Exergy Analysis of Garri (1) Power Plant

A thesis
Submitted in partial Fulfillment of the Degree of Master of Science in Mechanical Engineering (Energy)

Prepared by
Mudathir Mohamed Idress Shamaoon

Supervised by
Dr. Mohamed Musadag El-Awad

August 2008
قال الله تعالى:

(الله نور السماوات والأرض مثل نور كمِشِكَة فيها مِصِباَح المِصِباَح في رَجَََّاجة الزَََّجاجة كَأنها كوكب دَرَي يِوْقَد من شَجَرة مِبَارَكة زِيَثُوْنَة لَا شَرِِّقِيَة وَلا غَرَِّبيَة يَكَاد زِيْتُها يُضِيء وَلَو لَم تَمْشِه نَار نُور على نُور يَهْدِي الله لِنوره مِن يَشَاء وَيَضِرب الله الأمِثال لِلنَّاس والله يَكُل شَيْٰء عَلِيمَ)

سورة النور
الأيَّة 35
Acknowledgements

I wish to thank my adviser, Dr. Mohammed Mosadag, for intellectual support, encouragement, and enthusiasm which made this thesis possible, and for his patience in connecting both my stylistic and scientific errors.

I thank my parents, brothers and colleges, for all their intellectual, emotional and spiritual support.

This research was supported by my colleges from the National Electricity Corporation NEC in Garri 1 power station, project and planning department and load dissipate center.
Abstract

The objective of this research is to present 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.
بحث

يتطلب البحث تحديد الأداء الشامل للأنظمة الحرارية، بما يمكن الملاحظة فيه من التغييرات في كل من التحليل، المنتجة للقدرة، كفاءة الحرارية، الدورات، التي يمكن أن تكون موجودة في المنظمة، يمكن أن تكون في اللف vandal يتم استخدام Excel لحساب النتائج التي يعرضها في اللف vandal يمكن أن تكون هذه الدورات كفاءة حسابها ببرنامج بواسطة Excel. Excel يعطي نتائج جيدة في هذا اللف vandal، يمكن أن تكون هذه الدورات كفاءة حسابها ببرنامج بواسطة Excel.

تعتبر الدورات العامة للف vandal

لا يمكن

الطة، التحليل

لدائم

الثاني

القانون

الأول

القانون

أساس

علي

علي

الحصول

القدرة

توليد

الأنظمة الحرارية.

يتكامل

الات

التي يتغير من كل تحليل، المنتجة للقدرة، كفاءة الحرارية، الدورات، التي يمكن أن تكون موجودة في المنظمة، يمكن أن تكون في اللف vandal يتم استخدام Excel لحساب النتائج التي يعرضها في اللف vandal يمكن أن تكون هذه الدورات كفاءة حسابها ببرنامج بواسطة Excel. Excel يعطي نتائج جيدة في هذا اللف vandal، يمكن أن تكون هذه الدورات كفاءة حسابها ببرنامج بواسطة Excel.

تعتبر الدورات العامة للف vandal

لا يمكن

الطة، التحليل

لدائم

الثاني

القانون

الأول

القانون

أساس

علي

علي

الحصول

القدرة

توليد

الأنظمة الحرارية.

يتكامل

الات

التي يتغير من كل تحليل، المنتجة للقدرة، كفاءة الحرارية، الدورات، التي يمكن أن تكون موجودة في المنظمة، يمكن أن تكون في اللف vandal يتم استخدام Excel لحساب النتائج التي يعرضها في اللف vandal يمكن أن تكون هذه الدورات كفاءة حسابها ببرنامج بواسطة Excel. Excel يعطي نتائج جيدة في هذا اللف vandal، يمكن أن تكون هذه الدورات كفاءة حسابها ببرنامج بواسطة Excel.

تعتبر الدورات العامة للف vandal

لا يمكن

الطة، التحليل

لدائم

الثاني

القانون

الأول

القانون

أساس

علي

علي

الحصول

القدرة

توليد

الأنظمة الحرارية.

يتكامل

الات

التي يتغير من كل تحليل، المنتجة للقدرة، كفاءة الحرارية، الدورات، التي يمكن أن تكون موجودة في المنظمة، يمكن أن تكون في اللف vandal يتم استخدام Excel لحساب النتائج التي يعرضها في اللف vandal يمكن أن تكون هذه الدورات كفاءة حسابها ببرنامج بواسطة Excel. Excel يعطي نتائج جيدة في هذا اللف vandal، يمكن أن تكون هذه الدورات كفاءة حسابها ببرنامج بواسطة Excel.

تعتبر الدورات العامة للف vandal

لا يمكن

الطة، التحليل

لدائم

الثاني

القانون

الأول

القانون

أساس

علي

علي

الحصول

القدرة

توليد

الأنظمة الحرارية.

يتكامل

الات

التي يتغير من كل تحليل، المنتجة للقدرة، كفاءة الحرارية، الدورات، التي يمكن أن تكون موجودة في المنظمة، يمكن أن تكون في اللف vandal يتم استخدام Excel لحساب النتائج التي يعرضها في اللف vandal يمكن أن تكون هذه الدورات كفاءة حسابها ببرنامج بواسطة Excel. Excel يعطي نتائج جيدة في هذا اللف vandal، يمكن أن تكون هذه الدورات كفاءة حسابها ببرنامج بواسطة Excel.

تعتبر الدورات العامة للف vandal

لا يمكن

الطة، التحليل

لدائم

الثاني

القانون

الأول

القانون

أساس

علي

علي

الحصول

القدرة

توليد

الأنظمة الحرارية.

يتكامل

الات

التي يتغير من كل تحليل، المنتجة للقدرة، كفاءة الحرارية، الدورات، التي يمكن أن تكون موجودة في المنظمة، يمكن أن تكون في اللف vandal يتم استخدام Excel لحساب النتائج التي يعرضها في اللف vandal يمكن أن تكون هذه الدورات كفاءة حسابها ببرنامج بواسطة Excel. Excel يعطي نتائج جيدة في هذا اللف vandal، يمكن أن تكون هذه الدورات كفاءة حسابها ببرنامج بواسطة Excel.

تعتبر الدورات العامة للف vandal

لا يمكن

الطة، التحليل

لدائم

الثاني

القانون

الأول

القانون

أساس

علي

علي

الحصول

القدرة

توليد

الأنظمة الحرارية.

يتكامل

الات

التي يتغير من كل تحليل، المنتجة للقدرة، كفاءة الحرارية، الدورات، التي يمكن أن تكون موجودة في المنظمة، يمكن أن تكون في اللف vandal يتم استخدام Excel لحساب النتائج التي يعرضها في اللف vandal يمكن أن تكون هذه الدورات كفاءة حسابها ببرنامج بواسطة Excel. Excel يعطي نتائج جيدة في هذا اللف vandal، يمكن أن تكون هذه الدورات كفاءة حسابها ببرنامج بواسطة Excel.

تعتبر الدورات العامة للف vandal

لا يمكن

الطة، التحليل

لدائم

الثاني

القانون

الأول

القانون

أساس

علي

علي

الحصول

القدرة

توليد

الأنظمة الحرارية.

يتكامل

الات

التي يتغير من كل تحليل، المنتجة للقدرة، كفاءة الحرارية، الدورات، التي يمكن أن تكون موجودة في المنظمة، يمكن أن تكون في اللف vandal يتم استخدام Excel لحساب النتائج التي يعرضها في اللف vandal يمكن أن تكون هذه الدورات كفاءة حسابها ببرنامج بواسطة Excel. Excel يعطي نتائج جيدة في هذا اللف vandal، يمكن أن تكون هذه الدورات كفاءة حسابها ببرنامج بواسطة Excel.

تعتبر الدورات العامة للف vandal

لا يمكن

الطة، التحليل

لدائم

الثاني

القانون

الأول

القانون

أساس

علي

علي

الحصول

القدرة

توليد

الأنظمة الحرارة
List of contents

<table>
<thead>
<tr>
<th>Subject</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>ACKNOWLEDGEMENT</td>
<td>I</td>
</tr>
<tr>
<td>ABSTRACT</td>
<td>II</td>
</tr>
<tr>
<td>LIST OF CONTENTS</td>
<td>V</td>
</tr>
<tr>
<td>LIST OF FIGURE</td>
<td>VIII</td>
</tr>
<tr>
<td>LIST OF TABLES</td>
<td>IX</td>
</tr>
<tr>
<td>ABBREVIATIONS</td>
<td>X</td>
</tr>
<tr>
<td>Nomenclature</td>
<td>XI</td>
</tr>
</tbody>
</table>

CHAPTER ONE .........................................

INTRODUCTION ....................................... Error! Bookmark not defined.

1.1. ELECTRICITY SUPPLY IN SUDAN Error! Bookmark not defined.

1.2. ELECTRICAL SYSTEM IN SUDAN ........ Error! Bookmark not defined.


1.3. SECOND-LAW OF THERMODYNAMIC ASPECTS OF DAILY LIFE Error! Bookmark not defined.

1.4. OBJECTIVES OF STUDY ........ Error! Bookmark not defined.

CHAPTER TWO .....................................

LITERATURE REVIEW Error! Bookmark not defined.

CHAPTER THREE ....................................

FUNDAMENTALS OF EXERGY ANALYSIS ........ Error! Bookmark not defined.

