II,turb} = 1 - \frac{T_O S_{gen}}{\psi_1 - \psi_2} \qquad (3\text{-}45)$$

Where $S_{gen} = S_2 - S_1$. For an *adiabatic compressor* with negligible kinetic and potential energies, the second-law efficiency becomes

$$\eta_{II,comp} = {w_{rev,in}}\Big/{w_{in}} = \frac{\psi_2 - \psi_1}{h_2 - h_1} \qquad \text{or} \qquad \eta_{II,comp} = 1 - \frac{T_0 S_{gen}}{h_2 - h_1} \tag{3-46}$$

Where again $S_{gen} = S_2 - S_1$.

For an adiabatic heat exchanger with two unmixed fluid streams (figure 3-3), the exergy supplied is the decrease in the exergy of the hot stream, and the exergy recovered is the increase in the exergy of the cold stream, provided that the cold stream is not at a lower temperature than the surroundings. Then the second-law efficiency of the heat exchanger becomes

$$\eta_{II,HX} = \frac{\dot{m}_{cold}(\psi_4 - \psi_3)}{\dot{m}_{hot}(\psi_1 - \psi_2)} \quad \text{or} \quad \eta_{II,HX} = 1 - \frac{T_0 \dot{S}_{gen}}{\dot{m}_{hot}(\psi_1 - \psi_2)} \tag{3-47}$$

where $\dot{S}_{gen} = \dot{m}_{hot}(s_2 - s_1)/\dot{m}_{cold}(s_4 - s_3)$.perhaps you are wondering what happens if the heat exchanger is not adiabatic; that is, it is losing some heat to its surroundings at T_0 If the temperature of the boundary (the outer surface of the heat exchanger) T_b is equal T_0, the definition above still holds (except the entropy generation term needs to be modified if the second definition is used). However, if $T_b > T_0$, then the exergy of the lost heat at the boundary should be included in the recovered exergy. Although no attempt is made in practice to utilize this exergy and it is allowed to be destroyed, the heat exchanger should not be held responsible for this destruction, which occurs outside its boundaries. If we are interested in the exergy destroyed during the process, not just within the boundaries of the device, then it makes sense to consider an extended system that includes the immediate surroundings of the device such that the boundaries of the new enlarged system are at T_0. The second-law efficiency of the extended system will reflect the effects of the irreversibilities that occur within and just outside the device.

An interesting situation arises when the temperature of the cold stream remains below the temperature of the surroundings at all times. In that case the exergy of the cold stream actually decrease instead of increasing. In such cases it is better to define the second-law efficiency as the ratio of the sum of the exergies of the outgoing streams to the sum of the exergies of the incoming streams.

For an adiabatic mixing chamber where a hot stream 1 is mixed with a cold stream 2, forming a mixture 3, the exergy supplied is the sum of the exergies of the hot and cold streams, and the exergy recovered is the exergy of the mixture. Then the second law efficiency of the mixing chamber becomes

$$\eta_{II.mix} = \frac{\dot{m}_3 \psi_3}{\dot{m}_1 \psi_1 + \dot{m}_2 \psi_2} \quad \text{or} \quad \eta_{II,mix} = 1 - \frac{T_0 \dot{S}_{gen}}{\dot{m}_1 \psi_1 + \dot{m}_2 \psi_2} \tag{3-48}$$

Where $\dot{m}_3 = \dot{m}_1 + \dot{m}_2$ and $\dot{S}_{gen} = \dot{m}_3 s_3 - \dot{m}_2 s_2 - \dot{m}_1 s_1$.

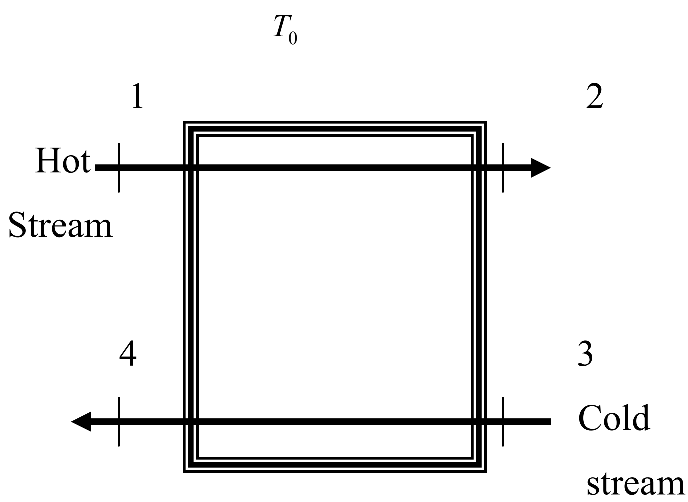

Figure (3-3): A heat exchanger with two unmixed streams.

CHAPTER FOUR

GARRI 1 POWER PLANT

4.1. SITE LOCATION

The power plant had been built in the north of the Khartoum refinery. The Khartoum refinery is a bout 70 km north from Khartoum and the distance between the refinery and the Nile River is a bout 12.5 km. about 5.5 km west of the refinery, the power plant site altitude is about 400 m, there is a highway passing from Khartoum city to northern Sudan, with which the access road of the refinery had been connected. The national highway and access road both are asphalt pavement. There is a railway lining about 3 km west of the Khartoum refinery.

4.2. WEATHER CONDITIONS

Many-year average atmosphere pressure	964.1mbar
Many-year average atmosphere temperature	30.8˚C
Extreme highest atmosphere temperature	47.2˚C
Extreme lowest atmosphere temperature	6.0˚C
Many-year average highest atmosphere temperatures	37.1˚C
Many-year average lowest atmosphere temperature	22.7˚C
Average relative humidity	29%
Many-year average rainfall	162.4mm
Mean wind velocity	3.9m/s
Max. wind velocity	32m/s
All-year wind prevalent direction	north wind
Seismic intensity	class VI

4.3. DESIGN OPERATING CONDITIONS

Ambient temperature	40˚C
Atmospheric pressure	0.966bar
Relative humidity	38%

4.4. PLANT CONFIGURATIONS

The combined cycle power plant consists of two 206B combined cycle, two gas turbine. Two HRSGs and one steam turbine are used per block, as shown in figure (4-1).

Each gas turbine type is PG 6001B; its output is approximately 40MW for ISO condition, the gas turbine is normally operated with double fuel, LPG and light diesel oil. The gas turbine generator, which is driven at 3000 rpm, is with air-cooler.

Each gas turbine exhausts gas lead to its associated HRSG. There is a 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 steam lines of each HRSG are led to the steam turbine. Steam turbine is condensing type with extraction steam.

4.5. OPERATING SCHEME OF THE CCPP

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. Fuel combusted in the combustion chamber before expanding in the turbine.

After expansion the flue 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.

Air and non condensable gases entering the water/steam cycle are collected at the coldest part of the condenser and evacuated. During normal operation the vacuum is maintained with Liquid-Ring vacuum pump.

Two 100% Liquid-Ring vacuum pump are used for start-up evacuation.