3.1 EXERGY: A MEASURE OF WORK POTENTIAL Error! Bookmark not defined.

3.2 EXERGY WORK POTENTIAL OF ENERGY Error! Bookmark not defined.

3.3 REVERSIBLE WORK AND IRREVERSIBILITY Error! Bookmark not defined.

3.4 SECOND-LAW EFFICIENCY, \( \eta \) ... Error! Bookmark not defined.

3.5 EXERGY CHANGE OF A SYSTEM Error! Bookmark not defined.

3.5.1 Exergy of a Fixed Mass: ........ Error! Bookmark not defined.

3.5.1.2 Exergy of a Flow Stream: Flow (or Stream Exergy ........ Error! Bookmark not defined.
3.6 THE DECREASE OF EXERGY PRINCIPLE AND EXERGY DESTRUCTION
3.6.1 Exergy Destruction
3.7 EXERGY BALANCE: CLOSED SYSTEMS
3.8 EXERGY BALANCE: CONTROL VOLUME
3.8.1 Exergy Balance for Steady-Flow System
3.8.2 Reversible Work
3.8.3 Second-Law Efficiency of Steady-Flow Devices
CHAPTER FOUR
GARRI 1 POWER PLANT
4.1. SITE LOCATION
4.2. WEATHER CONDITIONS
4.3. DESIGN OPERATING CONDITIONS
4.4. PLANT CONFIGURATIONS
4.5. OPERATING SCHEME OF THE CCPP
4.6. GAS TURBINE WITH EXHAUST GAS SYSTEM
4.6.1 Gas Turbine
4.6.2 Exhaust Gas System
4.7. WATER STEAM CYCLE
4.8. HRSG AND ITS AUXILIARY SYSTEM
4.9. THE MAIN STEAM AND BYPASS SYSTEM
4.10. DEAERATOR HEATING STEAM SYSTEM
4.11. BOILER FEED-WATER SYSTEM
4.12. CONDENSATE SYSTEM
4.13. CIRCULATING COOLING WATER SYSTEM
4.14. AUXILIARY SYSTEMS
4.19. HEAT RECOVERY STEAM GENERATOR (HRSG)
4.19.1 General
4.19.2 Type of Boiler
4.19.3 Boiler Performance Features......Error! Bookmark not defined.
4.19.4 Boiler Inlet Duct, Main Stack and Duct Accessories ...... Error!
Bookmark not defined.
4.20. STEAM TURBINE ..........................Error! Bookmark not defined.
CHAPTER FIVE .......................................Error! Bookmark not defined.
EXERGY ANALYSIS FOR GARRI 1 POWER PLANT USING EXCEL
.....................................................................Error! Bookmark not defined.
5.1. EXERGY ANALYSIS FOR STEAM CYCLE (TABLES (5-1,2,3))
.....................................................................Error! Bookmark not defined.
5.1.1. HRSG:.........................................Error! Bookmark not defined.
5.1.2. Steam turbine (ST): ......................Error! Bookmark not defined.
5.1.3. Condenser................................Error! Bookmark not defined.
5.1.4. Feeding system (pumps)...............Error! Bookmark not defined.
5.2. EXERGY ANALYSIS OF MAIN EQUIPMENT FOR GARRI
POWER PLANT (TABLE (5-4)).........Error! Bookmark not defined.
5.2.1. Gas turbine one set ......................Error! Bookmark not defined.
5.2.2. HRSG one set..............................Error! Bookmark not defined.
5.2.3. Steam turbine set (combined by 2 HRSGs)Error! Bookmark not defined.
CHAPTER SIX.................................Error! Bookmark not defined.
RESULTS AND DISCUSSION .................Error! Bookmark not defined.
6.1. EXERGY ANALYSIS OF CCPP ....Error! Bookmark not defined.
6.1.1. Brayton cycle theoretical efficiencyError! Bookmark not defined.
6.1.2. Rankine cycle theoretical efficiencyError! Bookmark not defined.
6.2. EFFECT OF PARTIAL LOAD ON CCPP EFFICIENCY..... Error!
Bookmark not defined.
6.3. THERMODYNAMICS ANALYSIS Error! Bookmark not defined.
6.4. EVALUATION OF TEMPERATURE AND HEAT ABSORPTION
FOR HRSG ..............................................Error! Bookmark not defined.
6.5. EFFECT OF PINCH POINT ON PLANT PERFORMANCE Error!
Bookmark not defined.
CHAPTER SEVEN ...............................Error! Bookmark not defined.
CONCLUSION AND RECOMMENDATIONError! Bookmark not defined.
7.1. CONCLUSION ...............................Error! Bookmark not defined.
7.2. RECOMMENDATION........................Error! Bookmark not defined.
References ..............................................Error! Bookmark not defined.
List of figures

Subject                                                                 page
1-1 High load forecast planting programme ----------------------------------- 8
2-1 Schematic flow diagram for Plant A -------------------------------------- 17
2-2 Schematic flow diagram for Plant B -------------------------------------- 18
2-3 Schematic diagram of EGAT-Block 1 (Combined cycle) ------------------- 23
2-4 Schematic diagram of the combined Brayton/Rankine power cycle Reheat ----------------------------------------------- 25
3-1 The isolated system considered in the development of the decrease of Exergy principle ----------------------------------------------- 46
3-2 Exergy is transferred into or out of a control volume by mass as work and heat transfer ---------------------------------- 52
3-3 A heat exchanger with two unmixed streams ---------------------------- 57
4-1 GARRI 1 Block configuration ------------------------------------------ 62
6-1 T–S diagram of combined cycle plant ----------------------------------- 93
6-2 Thermodynamic analysis of Garri 1 ---------------------------------- 98
6-3 Gas and steam temperature profile of HRSG for 100 % load LDO ---- 100
6-4 Gas and steam temperature profile of HRSG for 100% load LPG ---- 100
6-5 Gas and steam temperature profile of HRSG for 66% load LDO ----- 101
## List of tables

<table>
<thead>
<tr>
<th>Subject</th>
<th>page</th>
</tr>
</thead>
<tbody>
<tr>
<td>1-1 Existing thermal and hydro power capacity in the national grid</td>
<td>5</td>
</tr>
<tr>
<td>2005</td>
<td></td>
</tr>
<tr>
<td>1-2 Isolated power plants</td>
<td>6</td>
</tr>
<tr>
<td>1-3 Generation in the five years plan (2005-2010)</td>
<td>7</td>
</tr>
<tr>
<td>1-4 High load forecast planting programme</td>
<td>9</td>
</tr>
<tr>
<td>2-1 Description of plant A and B</td>
<td>18</td>
</tr>
<tr>
<td>2-2 Gas turbine design information for plant A and B</td>
<td>19</td>
</tr>
<tr>
<td>2-3 Conditions and parameters used in the calculation</td>
<td>22</td>
</tr>
<tr>
<td>4-1 Boiler all heating surfaces thermodynamic special property</td>
<td>72</td>
</tr>
<tr>
<td>(design operating conditions) GT fuel LDO</td>
<td></td>
</tr>
<tr>
<td>4-2 Boiler all heating surfaces thermodynamic special property</td>
<td>73</td>
</tr>
<tr>
<td>(design operating conditions) GT fuel LPG</td>
<td></td>
</tr>
<tr>
<td>4-3 Boiler all heating surfaces thermodynamic special property</td>
<td>74</td>
</tr>
<tr>
<td>(66% load operating conditions) GT fuel LDO</td>
<td></td>
</tr>
<tr>
<td>4-4 The list of summary of data for CCPP (100% load condition)</td>
<td>75</td>
</tr>
<tr>
<td>4-5 Main performance data (100% load condition)</td>
<td>77</td>
</tr>
<tr>
<td>5-1 Exergy analysis for steam cycle (gas turbine fuel light desil oil</td>
<td>85</td>
</tr>
<tr>
<td>for 100% load)</td>
<td></td>
</tr>
<tr>
<td>5-2 Exergy analysis for steam cycle (gas turbine fuel LPG)</td>
<td>86</td>
</tr>
<tr>
<td>for 100% load)</td>
<td></td>
</tr>
<tr>
<td>5-3 Exergy analysis for steam cycle (gas turbine fuel light desil oil</td>
<td>87</td>
</tr>
<tr>
<td>for 66% load)</td>
<td></td>
</tr>
<tr>
<td>5-4 Exergy analysis of main equipment for garri power plant</td>
<td>88</td>
</tr>
<tr>
<td>6-1 Exergy destruction for garri 1 components</td>
<td>96</td>
</tr>
</tbody>
</table>
## Abbreviation

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

Subscript:
- **i** inlet
- **o** outlet
Nomenclature

\[ P \] pressure absolute (MPa)
\[ P_0 \] atmospheric pressure (MPa)
\[ T \] temperature (°C)
\[ T_0 \] atmospheric temperature (°C)
\[ \psi \] exergy (kJ/kg)
\[ \psi_f \] fuel exergy (kJ/s)
\[ \psi_p \] process heat exergy (kJ/s)
\[ \varepsilon_f \] fuel exergy factor
\[ \varepsilon_s \] 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)
\[ \eta_{CC} \] thermal efficiency for combined cycle plant
\[ \eta_{GT} \] thermal efficiency for gas turbine
\[ \eta_{HRSG} \] thermal efficiency for heat recovery steam generator
\[ \eta_{ST} \] thermal efficiency for steam cycle
\[ \eta_{Rankine} \] thermal efficiency for Rankine cycle
\[ \eta_i \] first law efficiency
\[ \eta_{II} \] 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:

<table>
<thead>
<tr>
<th>Project</th>
<th>Capacity</th>
<th>Status</th>
</tr>
</thead>
<tbody>
<tr>
<td>Garri 2 upgrading</td>
<td>100 MW.</td>
<td>Implemented.</td>
</tr>
<tr>
<td>Khartoum North</td>
<td>200 MW.</td>
<td>Under Construction</td>
</tr>
<tr>
<td>Kosti</td>
<td>500 MW.</td>
<td>Under Construction</td>
</tr>
<tr>
<td>Garri -4</td>
<td>100 MW.</td>
<td>Under Construction</td>
</tr>
</tbody>
</table>

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

<table>
<thead>
<tr>
<th>Description</th>
<th>MW</th>
</tr>
</thead>
<tbody>
<tr>
<td>TOTAL OF HYDRO</td>
<td>343.3</td>
</tr>
<tr>
<td>TOTAL OF THERMAL</td>
<td>730.7</td>
</tr>
<tr>
<td>TOTAL GENERATION</td>
<td>1074</td>
</tr>
</tbody>
</table>
Table (1-1): Existing thermal and hydro power capacity in the national grid 2005 [1].

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

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


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].

<table>
<thead>
<tr>
<th>Year</th>
<th>Additional Generation Plan (2006-2010)</th>
<th>MW</th>
</tr>
</thead>
<tbody>
<tr>
<td>2006</td>
<td>35 MW CCGT Garri (2)</td>
<td>35</td>
</tr>
<tr>
<td></td>
<td>70 MW CCGT Garri (2)</td>
<td>70</td>
</tr>
<tr>
<td></td>
<td>2x35 MW Fast Track Kilo (x)</td>
<td>70</td>
</tr>
<tr>
<td></td>
<td>1×55 MW - 1\textsuperscript{st} Unit Garri (4)</td>
<td>55</td>
</tr>
<tr>
<td></td>
<td>2×125 MW Morewe Hydro. Main Nile</td>
<td>250</td>
</tr>
<tr>
<td></td>
<td>2×25 Kassala Diesel</td>
<td>50</td>
</tr>
<tr>
<td>2007</td>
<td>1x100 MW Steam Kh. N (Phase 3)</td>
<td>100</td>
</tr>
<tr>
<td></td>
<td>2×125 MW Steam Kosti</td>
<td>250</td>
</tr>
<tr>
<td></td>
<td>2×135 MW Steam Generation at EL Fula</td>
<td>270</td>
</tr>
<tr>
<td></td>
<td>1×55 MW 2\textsuperscript{nd} Unit Garri (4)</td>
<td>55</td>
</tr>
<tr>
<td>2008</td>
<td>2×135 MW Garri(3)</td>
<td>270</td>
</tr>
<tr>
<td></td>
<td>1x100 MW Steam Kh. N (Phase 3)</td>
<td>100</td>
</tr>
<tr>
<td></td>
<td>2x135 MW Steam ELBagair</td>
<td>270</td>
</tr>
<tr>
<td></td>
<td>2X 135 MW - Coal/generation Port Sudan</td>
<td>270</td>
</tr>
<tr>
<td></td>
<td>3×125 MW Morewe Hydro</td>
<td>375</td>
</tr>
<tr>
<td></td>
<td>1×135 MW Steam Generation at EL Fula</td>
<td>135</td>
</tr>
<tr>
<td></td>
<td>1x125 MW Steam Kosti</td>
<td>125</td>
</tr>
<tr>
<td>2009</td>
<td>2X 135 MW - Coal/generation Port Sudan</td>
<td>270</td>
</tr>
<tr>
<td></td>
<td>2×135 MW Garri(3)</td>
<td>270</td>
</tr>
<tr>
<td></td>
<td>1×135 MW Steam Generation at EL Fula</td>
<td>135</td>
</tr>
<tr>
<td></td>
<td>3×125 MW Morewe Hydro</td>
<td>375</td>
</tr>
<tr>
<td></td>
<td>2x135 MW Steam ELBagair</td>
<td>270</td>
</tr>
<tr>
<td>2010</td>
<td>1x125 MW Steam Kosti</td>
<td>125</td>
</tr>
</tbody>
</table>
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 a gap between the generation and demand.

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

<table>
<thead>
<tr>
<th>YEAR</th>
<th>TOTAL CAPACITY (MW)</th>
<th>DEMAND (MW)</th>
<th>RESERVE MARGINE (MW)</th>
<th>RESERVE MARGINE (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>2006</td>
<td>871</td>
<td>1228</td>
<td>-357</td>
<td>-29</td>
</tr>
<tr>
<td>2007</td>
<td>1187</td>
<td>2007</td>
<td>-820</td>
<td>-41</td>
</tr>
<tr>
<td>2008</td>
<td>1627</td>
<td>2699</td>
<td>-1072</td>
<td>-40</td>
</tr>
<tr>
<td>2009</td>
<td>4333</td>
<td>3549</td>
<td>784</td>
<td>22</td>
</tr>
<tr>
<td>2010</td>
<td>5486</td>
<td>4354</td>
<td>1132</td>
<td>26</td>
</tr>
</tbody>
</table>

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 their 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 law 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 output (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 reversible 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 cogeneration based on $T_0 = 15^\circ\text{C}$ and $P_0 = 101\text{ kPa}$ 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 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 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

<table>
<thead>
<tr>
<th></th>
<th>Plant A</th>
<th>Plant B</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Plant type</strong></td>
<td>5 combined cycle units</td>
<td>1 cogeneration cycle unit</td>
</tr>
<tr>
<td><strong>Combination</strong></td>
<td>3GTGs+3HRSGs+1STG</td>
<td>3 aero derivative GTGs+3HRSG+1STG</td>
</tr>
<tr>
<td><strong>Total installation capacity (MW)</strong></td>
<td>2200</td>
<td>150+45 ton/h steam</td>
</tr>
<tr>
<td><strong>Site elevation (meter above MSL)</strong></td>
<td>4</td>
<td>138</td>
</tr>
<tr>
<td><strong>Annual average RH(%)</strong></td>
<td>80</td>
<td>78.5</td>
</tr>
<tr>
<td><strong>Highest/lowest daily T(°C)</strong></td>
<td>35.4/7.4</td>
<td>38/6</td>
</tr>
<tr>
<td><strong>Average annual T(°C)</strong></td>
<td>25.1</td>
<td>22.3</td>
</tr>
<tr>
<td><strong>Design condition (°C and %RH)</strong></td>
<td>32 and 90</td>
<td>32 and 90</td>
</tr>
<tr>
<td><strong>Preheated fuel gas T(°C)</strong></td>
<td>120</td>
<td>N/A</td>
</tr>
<tr>
<td><strong>Air inlet cooling T(°C)</strong></td>
<td>N/A</td>
<td>5.5+0.5</td>
</tr>
<tr>
<td><strong>Commercially operated</strong></td>
<td>January 1999</td>
<td>January 2000</td>
</tr>
</tbody>
</table>
Table 2-2 gas turbine design information for plants A and B

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

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

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

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.

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

\[ \text{Work} = f(\text{initial state, process path, final state}) \]

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 \( T_0 = 40\,^\circ\text{C} \) and \( P_0 = 0.966 \text{ bar} \). 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 explained 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 device will actually deliver upon installation. Rather, it represents the upper limit on the amount of work a device 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 alone, 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,
\[ W_{\text{surr}} = P_0(V_2 - V_1) \] (3-1)

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

\[ W_u = W - W_{\text{surr}} = W - P_0(V_2 - V_1) \] (3-2)

When a system is expanding and doing work, part of the work done is used to overcome the atmospheric pressure and thus \( W_{\text{surr}} \) represents a loss. When a system is compressed, however, the atmospheric pressure helps the compression process, and thus \( W_{\text{surr}} \) 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 \( W_{\text{rev}} \) 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 $W_{\text{rev}}$ 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:

$$I = W_{\text{rev, out}} - W_{u, out} \quad \text{or} \quad I = W_{u, in} - W_{\text{rev, in}} \quad (3-3)$$

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 $W_{\text{rev}} \geq W_u$ for work producing devices and $W_{\text{rev}} \leq W_u$ 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, $\eta_{II}$

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 \( \eta_{II} \) as the ratio of the actual thermal efficiency to the maximum possible (reversible) thermal efficiency under the same conditions:

\[ \eta_{II} = \frac{\eta_{sh}}{\eta_{sh,rev}} \quad \text{(heat engines)} \]  

(3-4)

Based on this definition, the second law efficiencies of the two heat engines discussed above are
\[ \eta_u = \frac{0.30}{0.50} = 0.60 \quad \text{and} \quad \eta_u = \frac{0.30}{0.70} = 0.43 \]

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:

\[ \eta_u = \frac{W_u}{W_{rev}} \quad \text{(work producing devices)} \quad (3-5) \]

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:

\[ \eta_u = \frac{W_{rev}}{W_u} \quad \text{(work consuming devices)} \quad (3-6) \]

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

\[ \eta_u = \frac{COP}{COP_{rev}} \quad \text{(refrigerators and heat pumps)} \quad (3-7) \]

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 mind, we defined the second law efficiency of a system during a process as

\[ \eta_s = \frac{\text{exergy recovered}}{\text{exergy supplied}} = 1 - \frac{\text{exergy destroyed}}{\text{exergy supplied}} \quad (3-8) \]

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 interaction 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 low 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 referred 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 across 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
\[ \partial E_{in} - \partial E_{out} = \partial E_{system} \quad (3-9) \]

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 \( \partial W = PDV \).

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,

\[ \partial W = PDV = (P - P_0)dV + P_0dV = \partial W_{b,\text{useful}} + P_0dV \quad (3-10) \]

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

\[ \partial W_{HE} = (1 - T_0/T)\partial Q = \partial Q - (T_0/T)\partial Q = \partial Q - (-T_0\partial S) = \partial Q = \partial W_{HE} - T_0\partial S \quad (3-11) \]
Substituting the $\partial W$ and $\partial Q$ expression in Eqs. 3-10 and 3-11 into the energy balance relation Eqs. 3-11 gives after rearranging,

$$\partial W_{\text{total,useful}} = \partial W_{\text{HE}} + \partial W_{\text{h,useful}} = -dU - P_0dV + T_0dS$$

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

$$W_{\text{total,useful}} = (U - U_0) + P_0(V - V_0) - T_0(S - S_0)$$  \hspace{1cm} (3-12)

Where $W_{\text{total,useful}}$ 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

$$X = (U - U_0) + P_0(V - V_0) - T_0(S - S_0) + m C^2/2 + mgz$$  \hspace{1cm} (3-13)

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

$$\Phi = (u - u_0) + P_0(v - v_0) - T_0(s - s_0) + C^2/2 + gz$$  \hspace{1cm} (3-14)

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

\[
\Delta X = X_2 - X_1 = m(\Phi_2 - \Phi_1) = (E_2 - E_1) + P_0(V_2 - V_1) - T_0(S_2 - S_1) \\
= (U_2 - U_1) + P_0(V_2 - V_1) - T_0(S_2 - S_1) + m(C_v^2 - C_l^2)/2 + mg(Z_2 - Z_1)
\]