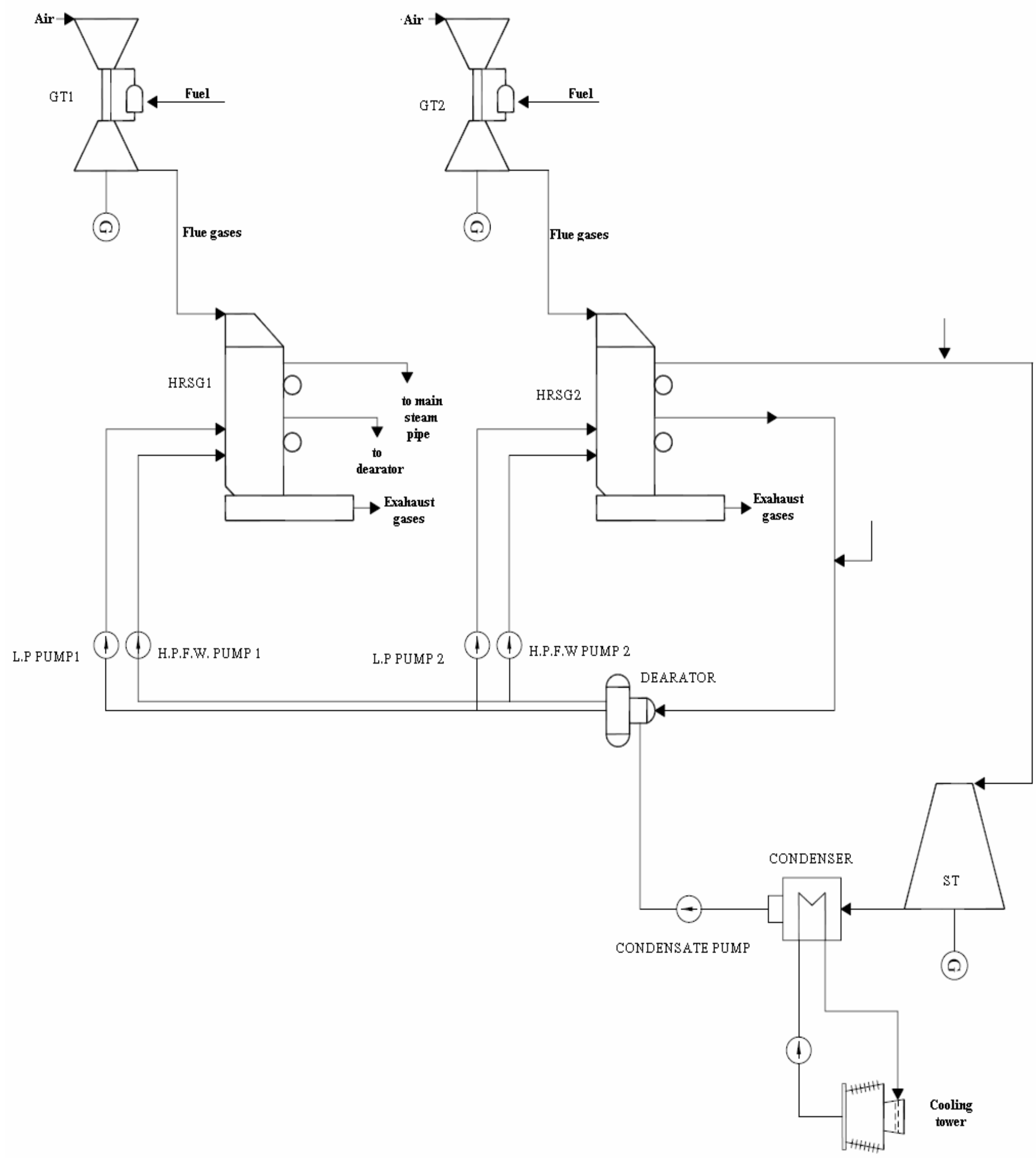

Figure (4-1): GARRI 1 Block configuration.

The condensate and make up water accumulating in the condenser hotwell is delivered by one of the two 100% condensate pumps to the dearator water tank each condensate pump is provided with separate suction lines from the hotwell ensuring a short and direct connection. One pump is in operation, the second serve as standby and is switched on automatically in case of failure of the running pump. The level in hotwell is kept constant by control of the make up feed. A condensate minimum flow check valve is provided to insure minimum flow through the condensate pumps.

The feed water is fed back to the HRSGs by three 50% constant speed HP feed-water pumps and by two 100% constant speed LP feed-water pumps. One pump is at standby and is switched on automatically in case of failure of a running pump. A feed-water pump minimum flow check valve is provided to ensure minimum flow through the feed water pump.

To increase the operation flexibility during start up, shut down and abnormal operating conditions separated bypass stations for HP steam source is provided. The HP bypass stations are designed to accommodate 100% of the maximum steam production into the condenser. Each bypass station consists of an isolation valve, a steam pressure reducing valve, a desuperheating station with the associated measurement, control and production device. Main condensate is used for desuperheating the steam down to the saturation point before entering in the condenser.

The 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.

4.6. GAS TURBINE WITH EXHAUST GAS SYSTEM

4.6.1 Gas Turbine (Frame 6 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.

(1) Low installed cost owing to standardization and to factory assembly and test. This makes the installation of the station easy and keeps the cost per installed kilowatt low because the package power station is quickly ready to be put into operation.

(2) Minimum site investment: the required space and site preparations are minimum. The overall dimensions of the slab type foundation are only 38 x 6 m^2.

(3) Low standby cost: fast start up and shut down reduce conventional standby cost. The power requirements to keep the plant in standby 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 plants 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.

4.6.2 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 damper 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 and important role in boiler start-up and load regulation.

4.7. WATER STEAM CYCLE

Thermodynamic system of Gas-Steam Combined Cycling power station including gas turbine (GT) system, heat recovery steam generator (HRSG) and steam turbine (ST) steam and water system.

Two blocks of Gas-Steam Cycling units are designed for this plant; each block consists of two Gas Turbine, two 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.

4.8. HRSG AND ITS AUXILIARY SYSTEM

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

4.9. THE MAIN STEAM AND BYPASS SYSTEM

The main steam system connects the two HRSGs supply steam to one 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 allow each HRSG to be started independently. The bypass steam is expanded and desuperheated 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 start up.

-steam dump in case of steam turbine load rejection, so that the gas turbine can be kept operating.

4.10. DEAERATOR HEATING STEAM SYSTEM

Feed water is heated and deaerated in a spray type feed-water tank. This horizontal cylindrical tank reduces the water dissolved oxygen content to an acceptable value by the HRSGs. The feed-water heating before start-up is provided by auxiliary boiler steam fuel oil is light diesel oil. But if fuel is LPG, feed-water heating steam before start up is provided also by auxiliary boiler. In normal operation, the heating steam is provided by the HRSG low temperature evaporators or by the HP drum, at low load, and diffused into the water mass by vertical perforated tubes.

4.11. BOILER FEED-WATER SYSTEM

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

4.12. CONDENSATE SYSTEM

Two condensate pumps, one in operation and one standby, are provided for one steam turbine to transits condensate through turbine gland stem cooler to the deaerator. 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 attemperature.

4.13. 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 then collected in the tower basin. The condenser is equipped with a tube cleaning device of the sponge ball type.

4.14. AUXILARY SYSTEMS

Cooling water to auxiliary equipment of gas turbine is from circulating water pump room and return to circulating cooling tower. Steam turbine and gas turbine cooler cooling water, ST generator air cooler cooling water are from main cooling water system and return to the return pipe of cooling water. Service water comes from circulating water pump room and be used to cool pump bearing and return to circulating water pump room. Back water will be discharged to waste water pit, if service water is polluted.

One storage water tank is provided to two units. Make-up water is from one storage water tank to two condenser hotwell. Two deaerator filling water pumps are provided to two units.

Two 100% liquid-ring vacuum pumps are provided to one steam turbine, one in work during normal operation and two may in work during start up.

Each HRSG is provided with two low pressure circulating pumps, one is in operation, one standby.

At the auxiliary boiler outlet there are two lines; one is connected to the LPG heaters for LPG heating during unit start-up and at single cycle operation, another is connected to the deaerator for feed water heating during the unit start-up. There is one auxiliary boiler only for this plant. The fuel of auxiliary boiler is light diesel. In normal operation (combined cycle), the LPG heating steam is coming from steam turbine extraction.

4.19. HEAT RECOVERY STEAM GENERATOR (HRSG)

4.19.1 General

Included in the whole heat recovery steam generator system are boiler inlet duct, HRSG boiler proper, main stack, relevant duct accessories and its control system. HRSG is mainly used to recover waste heat from GT 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 deaerator.

4.19.2 Type of Boiler

The boiler is natural circulation water tube boiler with horizontal flue and vertical helical finned pipe. Located in the horizontal flue are helical finned pipes of heat-transmitting components.

4.19.3 Boiler Performance Features

- *A safe and Reliable Operation*

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

- *Incorporation of the specific feature 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 features of both the natural and forced circulation boilers.

- *Light weight and small size*

The boiler weight and physical dimensions of this type of boiler excel those of analogous types of foreign and home-made natural circulation boilers, and it is also comparable to those of conventional forced circulation boilers.

- *High maneuverability*

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

- *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.

4.19.4 Boiler Inlet Duct, Main Stack and Duct Accessories

Boiler inlet duct is used for leading GT exhaust gas into HRSG which can insure good flow distribution, low flue gas resistance, high sealing quality and high capability to withstand heat shock. Duct accessories include expansion joints at every location and all kinds of interfaces which can insure duct free expansion under the high temperature condition and can meet the requirements of interfaces for boiler supervision and control. The height of 30 m is determined for main stack in consideration of requirements of flue gas discharge.

4.20. STEAM TURBINE

The rotor of the turbine is composed of one speed stage and 18 pressure stages. Twisted blades are fixed in the last three stages. For the other stages, straight blades are used. Live steam after passing through the main stop valve is divided into two paths entering the turbine. Screen is installed inside the main stop valve in order to prevent water drops or other foreign matter from entering the turbine. After doing work in process of expansion, the steam flows into the condenser and is condensed into condenser which is then pumped by a condenser pump into the shaft sealing steam cooler, etc. then it will be boosted by the feed-water pump flows into the HRSG after oxygen being removed.

A part condenser through the condenser pump is to be sent back to the upper half of condenser to keep the normal operation of condenser and condenser pump during turbine starting with the sliding pressure and temperature. To eliminate any uneven thermal expansion, large steam pipes such as main steam pipes and the extraction steam pipes should be arranged symmetrically. It is necessary to use compensation bends for absorption expansion. [8]

Table 4-1: BOILER ALL HEATING SURFACES THERMODYNAMIC SPECIAL PROPERTY (DESIGN OPERATING CONDITIONS).