Or, on a unit mass basis,

\[
\Delta \Phi = \Phi_2 - \Phi_1 \\
= (u_2 - u_1) + P_0(v_2 - v_1) - T_0(s_2 - s_1) + (C_v^2 - C_l^2)/2 + g(z_2 - z_1) \\
= (e_2 - e_1) + P_0(v_2 - v_1) - T_0(s_2 - s_1)
\]

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

\[
X_{\text{system}} = \int \Phi \rho dV = \int v \Phi \rho dV
\]

Where \( V \) is the volume of the system and \( \rho \) 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 \(w_{\text{flow}} = P \nu\) where \(\nu\) 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 \(\nu\). Noting that the flow work is \(P \nu\) and the work done against the atmospheric is \(P_0 \nu\), the exergy of flow energy can be expressed as

\[
 x_{\text{flow}} = P \nu - P_0 \nu = (P - P_0) \nu \tag{3-18}
\]

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.
\[ x_{\text{flowing fluid}} = x_{\text{nonflowing fluid}} + x_{\text{flow}} \]  
\[ = (u - u_0) + P_0(v - v_0) - T_0(s - s_0) + C^2/2 + gz + (P - P_0)v \]  
\[ = (u + P_0) - (u_0 + P_0v_0) - T_0(s - s_0) + C^2/2 + gz \]  
\[ = (h - h_0) - T_0(s - s_0) + C^2/2 + gz \]

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

**Flowexergy:** \( \psi = (h - h_0) - T_0(s - s_0) + C^2/2 + gz \)  
\( (3-20) \)

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

\[ \Delta \psi = \psi_2 - \psi_1 = (h_2 - h_1) - T_0(s_2 - s_1) + (C^2_2 - C^2_1)/2 + g(z_2 - z_1) \]  
\( (3-21) \)

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 \( W_{\text{rev}} \). 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 DESTRUCTION

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

\[
\begin{align*}
\text{Energy balance:} & \quad E_{in} - E_{out} = \Delta E_{system} \rightarrow 0 = E_2 - E_1 \\
\text{Entropy balance:} & \quad S_{in} - S_{out} + S_{gen} = \Delta S_{system} \rightarrow S_{gen} = S_2 - S_1
\end{align*}
\]

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

\[-T_0 S_{gen} = E_2 - E_1 - T_0 (S_2 - S_1) \quad (3-22)\]

From equation 3-15 we have

\[X_2 - X_1 = (E_2 - E_1) + P_0 (V_2 - V_1) - T_0 (S_2 - S_1) \quad (3-23)\]
\[ = (E_2 - E_1) - T_0(S_2 - S_1) \]

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

\[ T_0S_{\text{gen}} = X_2 - X_1 \quad (3-24) \]

Since \( T_0 \) is the absolute temperature of the environment and thus a positive quantity, \( S_{\text{gen}} \geq 0 \), and thus \( T_0S_{\text{gen}} \geq 0 \). Then we conclude that

\[ \Delta X_{\text{isolated}} = (X_2 - X_1)_{\text{isolated}} \leq 0 \quad (3-25) \]

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.
No heat, work or mass transfer

Isolated system

\[ \Delta X_{\text{isolated}} \leq 0 \]

(or \( X_{\text{destroyed}} \geq 0 \))

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

\[ X_{\text{destroyed}} = T_0 S_{\text{gen}} \geq 0 \]  \hspace{1cm} (3-26)
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 \( (X_{\text{destroyed, rev}} = 0) \).

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:

\[
X_{\text{destroyed}} = \begin{cases} 
> 0 & \text{Irreversible process} \\
= 0 & \text{Reversible process} \\
< 0 & \text{Impossible process}
\end{cases} \quad (3-27)
\]

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

\[
\begin{align*}
\begin{pmatrix}
\text{Total exergy entering} \\
\text{Total exergy leaving} \\
\text{Total exergy destroyed}
\end{pmatrix}
&= \begin{pmatrix}
\text{Change in the total exergy of the system}
\end{pmatrix} \\
\begin{array}{c}
\text{Change in the total exergy of the system}
\end{array}
\end{align*}
\]

Or

\[
X_{in} - X_{out} - X_{destroyed} = \Delta X_{system}
\]

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:

\[
(X_{in} - X_{out}) - X_{destroyed} = \Delta X_{system} \quad \text{(kJ)}
\]  

Net exergy transfer  exergy change
by heat, work, and mass  destruction in exergy
Or in the rate form, as

*General, rate form:*

\[(\dot{X}_{in} - \dot{X}_{out}) - \dot{X}_{destroyed} = \Delta \dot{X}_{system} \quad \text{(kW)} \tag{3-30}\]

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

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

\[\dot{X}_{heat} = (1 - T_0/T)\dot{Q}, \quad \dot{X}_{work} = \dot{W}_{useful} \quad \text{and} \quad \dot{X}_{mass} = \dot{m} \psi, \quad \text{respectively, and} \]

\[\Delta X_{system} = dX_{system}/dt. \quad \text{The exergy balance can also be expressed per unit mass as} \]

*General, unit-mass basis:*

\[(x_{in} - x_{out}) - x_{destroyed} = \Delta x_{system} \quad \text{(kJ/kg)} \tag{3-31}\]

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,

\[X_{destroyed} = T_0 S_{gen} \quad \text{or} \quad \dot{X}_{destroyed} = T_0 \dot{S}_{gen} \tag{3-32}\]

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

\[
\text{Closed system: } X_{\text{heat}} - X_{\text{work}} - X_{\text{destroyed}} = \Delta X_{\text{system}} \quad (3-33)
\]

Or

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

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 \( \Delta t \) and taking the limit as \( \Delta t \to 0 \) gives the rate form of the exergy balance for a closed system,

\[
\text{Rate form: } \\
\sum (1 - \frac{T_0}{T_k}) \dot{Q}_K - \left[ \dot{W} - P_0 \frac{dV_{\text{system}}}{dt} \right] - T_0 \dot{S}_{\text{gen}} = \frac{dX_{\text{system}}}{dt} \quad (3-35)
\]

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 \( W_{\text{rev}} \) by setting the exergy destruction term equal to zero. The work \( W \) in that case becomes the reversible work. That is, \( W = W_{\text{rev}} \) when \( X_{\text{destroyed}} = T_0 S_{\text{gen}} = 0 \).

Note that \( X_{\text{destroyed}} \) 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 \( X_{\text{destroyed}} = 0 \) 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 $T_0$.

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 across 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

\[
X_{\text{heat}} - X_{\text{work}} + X_{\text{mass,in}} - X_{\text{mass,out}} - X_{\text{destroyed}} = (X_2 - X_1)_{CV}
\]

(3-36)

Or

\[
\sum (1 - \frac{T_0}{T_k})Q_k - [W - P_0 (V_2 - V_1)] + \sum m_i \psi_i - \sum m_e \psi_e - X_{\text{destroyed}} = (X_2 - X_1)_{CV}
\]

(3-37)

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

\[
\sum (1 - \frac{T_0}{T_k})\dot{Q}_k - [\dot{W} - P_0 dV_{cv} / dt] + \sum \dot{m}_i \psi_i - \sum \dot{m}_e \psi_e - \dot{X}_{\text{destroyed}} = dX_{cv} / dt
\]

(3-38)

**Surrounding**

![Diagram of control volume with exergy flows](image-url)
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:**

$$\sum (1 - \frac{T_0}{T_k}) \dot{Q}_k - \dot{W} + \sum \dot{m}_i \psi_i - \sum \dot{m}_e \psi_e - \dot{X}_{destroyed} = 0 \quad (3-39)$$

For a single-stream (one-inlet, one-exit) steady-flow device, the relation above further reduce to
\[
\sum (1 - \frac{T_0}{T_k}) \dot{Q}_k - W + \dot{m}(\psi_1 - \psi_2) - \dot{X}_{\text{destroyed}} = 0 \quad (3-40)
\]

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:**

\[
\sum (1 - \frac{T_0}{T_k}) q_k - w + (\psi_1 - \psi_2) - x_{\text{destroyed}} = 0 \quad (kJ/kg) \quad (3-41)
\]

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 \( \dot{X}_{\text{destroyed}} = \dot{m}(\psi_1 - \psi_2) \), 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, \( W_{\text{rev}} \)**

The exergy balance relation presented above can be used to determine the reversible work \( W_{\text{rev}} \) by setting the exergy destroyed equal to zero. The work \( W \) in that case becomes the reversible work. That is,
General: \( W = W_{\text{rev}} \) when \( X_{\text{destroyed}} = 0 \) \hspace{1cm} (3-42)

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

\[
\text{Single-stream: } W_{\text{rev}} = \dot{m}(\psi_1 - \psi_2) + \sum (1 - \frac{T_0}{T_k}) \dot{Q}_k \text{ (kW)} \hspace{1cm} (3-43)
\]

Which reduces for an adiabatic device to

\[
\text{Adiabatic, single stream: } \dot{W}_{\text{rev}} = \dot{m}(\psi_1 - \psi_2) \hspace{1cm} (3-44)
\]

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, \( \eta_{II} \)

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

\[
\eta_{I\text{l,turb}} = \frac{\dot{w}}{w_{\text{rev}}} = \frac{h_1 - h_2}{\psi_1 - \psi_2} \hspace{1cm} \text{or} \hspace{1cm} \eta_{II,turb} = 1 - \frac{T_0 S_{\text{gen}}}{\psi_1 - \psi_2} \hspace{1cm} (3-45)
\]

Where \( S_{\text{gen}} = S_2 - S_1 \). For an adiabatic compressor with negligible kinetic and potential energies, the second-law efficiency becomes
\[
\eta_{II,\text{comp}} = \frac{\dot{w}_{\text{rev,in}}}{\dot{w}_{\text{in}}} = \frac{\psi_2 - \psi_1}{h_2 - h_1} \quad \text{or} \quad \eta_{II,\text{comp}} = 1 - \frac{T_0 S_{\text{gen}}}{h_2 - h_1} \tag{3-46}
\]

Where again \( S_{\text{gen}} = S_2 - S_1 \).