GT Model: PG6581B GT Fuel: Light Diesel Oil

No.	Item	Unit	HP superheat-er	HP evaporating Tube Bank	HP Economizer	LP boiler
1	Ambient temperature	˚C	40			
2	Inlet flue gas temperature	˚C	564	488.1	295	190.2
3	Outlet flue gas temperature	˚C	488.1	295	190.2	158.1
4	Inlet medium temperature	˚C	286.4	286.4	104	144.1
5	Outlet medium temperature	˚C	468	286.4	279.5	144.1
6	Average temperature difference	˚C	130.6	61.2	41.3	27
7	Average flue gas temperature	˚C	512.3	347.7	233	171
8	Average flue gas speed	m/s	17.1	14.7	11.9	10.3

Table 4-2: BOILER ALL HEATING SURFACES THERMODYNAMIC SPECIAL PROPERTY (DESIGN OPERATING CONDITIONS).

GT. Model: PG6581B G.T Fuel: LPG

No.	Item	Unit	HP superheat-er	HP evaporating Tube Bank	HP Economizer	LP boiler
1	Ambient temperature	˚C	40			
2	Inlet flue gas temperature	˚C	564	488.2	297.3	190.9
3	Outlet flue gas temperature	˚C	488.2	297.3	190.9	158.5
4	Inlet medium temperature	˚C	288.6	288.6	104	144.3
5	Outlet medium temperature	˚C	468	288.6	281.6	144.3
6	Average temperature difference	˚C	129.8	60.8	41.6	27.2
7	Average flue gas temperature	˚C	512.7	349.4	234.4	171.5
8	Average flue gas speed	m/s	17.2	14.8	12	10.4

Table 4-3: BOILER ALL HEATING SURFACES THERMODYNAMIC SPECIAL PROPERTY (66% LOAD OPERATING CONDITIONS)

G.T. Fuel: light diesel oil

Item	Unit	HP Superheater	HP Evaporating pipe bank	HP Economizer	LP boiler
Inlet flue gas temperature	˚C	564	488.1	295.0	190.2
Outlet flue gas temperature	˚C	488.1	295.0	190.2	158.1
Inlet medium temperature	˚C	267.22	267.22	104	113.32
Outlet medium temperature	˚C	265	267.22	263	113.32

Table 4-4: THE LIST OF SUMMARY OF DATA FOR CCPP (100% LOAD CONDITION)

NO.	Item	Unit	Unit 1& Unit2		Remarks
1	Gas Turbine				
1.1	Type				
1.2	Model		PG6001B		
1.3	Fuel		LDO	LPG	
1.4	Output at standard design conditior condition)	kW	39174	40053	
1.5	Output at rated design Condition	kW	31602	23366	Am.temp ˚C
1.6	Quantities	set	4		
1.7	Initial temperature	˚C	40	40	
1.8	Exhaust temperature	˚C	564	564	Am.temp ˚C
1.9	Exhaust flow rate	t/h	452	450	Am.temp ˚C
NO.	Item	Unit	Unit 1& Unit2		Remarks
2	Heat recovery steam Generator				
2.1	Type		Single pressure, horizonta firing		
2.2	Model		Q354/564.64-6.9/4		
2.3	Quantity	set	4		
2.4	100% load continuous rate	t/h	62.56	63.78	
2.5	Main steam temperature	˚C	468	468	Am.temp ˚C
2.6	Main steam pressure	MPa	7.0	7.0	Am.temp ˚C
2.7	Feed water temperature	˚C	104	104	To HRSC
2.8	Exhaust flow gas Temperature	˚C	153.9	153.9	Am.temp ˚C

NO.	Item	Unit	Unit 1& Unit2		Remarks
3	Steam Turbine				
3.1	Type		Directly condensing with extraction for heating LPG		
3.2	Model		1.35-6.7-1		
3.3	Quantity	Set	2		
3.4	Main steam flow-rate at ir main steam stop valve	˚C	2x62.56	2x63.78	Am.temp. ˚C
3.5	Pressure at inlet of main stop valve	MPa	6.7	6.7	Am.temp. ˚C
3.6	Temperature at inlet of main stop valve	˚C	465	465	Am.temp. ˚C
3.7	Extraction pressure	MPa		0.922	Am.temp. ˚C
3.8	Extraction temperature	˚C		242.9	Am.temp. ˚C
3.9	Extraction flow	t/h		6	Am.temp. ˚C
4	Back pressure	kPa	10.2	10.2	Am.temp. ˚C
4.1	Temperature of cooling	˚C	32	32	Am.temp. ˚C
4.2	Speed	r/min	3000	3000	Am.temp. ˚C
4.3	Output	kW	32720	32450	Am.temp. ˚C

Table 4-5: MAIN PERFORMANCE DATA (100% LOAD CONDITION)

NO	Item	Unit	Unit 1 & Unit 2	
1	Configuration mode		2 + 2 + 1	
2	Fuel		Diesel oil	LPG
3	Unit generating power	kW	95924	97182
3.1	Gas turbine output	kW	31602	32366
3.2	Steam turbine output	kW	32720	32450
3.3	Boiler 100% load continuous rate	t/h	62.56	63.78
3.4	Gas heat consumption	kJ/h	390×10^6	396×10^6
3.5	Fuel low heat value	kJ/kg	42697.2	45125
3.6	Gas turbine consumption heat rate	kJ/Kw.h	12340.99	12235.06
3.7	Gas turbine fuel consumption rate	g/Kw.h	289	271
3.8	Gas turbine efficiency	%	29.17	29.42
3.9	Gas turbine fuel Consumption	t/h	9.134	8.776
3.10	Plant heat consumption rate	kJ/Kw.h	8131.44	8149.66
3.11	Plant fuel consumption rate	g/kW.h	190.4	180.6
3.12	Plant efficiency	%	44.27	44.17

CHAPTER FIVE

EXERGY ANALYSIS FOR GARRI 1 POWER PLANT USING EXCEL

5.1. EXERGY ANALYSIS FOR STEAM CYCLE

5.1.1. HRSG:

The values of mass flow rate, temperature and pressure of steam/water at any point of are listed in tables (4-1) to (4-5). From steam table at a certain pressure and temperature the values of enthalpy and entropy are figured out, enthalpy H (kJ/s) is expressed as

$$H = M \times h$$

Process heat is expressed as

$$Q_P = H_O - H_i$$

Exergy of the steam/water per unit mass is defined from equation (3-22) as

$$\psi = (h - \overset{\nearrow 0}{h_0}) - T_0(s - \overset{\nearrow 0}{s_o}) + (\overset{\nearrow 0}{V^2/2}) + \overset{\nearrow 0}{gz}$$

Simplified to

$$\psi = h - T_0 s$$

Exergy of the steam/water is defined as

$$\Psi = m\,\psi$$

The exergy of steam/water produced from equation (3-23) is expressed as

$$\Delta\Psi = \Psi_2 - \Psi_1 = m\left[(h_2 - h_1) - T_0(s_2 - s_1)\right]$$

Or

$$\Delta\Psi = \Psi_i - \Psi_o = m\left[(h_i - h_o) - T_0(s_i - s_o)\right]$$

The exergy of steam/water produce per equipment is calculated as

$$\sum \Delta\Psi = \Delta\Psi_1 + \Delta\Psi_2 + \Delta\Psi_3 + \dots\dots\dots + \Delta\Psi_n$$

5.1.2. Steam turbine (ST):

From appendix (A) properties of steam/water mass flow rate, temperature, pressure and enthalpy for steam turbine are figured out in tables (5-1), (5-2) and (5-3), entropy enthalpy H, process heat Q_P , exergy of steam per unit mass, exergy of steam and exergy produced for steam turbine are calculated as in HRSG above.

5.1.3. Condenser:

From appendix (A) properties of steam/water mass flow rate, temperature, pressure and enthalpy for steam turbine are figured out in tables (5-1), (5-2) and (5-3), values of enthalpy for saturated steam at condenser inlet are calculated as

$$h = h_f + xh_{fg}$$

From the appendix (A) all the values of the dry ness fraction at inlet to condenser is limited to 0.88 so that in partial load calculations the average value is taken as approximation as

$$x = (x_1 + x_2 + x_3 + ...x_n)/n$$

Entropy, enthalpy H, process heat Q_P , exergy of steam per unit mass, exergy of steam and exergy produced for steam turbine are calculated as in HRSG above.

5.1.4. Feeding system (pumps):

The values of temperature and pressure of water at inlet and outlet of each pump is figured out from the actual data of pumps, mass flow rate, entropy, enthalpy H, process heat Q_P , exergy of steam per unit mass, exergy of steam and exergy produced for steam turbine are calculated as in HRSG above.

5.2. EXERGY ANALYSIS OF MAIN EQUIPMENT FOR GARRI POWER PLANT

5.2.1. Gas turbine one set:

From table (4-4) list of summary of data for CCPP 100 % load condition we figure out mass flow rate of fuel and exhaust. The mass flow rate of air can be calculated as

$$m_a = m_e - m_f$$

Also air/fuel ratio can be calculated to determined the amount of air for partial load as

$$A/F = m_a / m_f$$

Values of fuel enthalpy (L.H.V) are listed in table (4-5), enthalpy of air is calculated as below

$$h = h_a + wh_g$$

$$h_a = CpT$$

$$h_g = 2501.3 + 1.82T$$

$$w = 0.0622Pv/(P - Pv)$$

$$Pv = \Phi Pg \ \ at\ T$$

NOTE: values of Pg corresponding to T are listed in steam table.

Fuel energy is expressed as

$$E_f = m_f \times h_f \ \ (L.H.V)$$

The power out put of gas turbine listed in table (4-5).Gas turbine efficiency expressed as

$$\eta_{GT} = (W_{GT} / E_f) \times 100$$

The enthalpy of exhaust gases at inlet are given by the equation below

$$h = Cp_g \times T_{in}$$

The energy of the exhaust gases is expressed as

$$Energy = m_e \times H$$

First low efficiency can be calculated from the formula

$$\eta_I = \frac{(Q_P + W_{GT}) \times 100}{E_f}$$

5.2.2. HRSG one set:

The enthalpy of the exhaust gases at outlet is

$$h_o = Cp_g \times T_{out}$$

Process heat of exhaust gases Q_P expressed as

$$Q_P = H_i - H_O$$

values of steam/water enthalpy for HP&LP are listed in table (5-1), (5-2) and (5-3) so that at inlet to LP is that at inlet to evaporator and so at out let, for HP at inlet is that entering economizer and at outlet is that leaving superheater, m_S is considered as the mass flow rate leaving superheater, there is loss of steam at HP drum due to continuous blow down 1% of total mass flow rate.

The efficiency of the HRSG is expressed as

$$\eta_{HRSG} = \frac{(Q_P HP + Q_P LP) \times 100}{Q_P}$$

5.2.3. Steam turbine set (combined by 2 HRSGs):

Enthalpy of HP steam Hi expressed as

$$H_i HP = 2 \times m_s HP \times h_o$$

Also LP steam

$$H_i LP = 2 \times m_s LP \times h_o$$

$$\sum H_i = H_i HP + H_i LP$$

The values of steam power out put are listed in table (4-5) Ranking actual efficiency is calculated from the formula

$$\eta_I = \frac{W_{ST} \times 100}{\sum H_i}$$

Table 5-1: EXERGY ANALYSIS FOR STEAM CYCLE (GAS TURBINE FUEL LIGHT DESIL OIL FOR 100% LOAD).

EQUIPMENT	position	M(kg/s)	T(C°)	P(MPa)	h(kj/kg)	H(Kj/s)	Qp(kj/s)	S(kj/kg.K)	Ψ(kj/kg)	ψ(kj/s)	∑Δψ(kj/s)
HRSG: SUPER HEATER	i	34.167	286.4	7.064	2772	94710.924	19136.94	5.8108	953.2196	32568.654	-42824.3
	o	34.167	468	7	3332.1	113847.8607		6.6892	1238.38	42311.743	
EVAPURATOR	i	34.512	286.4	7.064	1270.3	43840.5936	50870.33	3.1268	291.6116	10064.1	
	o	34.167	286.4	7.064	2772	94710.924		5.8108	953.2196	32568.654	
ECONOMIZER	i	34.512	104	8.2	441.67	15242.91504	27324.19	1.3349	23.8463	822.98351	
	o	34.512	279.5	7.064	1233.4	42567.1008		3.0605	275.4635	9506.7963	
L.P EVAPURATOR	i	3.561	142.9	0.39	600.58	2138.66538	7600.67	1.776	44.692	159.14821	
	o	3.561	142.9	0.39	2735	9739.335		6.897	576.239	2051.9871	
ST: GOVERNER VALVE	i	34.167	465	6.7	3329	113741.943	36804.42	6.7041	1230.617	42046.481	46846.54
	o	33.22	48.6	0.0115	2316	76937.52	0	7.861	-144.493	-4800.0575	
CONDENSER: CONDENSATE STEAM	i	33.22	48.6	0.0115	2316	76937.52	70183.89	7.861	-144.493	-4800.0575	-4428.03
MAKE UP WATER	i	0.947	48.6	0.0115	203.3	192.5251	0	0.6853	-11.1989	-10.605358	
HOT WELL	o	34.167	48.6	0.0115	203.3	6946.1511	0	0.6853	-11.1989	-382.63282	
FEEDING SYSTEM: C.P	i	34.167	48.6	0.0115	203.3	6946.1511	375.837	0.6853	-11.1989	-382.63282	-46.4535
	o	34.167	51	1.1	214.3	7321.9881		0.7161	-9.8393	-336.17936	
H.P.F.W.P	i	34.167	104	0.12	435.76	14888.61192	201.927	1.3515	12.7405	435.30466	-272.509
	o	34.167	104	8.2	441.67	15090.53889		1.3449	20.7163	707.81382	
L.P.F.W.P	i	3.561	104	0.12	435.76	1551.74136	2.52831	1.3515	12.7405	45.368921	-3.41998
	o	3.561	104	1.1	436.47	1554.26967		1.3507	13.7009	48.788905	
	313										

Table 5-2: EXERGY ANALYSIS FOR STEAM CYCLE (GAS TURBINE FUEL LPG FOR 100% LOAD).

EQUIPMENT	position	M(kg/s)	T(C°)	P(MPa)	h(kj/kg)	H(Kj/s)	Qp(kj/s)	S(kj/kg.K)	Ψ(kj/kg)	ψ(kj/s)	∑Δψ(kj/s)
HRSG: SUPER HEATER	i	35.433	288.6	7.2928	2769	98113.98	19952.32	5.7945	955.3215	33849.91	-44400
	o	35.433	468	7	3332.1	118066.3		6.6892	1238.38	43879.53	
EVAPURATOR	i	35.791	288.6	7.2928	1282	45884.06	52229.92	3.1472	296.9264	10627.29	
	o	35.433	288.6	7.2928	2769	98113.98		5.7945	955.3215	33849.91	
ECONOMIZER	i	35.791	104	8.2	441.67	15807.81	28726.93	1.3349	23.8463	853.4829	
	o	35.791	281.6	7.2928	1244.3	44534.74		3.0797	280.3539	10034.15	
L.P EVAPURATOR	i	3.694	144.3	0.4075	607.37	2243.625	7868.331	1.7828	49.3536	182.3122	
	o	3.694	144.3	0.4075	2737.4	10111.96		6.8867	581.8629	2149.402	
ST: GOVERNER VALVE	i	35.433	465	6.7	3329	117956.5	38869.24	6.7041	1230.617	43604.44	52424.2
	o	34.444	45.6	0.0099	2296.11	79087.21	0	8.1539	-256.061	-8819.75	
CONDENSER: CONDENSATE STEAM	i	34.444	45.6	0.0099	2296.11	79087.21	72466.27	8.1539	-256.061	-8819.75	-8428.39
MAKE UP WATER	i	1.667	20	0.0099	83.79	139.6779	0	0.2959	-8.8267	-14.7141	
HOT WELL	o	35.433	45.6	0.0099	190.8	6760.616	0	0.6462	-11.4606	-406.083	
FEEDING SYSTEM: C.P	i	35.433	45.6	0.0099	190.8	6760.616	389.763	0.6462	-11.4606	-406.083	-45.9566
	o	35.433	48	1.1	201.8	7150.379		0.6772	-10.1636	-360.127	
H.P.F.W.P	i	35.433	104	0.12	435.76	15440.28	209.409	1.3515	12.7405	451.4341	-282.607
	o	35.433	104	8.2	441.67	15649.69		1.3449	20.7163	734.0407	
L.P.F.W.P	i	3.694	104	0.12	435.76	1609.697	2.62274	1.3515	12.7405	47.06341	-3.54772
	o	3.694	104	1.1	436.47	1612.32		1.3507	13.7009	50.61112	
	313										

Table 5-3: EXERGY ANALYSIS FOR STEAM CYCLE (GAS TURBINE FUEL LIGHT DESIL OIL FOR 66% LOAD).