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

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

where \( \dot{S}_{\text{gen}} = \dot{m}_{\text{hot}} (s_2 - s_1) / \dot{m}_{\text{cold}} (s_4 - s_3) \).

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

\[
\eta_{II,\text{mix}} = \frac{\dot{m}_3 \psi_3}{\dot{m}_1 \psi_1 + \dot{m}_2 \psi_2} \quad \text{or} \quad \eta_{II,\text{mix}} = 1 - \frac{T_0 \dot{S}_\text{gen}}{\dot{m}_1 \psi_1 + \dot{m}_2 \psi_2}
\]

(3-48)

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

\[ T_0 \]

\[ \begin{array}{c}
1 \\
\text{Hot Stream}
\end{array} \hspace{5cm} \begin{array}{c}
4 \\
\text{4}
\end{array} \hspace{5cm} \begin{array}{c}
2 \\
\text{3}
\end{array} \hspace{5cm} \begin{array}{c}
\text{Cold Stream}
\end{array} \]

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 about 70 km north from Khartoum and the distance between the refinery and the Nile River is about 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

<table>
<thead>
<tr>
<th>Description</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Many-year average atmosphere pressure</td>
<td>964.1 mbar</td>
</tr>
<tr>
<td>Many-year average atmosphere temperature</td>
<td>30.8°C</td>
</tr>
<tr>
<td>Extreme highest atmosphere temperature</td>
<td>47.2°C</td>
</tr>
<tr>
<td>Extreme lowest atmosphere temperature</td>
<td>6.0°C</td>
</tr>
<tr>
<td>Many-year average highest atmosphere temperatures</td>
<td>37.1°C</td>
</tr>
<tr>
<td>Many-year average lowest atmosphere temperature</td>
<td>22.7°C</td>
</tr>
<tr>
<td>Average relative humidity</td>
<td>29%</td>
</tr>
<tr>
<td>Many-year average rainfall</td>
<td>162.4 mm</td>
</tr>
<tr>
<td>Mean wind velocity</td>
<td>3.9 m/s</td>
</tr>
<tr>
<td>Max. wind velocity</td>
<td>32 m/s</td>
</tr>
<tr>
<td>All-year wind prevalent direction</td>
<td>north wind</td>
</tr>
<tr>
<td>Seismic intensity</td>
<td>class VI</td>
</tr>
</tbody>
</table>

4.3. DESIGN OPERATING CONDITIONS

<table>
<thead>
<tr>
<th>Description</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Ambient temperature</td>
<td>40°C</td>
</tr>
<tr>
<td>Atmospheric pressure</td>
<td>0.966 bar</td>
</tr>
<tr>
<td>Relative humidity</td>
<td>38%</td>
</tr>
</tbody>
</table>
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.
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 ensure 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 m².

(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 at temperature.
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. AUXILIARY 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

<table>
<thead>
<tr>
<th>No.</th>
<th>Item</th>
<th>Unit</th>
<th>HP superheater</th>
<th>HP evaporating Tube Bank</th>
<th>HP Economizer</th>
<th>LP boiler</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Ambient temperature</td>
<td>°C</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>2</td>
<td>Inlet flue gas temperature</td>
<td>°C</td>
<td>564</td>
<td>488.1</td>
<td>295</td>
<td>190.2</td>
</tr>
<tr>
<td>3</td>
<td>Outlet flue gas temperature</td>
<td>°C</td>
<td>488.1</td>
<td>295</td>
<td>190.2</td>
<td>158.1</td>
</tr>
<tr>
<td>4</td>
<td>Inlet medium temperature</td>
<td>°C</td>
<td>286.4</td>
<td>286.4</td>
<td>104</td>
<td>144.1</td>
</tr>
<tr>
<td>5</td>
<td>Outlet medium temperature</td>
<td>°C</td>
<td>468</td>
<td>286.4</td>
<td>279.5</td>
<td>144.1</td>
</tr>
<tr>
<td>6</td>
<td>Average temperature difference</td>
<td>°C</td>
<td>130.6</td>
<td>61.2</td>
<td>41.3</td>
<td>27</td>
</tr>
<tr>
<td>7</td>
<td>Average flue gas temperature</td>
<td>°C</td>
<td>512.3</td>
<td>347.7</td>
<td>233</td>
<td>171</td>
</tr>
<tr>
<td>8</td>
<td>Average flue gas speed</td>
<td>m/s</td>
<td>17.1</td>
<td>14.7</td>
<td>11.9</td>
<td>10.3</td>
</tr>
</tbody>
</table>
Table 4-2: BOILER ALL HEATING SURFACES THERMODYNAMIC SPECIAL PROPERTY (DESIGN OPERATING CONDITIONS).

GT. Model: PG6581B            G.T Fuel: LPG

<table>
<thead>
<tr>
<th>No.</th>
<th>Item</th>
<th>Unit</th>
<th>HP superheater</th>
<th>HP evaporating Tube Bank</th>
<th>HP Economizer</th>
<th>LP boiler</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Ambient temperature</td>
<td>°C</td>
<td></td>
<td></td>
<td></td>
<td>40</td>
</tr>
<tr>
<td>2</td>
<td>Inlet flue gas</td>
<td>°C</td>
<td>564</td>
<td>488.2</td>
<td>297.3</td>
<td>190.9</td>
</tr>
<tr>
<td></td>
<td>temperature</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>3</td>
<td>Outlet flue gas</td>
<td>°C</td>
<td>488.2</td>
<td>297.3</td>
<td>190.9</td>
<td>158.5</td>
</tr>
<tr>
<td></td>
<td>gas temperature</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>4</td>
<td>Inlet medium</td>
<td>°C</td>
<td>288.6</td>
<td>288.6</td>
<td>104</td>
<td>144.3</td>
</tr>
<tr>
<td></td>
<td>temperature</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>5</td>
<td>Outlet medium</td>
<td>°C</td>
<td>468</td>
<td>288.6</td>
<td>281.6</td>
<td>144.3</td>
</tr>
<tr>
<td></td>
<td>medium temperature</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>6</td>
<td>Average temperature</td>
<td>°C</td>
<td>129.8</td>
<td>60.8</td>
<td>41.6</td>
<td>27.2</td>
</tr>
<tr>
<td></td>
<td>difference</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>7</td>
<td>Average flue gas</td>
<td>°C</td>
<td>512.7</td>
<td>349.4</td>
<td>234.4</td>
<td>171.5</td>
</tr>
<tr>
<td></td>
<td>temperature</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>8</td>
<td>Average flue gas</td>
<td>m/s</td>
<td>17.2</td>
<td>14.8</td>
<td>12</td>
<td>10.4</td>
</tr>
<tr>
<td></td>
<td>gas speed</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>
Table 4-3: BOILER ALL HEATING SURFACES THERMODYNAMIC SPECIAL PROPERTY (66% LOAD OPERATING CONDITIONS)

G.T. Fuel: light diesel oil

<table>
<thead>
<tr>
<th>Item</th>
<th>Unit</th>
<th>HP Superheater</th>
<th>HP Evaporating pipe bank</th>
<th>HP Economizer</th>
<th>LP boiler</th>
</tr>
</thead>
<tbody>
<tr>
<td>Inlet flue gas temperature</td>
<td>°C</td>
<td>564</td>
<td>488.1</td>
<td>295.0</td>
<td>190.2</td>
</tr>
<tr>
<td>Outlet flue gas temperature</td>
<td>°C</td>
<td>488.1</td>
<td>295.0</td>
<td>190.2</td>
<td>158.1</td>
</tr>
<tr>
<td>Inlet medium temperature</td>
<td>°C</td>
<td>267.22</td>
<td>267.22</td>
<td>104</td>
<td>113.32</td>
</tr>
<tr>
<td>Outlet medium temperature</td>
<td>°C</td>
<td>265</td>
<td>267.22</td>
<td>263</td>
<td>113.32</td>
</tr>
</tbody>
</table>
Table 4-4: THE LIST OF SUMMARY OF DATA FOR CCPP (100% LOAD CONDITION)

<table>
<thead>
<tr>
<th>NO.</th>
<th>Item</th>
<th>Unit</th>
<th>Unit 1 &amp; Unit2</th>
<th>Remarks</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Gas Turbine</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>1.1</td>
<td>Type</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>1.2</td>
<td>Model</td>
<td></td>
<td>PG6001B</td>
<td></td>
</tr>
<tr>
<td>1.3</td>
<td>Fuel</td>
<td></td>
<td>LDO</td>
<td>LPG</td>
</tr>
<tr>
<td>1.4</td>
<td>Output at standard design condition</td>
<td>kW</td>
<td>39174</td>
<td>40053</td>
</tr>
<tr>
<td>1.5</td>
<td>Output at rated design Condition</td>
<td>kW</td>
<td>31602</td>
<td>23366</td>
</tr>
<tr>
<td>1.6</td>
<td>Quantities</td>
<td>set</td>
<td>4</td>
<td></td>
</tr>
<tr>
<td>1.7</td>
<td>Initial temperature</td>
<td>°C</td>
<td>40</td>
<td>40</td>
</tr>
<tr>
<td>1.8</td>
<td>Exhaust temperature</td>
<td>°C</td>
<td>564</td>
<td>564</td>
</tr>
<tr>
<td>1.9</td>
<td>Exhaust flow rate</td>
<td>t/h</td>
<td>452</td>
<td>450</td>
</tr>
<tr>
<td>2</td>
<td>Heat recovery steam Generator</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>2.1</td>
<td>Type</td>
<td></td>
<td>Single pressure, horizontal firing</td>
<td></td>
</tr>
<tr>
<td>2.2</td>
<td>Model</td>
<td></td>
<td>Q354/564.64-6.9/4</td>
<td></td>
</tr>
<tr>
<td>2.3</td>
<td>Quantity</td>
<td>set</td>
<td>4</td>
<td></td>
</tr>
<tr>
<td>2.4</td>
<td>100% load continuous rate</td>
<td>t/h</td>
<td>62.56</td>
<td>63.78</td>
</tr>
<tr>
<td>2.5</td>
<td>Main steam temperature</td>
<td>°C</td>
<td>468</td>
<td>468</td>
</tr>
<tr>
<td>2.6</td>
<td>Main steam pressure</td>
<td>MPa</td>
<td>7.0</td>
<td>7.0</td>
</tr>
<tr>
<td>2.7</td>
<td>Feed water temperature</td>
<td>°C</td>
<td>104</td>
<td>104</td>
</tr>
<tr>
<td>2.8</td>
<td>Exhaust flow gas Temperature</td>
<td>°C</td>
<td>153.9</td>
<td>153.9</td>
</tr>
<tr>
<td>NO.</td>
<td>Item</td>
<td>Unit</td>
<td>Unit 1&amp; Unit2</td>
<td>Remarks</td>
</tr>
<tr>
<td>-----</td>
<td>-----------------------------------</td>
<td>-----------------------------------</td>
<td>--------------------------------</td>
<td>--------------------------------</td>
</tr>
<tr>
<td>3</td>
<td>Steam Turbine</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>3.1</td>
<td>Type</td>
<td>Directly condensing with</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>extraction for heating LPG</td>
<td></td>
<td></td>
</tr>
<tr>
<td>3.2</td>
<td>Model</td>
<td>1.35-6.7-1</td>
<td></td>
<td></td>
</tr>
<tr>
<td>3.3</td>
<td>Quantity</td>
<td>Set</td>
<td>2</td>
<td></td>
</tr>
<tr>
<td>3.4</td>
<td>Main steam flow-rate at inlet of</td>
<td>°C</td>
<td>2x62.56</td>
<td>2x63.78</td>
</tr>
<tr>
<td></td>
<td>main steam stop valve</td>
<td></td>
<td></td>
<td>Am.temp. °C</td>
</tr>
<tr>
<td>3.5</td>
<td>Pressure at inlet of main stop</td>
<td>MPa</td>
<td>6.7</td>
<td>6.7</td>
</tr>
<tr>
<td></td>
<td>valve</td>
<td></td>
<td></td>
<td>Am.temp. °C</td>
</tr>
<tr>
<td>3.6</td>
<td>Temperature at inlet of main</td>
<td>°C</td>
<td>465</td>
<td>465</td>
</tr>
<tr>
<td></td>
<td>stop valve</td>
<td></td>
<td></td>
<td>Am.temp. °C</td>
</tr>
<tr>
<td>3.7</td>
<td>Extraction pressure</td>
<td>MPa</td>
<td></td>
<td>0.922</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td>Am.temp. °C</td>
</tr>
<tr>
<td>3.8</td>
<td>Extraction temperature</td>
<td>°C</td>
<td></td>
<td>242.9</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td>Am.temp. °C</td>
</tr>
<tr>
<td>3.9</td>
<td>Extraction flow</td>
<td>t/h</td>
<td>6</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td>Am.temp. °C</td>
</tr>
<tr>
<td>4</td>
<td>Back pressure</td>
<td>kPa</td>
<td>10.2</td>
<td>10.2</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td>Am.temp. °C</td>
</tr>
<tr>
<td>4.1</td>
<td>Temperature of cooling</td>
<td>°C</td>
<td>32</td>
<td>32</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td>Am.temp. °C</td>
</tr>
<tr>
<td>4.2</td>
<td>Speed</td>
<td>r/min</td>
<td>3000</td>
<td>3000</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td>Am.temp. °C</td>
</tr>
<tr>
<td>4.3</td>
<td>Output</td>
<td>kW</td>
<td>32720</td>
<td>32450</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td>Am.temp. °C</td>
</tr>
</tbody>
</table>
Table 4-5: MAIN PERFORMANCE DATA (100% LOAD CONDITION)

<table>
<thead>
<tr>
<th>NO</th>
<th>Item</th>
<th>Unit</th>
<th>Unit 1 &amp; Unit 2</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Configuration mode</td>
<td></td>
<td>2 + 2 + 1</td>
</tr>
<tr>
<td>2</td>
<td>Fuel</td>
<td>Diesel oil</td>
<td>LPG</td>
</tr>
<tr>
<td>3</td>
<td>Unit generating power</td>
<td>kW</td>
<td>95924</td>
</tr>
<tr>
<td></td>
<td>Gas turbine output</td>
<td>kW</td>
<td>31602</td>
</tr>
<tr>
<td>3.1</td>
<td>Steam turbine output</td>
<td>kW</td>
<td>32720</td>
</tr>
<tr>
<td></td>
<td>Boiler 100% load continuous rate</td>
<td>t/h</td>
<td>62.56</td>
</tr>
<tr>
<td>3.4</td>
<td>Gas heat consumption</td>
<td>kJ/h</td>
<td>390×10^6</td>
</tr>
<tr>
<td>3.5</td>
<td>Fuel low heat value</td>
<td>kJ/kg</td>
<td>42697.2</td>
</tr>
<tr>
<td>3.6</td>
<td>Gas turbine consumption heat rate</td>
<td>kJ/Kw.h</td>
<td>12340.99</td>
</tr>
<tr>
<td>3.7</td>
<td>Gas turbine fuel consumption rate</td>
<td>g/Kw.h</td>
<td>289</td>
</tr>
<tr>
<td>3.8</td>
<td>Gas turbine efficiency</td>
<td>%</td>
<td>29.17</td>
</tr>
<tr>
<td>3.9</td>
<td>Gas turbine fuel Consumption</td>
<td>t/h</td>
<td>9.134</td>
</tr>
<tr>
<td>3.10</td>
<td>Plant heat consumption rate</td>
<td>kJ/Kw.h</td>
<td>8131.44</td>
</tr>
<tr>
<td>3.11</td>
<td>Plant fuel consumption rate</td>
<td>g/Kw.h</td>
<td>190.4</td>
</tr>
<tr>
<td>3.12</td>
<td>Plant efficiency</td>
<td>%</td>
<td>44.27</td>
</tr>
</tbody>
</table>
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

\[
\psi = (h - h_0) - T_0(s - s_a) + \left(\frac{V^2}{2}\right) + gz
\]

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

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

Or

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

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

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

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

$$h = h_f + x h_{fg}$$
From the appendix (A) all the values of the dryness fraction at inlet to condenser is limited to 0.88 so that in partial load calculations the average value is taken as approximation as

\[ x = \frac{x_1 + x_2 + x_3 + \ldots + x_n}{n} \]

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

\[ \frac{A}{F} = \frac{m_a}{m_f} \]

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

\[ h = h_u + wh_g \]

\[ h_u = C_p T \]

\[ h_g = 2501.3 + 1.82T \]

\[ w = 0.0622P_v/(P - P_v) \]

\[ P_v = \Phi P_g \text{ at } T \]

NOTE: values of \( P_g \) corresponding to \( T \) are listed in steam table.

Fuel energy is expressed as

\[ E_f = m_f \times h_f \text{ (L.H.V)} \]

The power output 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

\[ h = C_p g \times T_{in} \]
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

\[ h_o = C p_g \times T_{out} \]

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 outlet, 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 $H_i$ expressed as

$$H_i,HP = 2 \times m_{i,HP} \times h_o$$

Also LP steam

$$H_i,LP = 2 \times m_{i,LP} \times h_o$$

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

The values of steam power output 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).