EQUIPMENT	position	M(kg/s)	T(C°)	P(MPa)	h(kj/kg)	H(Kj/s)	Qp(kj/s)	S(kj/kg.K)	Ψ(kj/kg)	ψ(kj/s)	∑Δψ(kj/s)
HRSG: SUPER HEATER	i	27.222	267.22	5.27	2791	75976.6	15198.04	5.9486	929.0882	25291.64	-33354
	o	27.222	465	5.2	3349.3	91174.64		6.845	1206.815	32851.92	
EVAPURATOR	i	27.497	267.22	5.27	1170.8	32193.49	43783.11	2.9505	247.2935	6799.829	
	o	27.222	267.22	5.27	2791	75976.6		5.9486	929.0882	25291.64	
ECONOMIZER	i	27.497	104	8.2	441.67	12144.6	19460.45	1.3349	23.8463	655.7017	
	o	27.497	263	5.27	1149.4	31605.05		2.9109	238.2883	6552.213	
L.P EVAPURATOR	i	3.333	113.32	0.16	475.2	1583.842	7399.26	1.4547	19.8789	66.25637	
	o	3.333	113.32	0.16	2695.2	8983.102		7.2002	441.5374	1471.644	
ST: GOVERNER VALVE	i	27.222	462	5.1	3343.6	91019.48	31148.99	6.8459	1200.833	32689.08	33091.91
	o	26.222	39.4	0.0025	2283.216	59870.49	0	7.3437	-15.3621	-402.825	
CONDENSER: CONDENSATE STEAM	i	26.222	39.4	0.0025	2283.216	59870.49	55465.38	7.3437	-15.3621	-402.825	-94.1473
MAKE UP WATER	i	1	20	0.0025	83.796	83.796	0	0.2959	-8.8207	-8.8207	
HOT WELL	o	27.222	39.4	0.0025	164.9	4488.908	0	0.5641	-11.6633	-317.498	
FEEDING SYSTEM: C.P	i	27.222	39.4	0.0025	164.9	4488.908	299.7142	0.5641	-11.6633	-317.498	-30.4669
	o	27.222	41.8	1.1	175.91	4788.622		0.5957	-10.5441	-287.031	
H.P.F.W.P	i	27.222	104	0.12	435.76	11862.26	160.882	1.3515	12.7405	346.8219	-217.117
	o	27.222	104	8.2	441.67	12023.14		1.3449	20.7163	563.9391	
L.P.F.W.P	i	3.333	104	0.12	435.76	1452.388	2.36643	1.3515	12.7405	42.46409	-3.20101
	o	3.333	104	1.1	436.47	1454.755		1.3507	13.7009	45.6651	
	313										

Table 5-4: EXERGY ANALYSIS OF MAIN EQUIPMENT FOR GARRI POWER PLANT.

GAS TURBINE (ONE SET)

load (%)	m(kg/s) fuel	m(kg/s) air	h(kj/kg) fuel	h(kj/kg) air	Ef (kj/s)	WGT (KW)	η (%)	Exhaust gas me(kg/s)	Exhaust gas H(kj/kg)	Exhaust gas energy(kj/s)	Qp (kj/s)	η1ST (%)
100	2.5372	123.0188	42697.2	88.098	108331.3	31602	29.17161	125.556	648.6	81435.62	53180.44	78.26216
100	2.4378	122.5622	45125	88.098	110005.7	32366	29.42211	125	648.6	81075	55143.91	79.55033
66	1.8717	94.101	42697.2	88.098	79916.35	20000	25.02617	95.9727	648.6	62247.89	43275.38	79.17702

HRSG (ONE SET)

LOAD (%)	Exhaust gas me(kg/s)	Exhaust gas hi(kj/kg)	Exhaust gas ho(kj/kg)	Exhaust gas Qp(kj/s)	HP steam ms(kg/s)	HP steam hi(kj/kg)	HP steam ho(kj/kg)	HP steam Qp(kj/s)	LP steam ms(kg/s)	LP steam hi(kj/kg)	LP steam ho(kj/kg)	LP steam Qp(kj/s)	ηHRSG (%)
100	125.556	648.6	181.815	58607.66	17.084	441.67	3332.1	49380.11	1.7805	600.58	2735	3800.335	90.73975
100	125	648.6	182.275	58290.63	17.717	441.67	3332.1	51209.75	1.847	607.37	2737.4	3934.165	94.60169
66	95.9727	648.6	181.815	44798.62	13.611	441.67	3349.3	39575.75	1.6665	475.2	2695.2	3699.63	96.59982

STEAM TURBINE SET(COMBINED BY 2 HRSGs)

LOAD (%)	HP steam Hi (kj/s)	LP steam Hi (kj/s)	∑Hi (kj/s)	W ST (KW)	η Rankine (%)
100	113851.2	9739.335	123590.5	32720	26.47452
100	118069.6	10111.96	128181.6	32450	25.31565
66	91174.64	8983.102	100157.7	25100	25.06047

CHAPTER SIX

RESULTS AND DISCUSSION

6.1. EXERGY ANALYSIS OF CCPP

A CCPP as shown in Fig. (4-1) includes both the Brayton cycle and Rankine cycle. It joins operation of the gas turbine at the ''hot end'' and the steam turbine at the ''cold end.'' The scale diagram of temperature vs. entropy for CCPP is plotted in Fig. (6-1). Referring to Figs.(4-1) and (6-1), gas turbine operates on the Brayton Cycle, i.e., intake air is compressed nearly isentropically from point 1 to 2, combusted at constant pressure from point 2 to 3, and then expanded nearly isentropically in the gas turbine from point 3 to 4, exhausting gas from point 4 to 5. The Brayton cycle as applied to a gas turbine is an open cycle, temperature at point 5 is higher than 100˚C and does not form a closed loop with the inlet air of point 1. For the Rankine cycle, points 9 to 6 shown in Fig. (6-1) are a heating process at constant pressure (pressure drop is shown in Table (5-1)). The actual turbine expansion from point 6 to 7 is an irreversible (non-isentropic) process. No pressure losses are encountered in the condenser process from point 7 to 8 because it is a two-phase condensation process. The processes of pumps and heaters from point 8 to 9 are nearly isentropic. The working fluid of the Rankine cycle is a vapor–liquid and the Brayton cycle is a gas. The two different working fluids are shown in Fig. (6-1), which is temperature and entropy (T–S) diagram showing the operating data at each process.

The useful products of a CCPP are electrical energy, W_E, and thermal energy, Q_P, in the form of superheated steam. The thermodynamic performance is based on the first law efficiency, and is defined as

$$\eta_I = \frac{(W_E + Q_P)\times 100}{E_f} \tag{6-1}$$

For simple cycle, W_E is equal to W_{GT} and for combined cycle, W_E is equal to W_{GT} + W_{ST}. Actually, the efficiency of a CCPP is reduced by the various inherent losses. One unavoidable loss is heat lost by radiation and/or convection, while a second is the internal loss caused by irreversible processes as discussed in the second law of thermodynamics.

Q_P is defined as the energy of steam generated in the HRSG or equipment and is calculated from high-pressure steam $ms_H(h-h_C)_H$ plus low-pressure steam $ms_L(h-h_C)_L$. The HRSG of GARRI 1 is a pure energy converter transferring heat from the GT exhaust gas without supplemental fuel combustion, therefore, no combustion takes place in the HRSG. Exergy analyses of GARRI 1 operated steam cycle for 100% load with LPG, 100% load with LDO and 66% load with LDO are shown on Tables (5-1) , (5-2) and (5-3) respectively. GARRI 1(one block) consists of two GTs and one ST forming a combined cycle unit. When GARRI 1 operates at 66% load, the two GTs and associated HRSGs operate at 66% load and the STG operates in the sliding pressure mode at 66% ST load. GT control is achieved by variable inlet guide vanes, adjusted to keep the exhaust gas temperature constant at variable GT loads. The exergy evaluation in the system has to consider the mass flow rate of the steam/ water. Theoretically in a steam/water cycle, the exergy of the steam/water produced in the HRSG plus power consumption of the pumps should be equal to the steam turbine gross power output plus the steam extracted by the heaters and exhausted to the condenser. However, from the second law of thermodynamics, the $\sum j\Delta Sj \geq 0$ and compared with Eq. (3-20) then $\sum j\Delta\Psi j \leq 0$. Tables (5-1), (5-2) and (5-3) show the amount of $\sum \Delta\boldsymbol{\psi}$ lost in the whole steam/water cycle amounts to 1.55% ,1.73% and 1.83, respectively, which supports the second law of thermodynamics.

The power to heat ratio for simple cycle (GT only),

$$R = \frac{W_{GT}}{Q_P} \tag{6-2}$$

and for combined cycle,

$$R = \frac{W_E}{Q'_P} \tag{6-3}$$

Where in $Q'_P = Q_P - W_{ST}$. From Table (5-4), only 26.47%, 25.31% and 25.06%, of steam energy can be converted to electrical power, which is the efficiency of Rankine cycle portion of GARRI 1 at 100% load LDO, 100% load LPG and 66% load LDO, respectively and at design conditions. If the electrical power and steam thermal energy are treated as equivalent energy levels, it is called the law of the energy conservation, part of the first law of thermodynamics. However, electrical power is much more valuable than steam/water thermal energy in a power generating plant. The energy can proceed in a certain direction but not in the reverse direction. Exergy, the essential concept in second law analysis, is always consumed or destroyed in any process. If less exergy is consumed, a cycle can produce more efficiently. Therefore, by using exergy to evaluate the power plant cycles, a more accurate performance of the system can be obtained

$$\eta_{II} = \frac{(W_E + \Delta\psi) \times 100}{\psi_f} \tag{6-4}$$

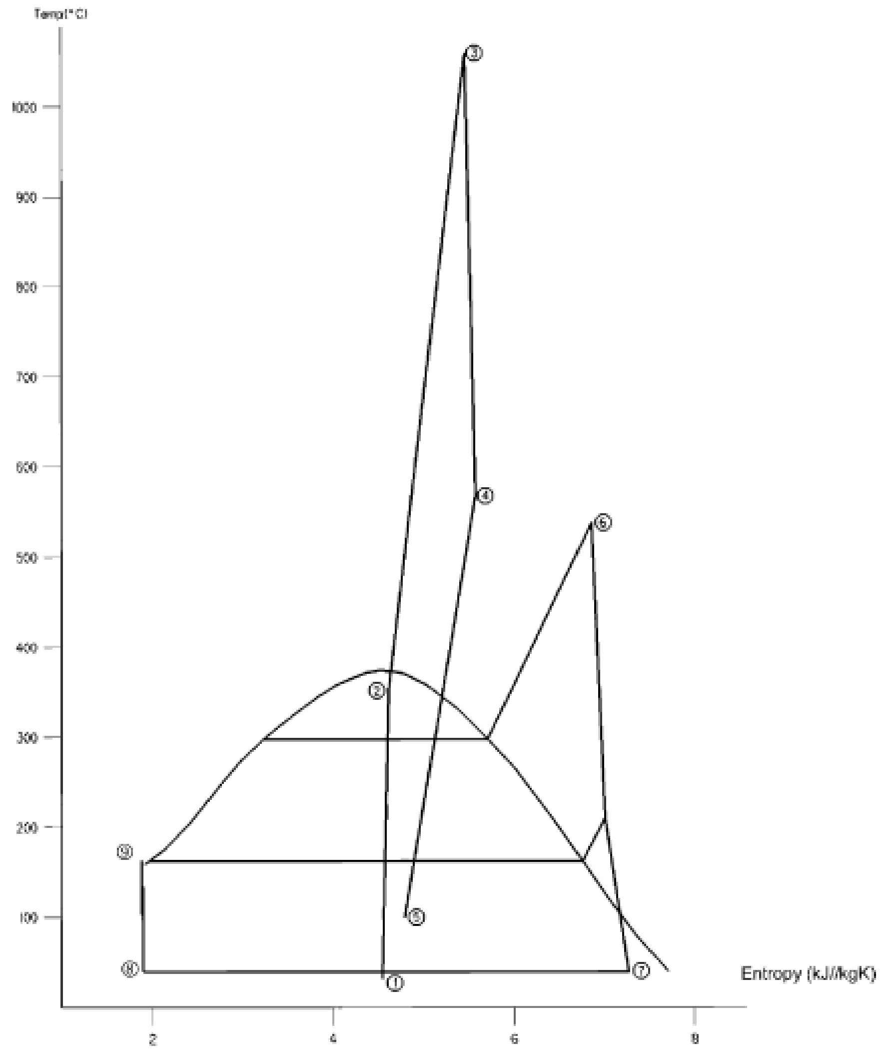

Fig. (6-1). T–S diagram of combined cycle plant.

The exergy factor of generated steam/water (ε_S) and the exergy factor of fuel input (ψ_f) can be expressed as follows:

$$\varepsilon_S = \frac{\Delta\psi}{Q_P} \tag{6-5}$$

$$\varepsilon_f = \frac{\psi_f}{E_f} = 1 \tag{6-6}$$

Substituting in equation (6-4)

$$\eta_{II} = \frac{W_E + \varepsilon_S Q_P + Q_P - Q_P}{\varepsilon_f E_f} = \frac{W_E + Q_P + (\varepsilon_S - 1)Q_P}{\varepsilon_f E_f}$$

$$= \frac{\eta_I}{\varepsilon_f}\left[1 + \frac{\varepsilon_f}{\eta_I} \times \frac{(\varepsilon_S - 1)Q_P}{\varepsilon_f E_f}\right] = \frac{\eta_I}{\varepsilon_f}\left[\frac{W_E + \Delta\psi}{\eta_I E_f}\right]$$

$$= \frac{\eta_I}{\varepsilon_f}\left[\frac{R + \varepsilon_S}{R + 1}\right] \tag{6-7}$$

6.1.1. Brayton cycle theoretical efficiency

The ideal Brayton cycle is composed of two adiabatic–reversible (isentropic) and two constant pressure processes (as shown in Fig. (6-1)). The theoretical GARRI 1 GT efficiency (η_{GT}) at ISO conditions with a flue gas inlet temperature of 1000˚C and exhaust temperature of 564˚C is 34.24%. This theoretical efficiency is 5.07 %, 4.82% and 9.21% higher than the actual gas turbine operating efficiency of 29.17%, 29.42% and 25.03% respectively, due to the mechanical losses and non-isentropic processes existing in the operation.

6.1.2. Rankine cycle theoretical efficiency

The conditions of the steam generated from the HRSG will affect the steam cycle efficiency. The Rankine cycle of the steam turbine for GARRI 1 utilizes steam as a working fluid and the condenser as a heat rejection reservoir, operating at constant pressure. Assume the turbine expansion and the feed water pumping processes are isentropic. The Rankine cycle efficiency is defined as the turbine work (enthalpy difference between the steam at the turbine inlet and at the turbine exhaust outlet) divided by the inlet steam enthalpy. The ideal Rankine cycle efficiencies for GARRI 1 with superheated pressure steam conditions is 32.36, 32.95 and 34.22% at a condensers pressure of 0.0115, 0.0099 and 0.0025MPa respectively.

6.2. EFFECT OF PARTIAL LOAD ON CCPP EFFICIENCY

Using the first law to evaluate the efficiency of GARRI 1 at 100% load (see Table (5-1)), the total steam energy at the two HRSG's HP and LP superheater outlets is 123587.196 kJ/s; the net energy output from steam turbine is 36804.42 kJ/s. The calculated first law efficiency (η) of steam cycle is 29.78 %. Using the second law, the total outlet exergy of HP and LP steam from the HRSG is 44363.73 kJ/s, the inlet exergy to steam turbine is 42046.48 kJ/s. The destroyed exergy from HRSG and pumps to steam turbine and condenser is 2317.25 kJ/s (5.46%) due to the piping insulation thermal loss. The destroyed exergy at the condenser is 4428.03kJ/s (8.64%). The total destroyed exergy in the GARRI 1 combined cycle is shown in table (6- 1). The exergy analyses of GARRI 1 operating at 66% load LDO is shown in Table (5-2) and the component exergy analyses is shown in Table (5-4).

From Table (5-4), The GARRI 1 actual efficiency (η_{cc}) is 44.27% [95924 kW/(216662.6 kJ/s)] for 100% load with LDO power output from the operation record. For the 66% LDO power output of the CCPP, the GARRI 1 actual operation efficiency is 40.73%. The actual operation efficiency 100% load LPG is 44.17 %.

The exergy loss at 66% load is 1.2 times that of 100% load due to the lower steam pressure in the HRSG. It can be verified from the steam T–S diagram, as the LP steam has higher entropy value than HP steam at the same temperature. Therefore, the GARRI 1 operating efficiency (η_{II}) at 100% load is 2.44% higher than at 66% load.

TABLE (6-1): EXERGY DESTRUCTION FOR GARRI 1 COMPONENTS.

Equipment	unit	EXERGY DESTRUCTION		
		100% LDO	100% LPG	66%LDO
HRSG	kJ/s	22601.42	20736	16656.22
S.T	kJ/s	10042.14	13554.96	1942.92
CONDENSE	kJ/s	4428.03	8428.39	94.147
C.P	kJ/s	329.384	343.806	269.247
H.P.F.W.P	kJ/s	70.582	73.198	56.235
L.P.F.W.P	kJ/s	0.89167	0.92498	0.83458
GT	kJ/s	67578.86	29779.9	54187.76

6.3. THERMODYNAMICS ANALYSIS

The first law efficiency of simple cycle η_I = 78.26% is calculated by Eq. (6-1), where the W_{GT}, Q_P and E_f are shown in Table (5-4), item 1. The second law efficiency of simple cycle η_{II} = 48.93%, where in the heat to power ratio (*R*) is equal to 0.5942 which is calculated by Eq. (6-2) and ε_S = 0.4026 which is calculated by Eq.(6-3). $\Delta\psi$ is listed in Table (5-1).