<table>
<thead>
<tr>
<th>EQUIPMENT</th>
<th>position</th>
<th>m(kg/s)</th>
<th>T(°C)</th>
<th>P(MPa)</th>
<th>h(kJ/kg)</th>
<th>H(kJ/kg)</th>
<th>G(pkj/s)</th>
<th>S(kJ/kgK)</th>
<th>Y(kJ/kg)</th>
<th>ψ(kJ/s)</th>
<th>θ(kJ/s)</th>
</tr>
</thead>
<tbody>
<tr>
<td>HRSG: SUPER HEATER</td>
<td>i</td>
<td>34.167</td>
<td>388.4</td>
<td>7.084</td>
<td>2772</td>
<td>94710.924</td>
<td>19136.94</td>
<td>6.8108</td>
<td>953.216</td>
<td>953.216</td>
<td>42364.3</td>
</tr>
<tr>
<td></td>
<td>o</td>
<td>34.167</td>
<td>469.7</td>
<td>7</td>
<td>3322.1</td>
<td>113947.6607</td>
<td>6.6992</td>
<td>1238.30</td>
<td>43311.745</td>
<td></td>
<td></td>
</tr>
<tr>
<td>EVAPURATOR i</td>
<td>34.512</td>
<td>388.4</td>
<td>7.084</td>
<td>1270.3</td>
<td>43840.6593</td>
<td>59360.33</td>
<td>3.1263</td>
<td>201.6116</td>
<td>10054.1</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>o</td>
<td>34.167</td>
<td>388.4</td>
<td>7.084</td>
<td>2772</td>
<td>94710.924</td>
<td>5.8108</td>
<td>853.216</td>
<td>32588.654</td>
<td></td>
<td></td>
</tr>
<tr>
<td>ECONOMIZOR i</td>
<td>34.512</td>
<td>104.2</td>
<td>8.2</td>
<td>441.57</td>
<td>15242.9150</td>
<td>27234.19</td>
<td>1.3349</td>
<td>23.8436</td>
<td>822.9531</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>o</td>
<td>34.167</td>
<td>279.5</td>
<td>7.084</td>
<td>1293.4</td>
<td>42567.1008</td>
<td>3.8865</td>
<td>275.4522</td>
<td>3508.7963</td>
<td></td>
<td></td>
</tr>
<tr>
<td>L.P. EVAPURATOR i</td>
<td>3.951</td>
<td>142.9</td>
<td>0.39</td>
<td>600.58</td>
<td>2136.6663</td>
<td>7600.67</td>
<td>1.776</td>
<td>44.652</td>
<td>153.1421</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>o</td>
<td>3.891</td>
<td>142.9</td>
<td>0.39</td>
<td>2735</td>
<td>9798.31</td>
<td>6.897</td>
<td>575.239</td>
<td>2051.9871</td>
<td></td>
<td></td>
</tr>
<tr>
<td>ST: GOVERNOR VALVE</td>
<td>i</td>
<td>34.167</td>
<td>465.7</td>
<td>6.7</td>
<td>3339</td>
<td>113741.94</td>
<td>35804.42</td>
<td>6.7041</td>
<td>12010.617</td>
<td>40246.101</td>
<td>46846.54</td>
</tr>
<tr>
<td></td>
<td>o</td>
<td>33.22</td>
<td>466.8</td>
<td>0.0115</td>
<td>2315</td>
<td>79967.52</td>
<td>0</td>
<td>7.061</td>
<td>-144.483</td>
<td>-4000.1575</td>
<td>4425.03</td>
</tr>
<tr>
<td>CONDENSER: CONDENSATE STEAM i</td>
<td>33.22</td>
<td>466.8</td>
<td>0.0115</td>
<td>2315</td>
<td>79967.52</td>
<td>70193.68</td>
<td>7.061</td>
<td>-144.483</td>
<td>-4000.1575</td>
<td>4425.03</td>
<td></td>
</tr>
<tr>
<td></td>
<td>o</td>
<td>33.22</td>
<td>466.8</td>
<td>0.0115</td>
<td>2315</td>
<td>79967.52</td>
<td>0</td>
<td>7.061</td>
<td>-144.483</td>
<td>-4000.1575</td>
<td>4425.03</td>
</tr>
<tr>
<td>MAKE UP WATER i</td>
<td>0.947</td>
<td>465.8</td>
<td>0.0115</td>
<td>203.3</td>
<td>1922551</td>
<td>0</td>
<td>0.663</td>
<td>-11.1989</td>
<td>10695598</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>o</td>
<td>0.947</td>
<td>466.8</td>
<td>0.0115</td>
<td>203.3</td>
<td>6946.151</td>
<td>0</td>
<td>0.663</td>
<td>-11.1989</td>
<td>10695598</td>
<td></td>
</tr>
<tr>
<td>HOT WELL i</td>
<td>34.167</td>
<td>465.8</td>
<td>0.0115</td>
<td>203.3</td>
<td>6946.151</td>
<td>275.637</td>
<td>0.7161</td>
<td>-9.6039</td>
<td>236.1736</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>o</td>
<td>34.167</td>
<td>51.1</td>
<td>1.1</td>
<td>214.3</td>
<td>7231.9881</td>
<td>0</td>
<td>7.161</td>
<td>-9.0939</td>
<td>236.1736</td>
<td></td>
</tr>
<tr>
<td>FEEDING SYSTEM: C.P i</td>
<td>34.167</td>
<td>465.8</td>
<td>0.0115</td>
<td>203.3</td>
<td>6946.151</td>
<td>375.637</td>
<td>0.663</td>
<td>-11.1989</td>
<td>382.63282</td>
<td>46.4536</td>
<td></td>
</tr>
<tr>
<td></td>
<td>o</td>
<td>34.167</td>
<td>51.1</td>
<td>1.1</td>
<td>214.3</td>
<td>7231.9881</td>
<td>0</td>
<td>7.161</td>
<td>-9.0939</td>
<td>236.1736</td>
<td></td>
</tr>
<tr>
<td>H.P. F.W.P i</td>
<td>34.167</td>
<td>104.2</td>
<td>0.12</td>
<td>435.76</td>
<td>14988.6112</td>
<td>201.927</td>
<td>1.3515</td>
<td>12.7456</td>
<td>453.30466</td>
<td>372.503</td>
<td></td>
</tr>
<tr>
<td></td>
<td>o</td>
<td>34.167</td>
<td>104.2</td>
<td>0.12</td>
<td>441.56</td>
<td>15239.5309</td>
<td>1.3492</td>
<td>20.7163</td>
<td>707.81362</td>
<td></td>
<td></td>
</tr>
<tr>
<td>L.P. F.W.P i</td>
<td>3.951</td>
<td>104.2</td>
<td>0.12</td>
<td>435.76</td>
<td>15217.4136</td>
<td>2.52931</td>
<td>1.3515</td>
<td>12.7456</td>
<td>453.30466</td>
<td>3.41968</td>
<td></td>
</tr>
<tr>
<td></td>
<td>o</td>
<td>3.951</td>
<td>104.2</td>
<td>0.12</td>
<td>456.47</td>
<td>15642.2666</td>
<td>1.3507</td>
<td>13.7008</td>
<td>48.79885</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

85
Table 5-2: EXERGY ANALYSIS FOR STEAM CYCLE (GAS TURBINE FUEL LPG FOR 100% LOAD).

<table>
<thead>
<tr>
<th>EQUIPMENT</th>
<th>M(kg/s)</th>
<th>T(C)</th>
<th>P(MPa)</th>
<th>h(10^3(kgF)/m^2)</th>
<th>h(10^3(kgF)/m^2)</th>
<th>Q(kW)</th>
<th>S(kW)</th>
<th>S1(kW)</th>
<th>S2(kW)</th>
<th>S3(kW)</th>
<th>S4(kW)</th>
</tr>
</thead>
<tbody>
<tr>
<td>HiGAS</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>SUPER HEATER</td>
<td>i</td>
<td>35.433</td>
<td>289.6</td>
<td>7.2938</td>
<td>2789</td>
<td>9611.98</td>
<td>19692.32</td>
<td>5.7</td>
<td>5.7</td>
<td>5.7</td>
<td>5.7</td>
</tr>
<tr>
<td></td>
<td>o</td>
<td>35.433</td>
<td>468</td>
<td>7</td>
<td>3032.1</td>
<td>118066.3</td>
<td>6.6982</td>
<td>1238.39</td>
<td>4697.53</td>
<td></td>
<td></td>
</tr>
<tr>
<td>EVAPURATOR</td>
<td>i</td>
<td>35.433</td>
<td>289.6</td>
<td>7.2938</td>
<td>2789</td>
<td>9611.98</td>
<td>5.7945</td>
<td>955.82</td>
<td>3368.91</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>o</td>
<td>35.433</td>
<td>289.6</td>
<td>7.2938</td>
<td>2789</td>
<td>9611.98</td>
<td>5.7945</td>
<td>955.82</td>
<td>3368.91</td>
<td></td>
<td></td>
</tr>
<tr>
<td>ECONOMIZER</td>
<td>i</td>
<td>35.791</td>
<td>104</td>
<td>8.2</td>
<td>441.67</td>
<td>6007.61</td>
<td>2972.25</td>
<td>3.1</td>
<td>3.1</td>
<td>3.1</td>
<td>3.1</td>
</tr>
<tr>
<td></td>
<td>o</td>
<td>35.791</td>
<td>281.6</td>
<td>7.2938</td>
<td>2789</td>
<td>9611.98</td>
<td>5.7945</td>
<td>955.82</td>
<td>3368.91</td>
<td></td>
<td></td>
</tr>
<tr>
<td>LP EVAPURATOR</td>
<td>i</td>
<td>3.834</td>
<td>144.3</td>
<td>0.4075</td>
<td>607.3</td>
<td>2343.625</td>
<td>1788.33</td>
<td>1.7</td>
<td>1.7</td>
<td>1.7</td>
<td>1.7</td>
</tr>
<tr>
<td></td>
<td>o</td>
<td>3.834</td>
<td>144.3</td>
<td>0.4075</td>
<td>607.3</td>
<td>2343.625</td>
<td>1788.33</td>
<td>1.7</td>
<td>1.7</td>
<td>1.7</td>
<td>1.7</td>
</tr>
<tr>
<td>ST: GOVERNOR VALVE</td>
<td>i</td>
<td>35.433</td>
<td>466</td>
<td>6.7</td>
<td>3339</td>
<td>117955.5</td>
<td>88869.24</td>
<td>6.7</td>
<td>6.7</td>
<td>6.7</td>
<td>6.7</td>
</tr>
<tr>
<td></td>
<td>o</td>
<td>34.444</td>
<td>45.6</td>
<td>0.0039</td>
<td>2096.11</td>
<td>79967.21</td>
<td>8.1539</td>
<td>255.061</td>
<td>619.175</td>
<td></td>
<td></td>
</tr>
<tr>
<td>CONDENSER: CONDENSATE STEAM</td>
<td>i</td>
<td>34.444</td>
<td>45.6</td>
<td>0.0039</td>
<td>2096.11</td>
<td>79967.21</td>
<td>7.2965.27</td>
<td>8.1539</td>
<td>255.061</td>
<td>619.175</td>
<td>8-436.39</td>
</tr>
<tr>
<td>MAKE UP WATER</td>
<td>i</td>
<td>1.867</td>
<td>20</td>
<td>0.0038</td>
<td>63.79</td>
<td>136.8778</td>
<td>0.0038</td>
<td>-8.5984</td>
<td>-17.4141</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>o</td>
<td>35.433</td>
<td>45.6</td>
<td>0.0039</td>
<td>190.8</td>
<td>6760.616</td>
<td>0.0038</td>
<td>-11.4606</td>
<td>-400.063</td>
<td>45.9556</td>
<td></td>
</tr>
<tr>
<td>HOT WELL</td>
<td>i</td>
<td>35.433</td>
<td>45.6</td>
<td>0.0039</td>
<td>190.8</td>
<td>6760.616</td>
<td>0.0038</td>
<td>-11.4606</td>
<td>-400.063</td>
<td>45.9556</td>
<td></td>
</tr>
<tr>
<td></td>
<td>o</td>
<td>35.433</td>
<td>45.6</td>
<td>0.0039</td>
<td>190.8</td>
<td>6760.616</td>
<td>0.0038</td>
<td>-11.4606</td>
<td>-400.063</td>
<td>45.9556</td>
<td></td>
</tr>
<tr>
<td>FEEDING SYSTEM: C.P</td>
<td>i</td>
<td>35.433</td>
<td>45.6</td>
<td>0.0039</td>
<td>190.8</td>
<td>6760.616</td>
<td>0.0038</td>
<td>-11.4606</td>
<td>-400.063</td>
<td>45.9556</td>
<td></td>
</tr>
<tr>
<td></td>
<td>o</td>
<td>35.433</td>
<td>45.6</td>
<td>0.0039</td>
<td>190.8</td>
<td>6760.616</td>
<td>0.0038</td>
<td>-11.4606</td>
<td>-400.063</td>
<td>45.9556</td>
<td></td>
</tr>
<tr>
<td>H.P.F.W.P</td>
<td>i</td>
<td>35.433</td>
<td>104</td>
<td>0.12</td>
<td>455.76</td>
<td>16440.28</td>
<td>203.409</td>
<td>1.3515</td>
<td>12.4705</td>
<td>451.4034</td>
<td>-282.407</td>
</tr>
<tr>
<td></td>
<td>o</td>
<td>35.433</td>
<td>104</td>
<td>8.2</td>
<td>441.67</td>
<td>6649.68</td>
<td>1.3449</td>
<td>20.7163</td>
<td>734.0407</td>
<td></td>
<td></td>
</tr>
<tr>
<td>L.P.F.W.P</td>
<td>i</td>
<td>3.834</td>
<td>104</td>
<td>0.12</td>
<td>455.76</td>
<td>16440.28</td>
<td>2.62274</td>
<td>1.3515</td>
<td>12.4705</td>
<td>47.6304</td>
<td>-3.54772</td>
</tr>
<tr>
<td></td>
<td>o</td>
<td>3.834</td>
<td>104</td>
<td>1.1</td>
<td>455.47</td>
<td>1671.32</td>
<td>1.3937</td>
<td>13.7309</td>
<td>50.61112</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