The first law efficiency of combined cycle is same as simple cycle; however, the steam turbine power output needs to be deducted from *Qp* to obtain Q'_P, which is equal to 73640.88 kJ/s. The second law efficiency of combined cycle η_{II} = 64.04%, wherein R is equal to W_E/Q'_P = 1.3026 and $\varepsilon_S = \Delta\psi/Q'_P$ = 0.5815.

The exhaust gas leaves the gas turbine through a horizontal outlet. Then the gas enters a 90˚ upward bend at the base of the vertical tower section, which contains the HRSG heating surfaces. The HRSG heating surfaces arranged in the direction of gas flow are HP superheater, HP evaporator, HP economizer and LP evaporator forming a dual pressure boiler. The three main parts producing HP steam are firstly, the HP economizer, secondly, the HP evaporator, and the lastly, HP superheater, all working at a HP steam pressure. The condensate at 51 ˚C enters to the feed water tank. It is heated to saturation and deaerated in the feed water tank. Then, the low-pressure feed pump takes some deaerated water from the feed tank and discharges this deaerated feed to the LP evaporator at 104 ˚C and the feed water is evaporated. The vapor after being separated from the unevaporated water in the LP steam drum passes again to the deaerator. The HP feed water pump takes the balance of the deaerated water from the feed tank and discharges it into the HP economizer. The feed water temperature at HP economizer inlet is (104 ˚C), a sub-cooled liquid state. The sub-cooled liquid is heated in the economizer and exits at (279.5 ˚C) to the HP evaporator, where it is changed to the saturated vapor state. The vapor is separated from the liquid in the HP drum and the saturated vapor enters the HP superheater exiting at a superheated temperature of 468 ˚C.

The combustion gas, in counter flow, enters the HP superheater at the gas turbine exit condition of (564 ˚C), goes through the HP superheater and then the HP evaporator where it exits at a temperature of (295 ˚C). The flue gas enters the LP evaporator and exit at Te (158.1˚C). The gas and steam temperatures profile of the HRSG are shown in Fig. 6-3.

In a dual pressure system, there is a substantial increase in the amount of heat recovered in the HRSG. Usually, the HP steam can be optimized at a higher pressure than the single pressure cycle; more of the energy is transferred into exergy. In addition, LP steam is produced, recovering the heat at the low temperature end of the HRSG and lowering the stack gas temperature by the addition of LP evaporator. The efficiency analysis is listed in Fig. (6-2).

Fig. (6-2) is based on the temperature of the flue gas, which at pinch point of HP portion is 295˚C. The pinch point is fixed at 8.6˚C of HP portion for Plant . The actual second law efficiency of the simple cycle is lower than the combined cycle because the power-to-heat ratio (R) is only 0.5942 instead of 1.3026. The higher exergy ratio of generated steam/water and the power-to-heat ratio both increase the thermal efficiency. Fig. (6-2) depicts the efficiency evaluation based on first law without considering the exergy of generated steam/water.

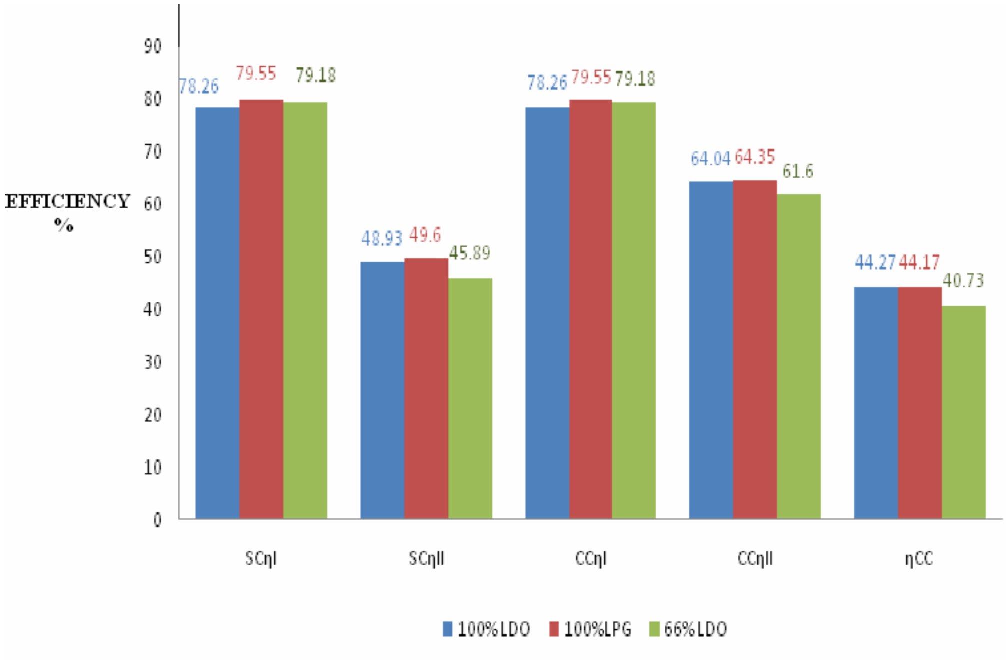

Figure (6-2).Thermodynamic analysis of Garri 1

6.4. EVALUATION OF TEMPERATURE AND HEAT ABSORPTION FOR HRSG

The temperature vs. heat absorption diagrams (Fig. 6-3,4,5) shows the transferred heat flow Q (horizontal axis), as the flue gas passes through the HRSG in percent of heat absorbed and the vertical scale is temperature. The key parameters Q (heat flow) and Cp (specific heat) are nearly constant over the entire passage through the HRSG resulting in a straight line representation of the flue gas temperature. The temperature drop over each heating surface is therefore proportional to the heat absorbed by the steam/water system. The highest quantity of heat being transferred is in the HP evaporator and the smallest is in the LP evaporator. The quantity of heat transferred in the LP evaporator appears with a very short and narrow space.

In designing the HRSG, the incoming condensate entering the feed water tank. The feed water tank is designed to operate as a deaerator. To achieve proper deaeration in the feed water tank, a temperature difference is required between the condensate entering the feed water tank and the feed water tank temperature itself. Both the LP and HP feed water pumps take suction from the feed water tank and supply feed water at essentially feed water tank temperature to the LP drum/evaporator and to the HP economizer and drum/evaporator, respectively.

The inlet temperature of the feed water entering the HP economizer is nearly the same as the feed water entering the LP drum, as both are fed from the common feed water tank via separate feed water pumps. Due to the different pressure levels, the HP feed water will be heated slightly more by the feed water pump than the LP feed water.

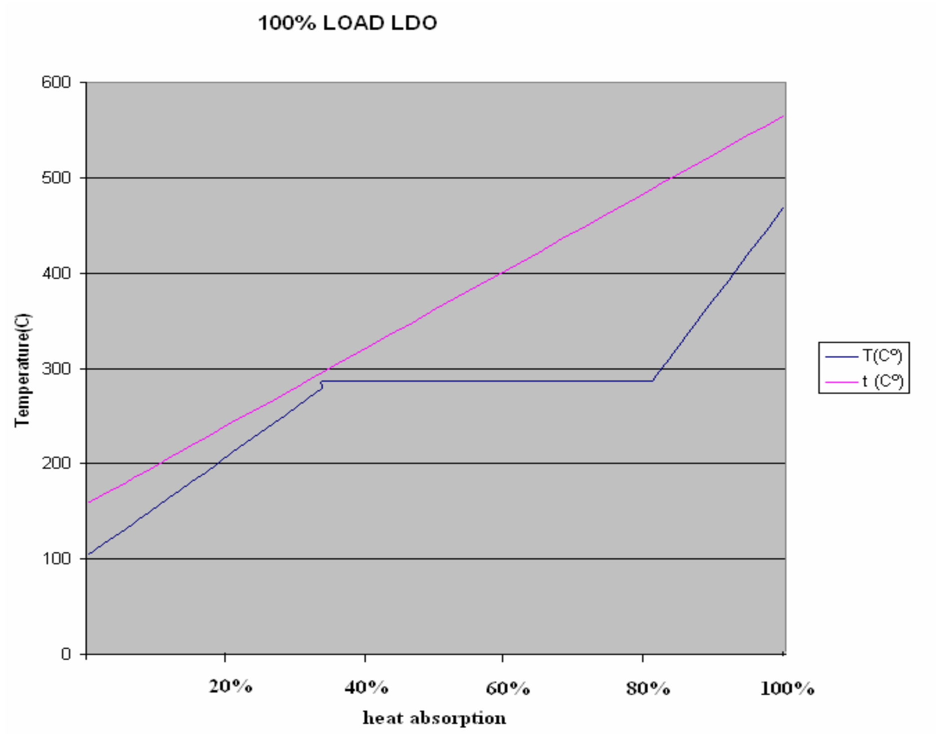

Figure 6-3: Gas and steam temperature profile of HRSG for 100 % load LDO

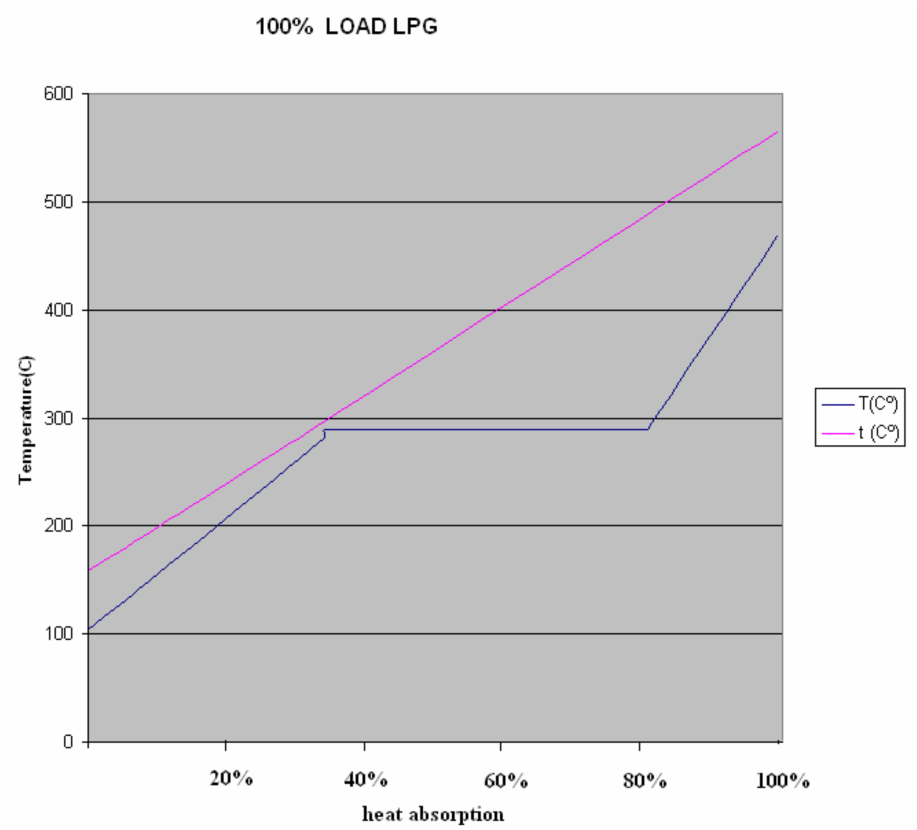

Figure 6-4: Gas and steam temperature profile of HRSG for 100% load LPG

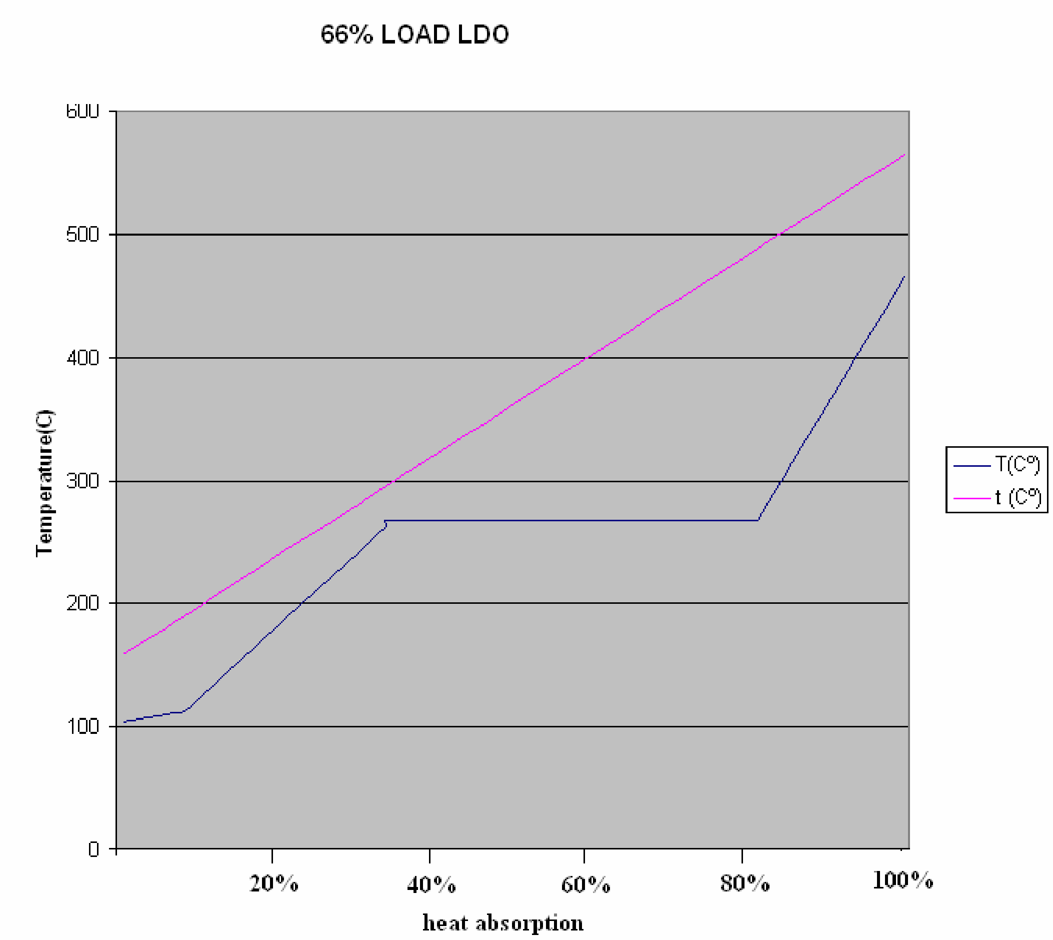

Figure 6-5: Gas and steam temperature profile of HRSG for 66% load LDO

6.5. EFFECT OF PINCH POINT ON PLANT PERFORMANCE

The pinch point is the minimum difference between the gas temperature leaving the evaporator section of the HRSG and the saturation temperature corresponding to the steam pressure in the evaporator section. Generally, lowering the pinch point results in an increase of total heat recovered in the evaporator section. However, lowering the pinch point will decrease the logarithmic mean temperature difference (LMTD) in the HRSG, requiring more heat transfer surface area. This significantly increases the equipment cost.

The pinch point for the HP portion is $\Delta TH = 8.6^{\circ}C$ (295-286.4 ˚C) .This pinch point is important parameter affecting the thermal performance of the system.

If the pinch point of an HRSG is changed to 10, 20 or 30 ˚C and the pressure of process steam/water is fixed, the total required heating surface of HRSG will be reduced, leading to a reduction in steam temperature. This change will affect the steam cycle thermal performance.

CHAPTER SEVEN

CONCLUSION AND RECOMMENDATION

7.1. CONCLUSION

Based on first law (energy) and second law (exergy) analysis, the formulas for dual pressure HRSG have been developed for thermodynamic performance and engineering betterment of combustion gas turbine based power generation system. The plant performance can be improved by improving the system design such as using dual pressure steam cycles, inlet air cooling for the gas turbine, and fuel gas preheating, etc. From fig. (6-2), the efficiency difference of CCPP between second law and actual plant operation at design condition is 19.77, 20.18 and 20.87 %, for 100% load LDO, 100% load LPG and 66% load LDO respectively, which is caused by a steam cycle efficiency of only 26.47%, 25.32% and 25.06% as shown in Table (5-4).

The higher exergy destruction occurs because the lower pressure steam has high entropy value at same temperature level. Therefore, the CCPP operated at 66% load has an efficiency of 3.54% lower than the 100% load. From figure (6-2) the second law efficiency difference between 100% load LPG and 100% load LDO is only 0.31%. From table (6-1), Maximum exergy loss was found at the gas turbine 67578.86 kJ/s which is (31.2 % of input). The second important loss was at HRSG 22601.42 kJ/s which is (10.43 % of input). Exergy destruction at pumps is too small compared to that at other components.

7.2. RECOMMENDATION:

1. Performance test for 100% load, gas turbine using light diesel oil and liquefied petroleum gas can be considered in exergy analysis.
2. The cooling tower exergy destruction by mass as evaporated water loss (exergy transfer by mass), exergy transfer by heat and so on should be considered.
3. With well insulation of steam/water pipe exergy destruction within pipes and valves is comparatively small so can neglected.
4. The efficiency deviation between exergy analyses and performance test results can be improved by engineering design.
5. GARRI 1 power plant should be operated at full load using LPG as gas turbine fuel or full load using LDO and never be operated at partial load.
6. One of the important improvements on the combined cycle is the compressor inlet air cooling, now a days evaporative media type is carried out, so effect of this system on first and second low efficiency can be calculate.
7. According to National Electricity Corporation long term plan there are 6191MW driven from combined cycle power plants so when designing these combined cycles we have to consider the exergy and carried out through the elementary design meetings.

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List of contents