86
Table 5-3: EXERGY ANALYSIS FOR STEAM CYCLE (GAS TURBINE FUEL LIGHT DESIL OIL FOR 66% LOAD).

<table>
<thead>
<tr>
<th>EQUIPMENT</th>
<th>position</th>
<th>M(kg/s)</th>
<th>T(C)</th>
<th>P(MPa)</th>
<th>h(kJ/kg)</th>
<th>h1(kJ/kg)</th>
<th>h2(kJ/kg)</th>
<th>Gex(KJ)</th>
<th>R(kg/kJ)</th>
<th>m(kg/s)</th>
<th>Tout(K)</th>
</tr>
</thead>
<tbody>
<tr>
<td>HRSG: SUPER HEATER</td>
<td>i</td>
<td>27.222</td>
<td>367.22</td>
<td>5.27</td>
<td>2791.98</td>
<td>15198.04</td>
<td>5.8468</td>
<td>939.9082</td>
<td>1203.6144</td>
<td>30861.92</td>
<td>-33534</td>
</tr>
<tr>
<td></td>
<td>o</td>
<td>27.222</td>
<td>456</td>
<td>5.2</td>
<td>3049.32</td>
<td>91174.64</td>
<td>6.045</td>
<td>1203.6144</td>
<td>30861.92</td>
<td>-33534</td>
<td></td>
</tr>
<tr>
<td>EVAPURATOR</td>
<td>i</td>
<td>27.497</td>
<td>367.22</td>
<td>5.27</td>
<td>1170.83</td>
<td>32193.49</td>
<td>43763.11</td>
<td>2.9905</td>
<td>247.2356</td>
<td>6794.69</td>
<td>20941.64</td>
</tr>
<tr>
<td></td>
<td>o</td>
<td>27.497</td>
<td>263</td>
<td>5.27</td>
<td>7976.6</td>
<td>7596.6</td>
<td>5.9468</td>
<td>9256.982</td>
<td>2591.64</td>
<td>-33534</td>
<td></td>
</tr>
<tr>
<td>ECONOMIZER</td>
<td>i</td>
<td>27.497</td>
<td>104</td>
<td>8.2</td>
<td>441.67</td>
<td>12144.6</td>
<td>19462.45</td>
<td>1.3349</td>
<td>23.0643</td>
<td>655.7017</td>
<td></td>
</tr>
<tr>
<td></td>
<td>o</td>
<td>27.497</td>
<td>263</td>
<td>5.27</td>
<td>11494</td>
<td>31605.05</td>
<td>2.9109</td>
<td>236.2863</td>
<td>6552.213</td>
<td>-33534</td>
<td></td>
</tr>
<tr>
<td>L.P. EVAPURATOR</td>
<td>i</td>
<td>3.333</td>
<td>113.32</td>
<td>0.16</td>
<td>475.2</td>
<td>1683.94</td>
<td>2399.26</td>
<td>1.4547</td>
<td>19.6799</td>
<td>66.2563</td>
<td></td>
</tr>
<tr>
<td></td>
<td>o</td>
<td>3.333</td>
<td>113.32</td>
<td>0.16</td>
<td>2606.2</td>
<td>883.102</td>
<td>7.2002</td>
<td>441.5374</td>
<td>1471.644</td>
<td>-33534</td>
<td></td>
</tr>
<tr>
<td>ST: GOVERNOR VALVE</td>
<td>i</td>
<td>27.222</td>
<td>462</td>
<td>5.1</td>
<td>5343.6</td>
<td>91019.48</td>
<td>31148.99</td>
<td>6.6959</td>
<td>1203.83</td>
<td>30899.08</td>
<td>33091.91</td>
</tr>
<tr>
<td></td>
<td>o</td>
<td>25.222</td>
<td>39.4</td>
<td>0.0035</td>
<td>2263.216</td>
<td>56920.49</td>
<td>0</td>
<td>7.3437</td>
<td>-15.3621</td>
<td>-402.815</td>
<td></td>
</tr>
<tr>
<td>CONDENSER: CONDENSATE STEAM</td>
<td>i</td>
<td>26.222</td>
<td>39.4</td>
<td>0.0035</td>
<td>2263.216</td>
<td>56920.49</td>
<td>56485.38</td>
<td>7.3437</td>
<td>-15.3621</td>
<td>-402.815</td>
<td></td>
</tr>
<tr>
<td></td>
<td>o</td>
<td>26.222</td>
<td>39.4</td>
<td>0.0035</td>
<td>1649.8</td>
<td>4489.906</td>
<td>0</td>
<td>0.6564</td>
<td>-11.9633</td>
<td>-317.498</td>
<td></td>
</tr>
<tr>
<td>MAKE UP WATER</td>
<td>i</td>
<td>1</td>
<td>20</td>
<td>0.0025</td>
<td>63.796</td>
<td>63.796</td>
<td>0</td>
<td>0.0569</td>
<td>-8.9207</td>
<td>-8.9207</td>
<td></td>
</tr>
<tr>
<td>HOT WELL</td>
<td>o</td>
<td>27.222</td>
<td>39.4</td>
<td>0.0035</td>
<td>164.9</td>
<td>4489.906</td>
<td>0</td>
<td>0.6564</td>
<td>-11.9633</td>
<td>-317.498</td>
<td></td>
</tr>
<tr>
<td>FEEDING SYSTEM: C.P</td>
<td>i</td>
<td>27.222</td>
<td>39.4</td>
<td>0.0035</td>
<td>164.9</td>
<td>4489.906</td>
<td>239.7142</td>
<td>0.6341</td>
<td>-11.9633</td>
<td>-317.498</td>
<td></td>
</tr>
<tr>
<td></td>
<td>o</td>
<td>27.222</td>
<td>41.8</td>
<td>1.1</td>
<td>175.91</td>
<td>4786.622</td>
<td>0</td>
<td>0.6564</td>
<td>-10.5441</td>
<td>-387.031</td>
<td></td>
</tr>
<tr>
<td>H.P. F.W.P</td>
<td>i</td>
<td>27.222</td>
<td>104</td>
<td>0.12</td>
<td>435.76</td>
<td>11862.25</td>
<td>163.982</td>
<td>1.3515</td>
<td>12.7405</td>
<td>346.8219</td>
<td>317.117</td>
</tr>
<tr>
<td></td>
<td>o</td>
<td>27.222</td>
<td>104</td>
<td>8.2</td>
<td>441.67</td>
<td>12023.14</td>
<td>1.3449</td>
<td>20.7163</td>
<td>663.9394</td>
<td>-33534</td>
<td></td>
</tr>
<tr>
<td>L.P. F.W.P</td>
<td>i</td>
<td>3.333</td>
<td>104</td>
<td>0.12</td>
<td>435.76</td>
<td>1452.388</td>
<td>23084.43</td>
<td>1.3515</td>
<td>12.7405</td>
<td>42.4560</td>
<td>3.92101</td>
</tr>
<tr>
<td></td>
<td>o</td>
<td>3.333</td>
<td>9.1</td>
<td>1.1</td>
<td>145.47</td>
<td>1454.755</td>
<td>1.3607</td>
<td>13.7009</td>
<td>45.6661</td>
<td>-33534</td>
<td></td>
</tr>
</tbody>
</table>

313

87
Table 5-4: EXERGY ANALYSIS OF MAIN EQUIPMENT FOR GARRI POWER PLANT.

### GAS TURBINE (ONE SET)

<table>
<thead>
<tr>
<th>Load (%)</th>
<th>m(fuel) (kg/s)</th>
<th>h(t(fuel)) (kJ/kg)</th>
<th>E_t (kJ/s)</th>
<th>W(t) (kW)</th>
<th>η (%)</th>
<th>m(Exhaust gas) (kg/s)</th>
<th>h(Exhaust gas) (kJ/kg)</th>
<th>E_g (kJ/s)</th>
<th>W(g) (kW)</th>
<th>C_p (kJ/kg)</th>
<th>η(1ST) (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>100</td>
<td>2.5572</td>
<td>123.0118</td>
<td>4369.72</td>
<td>88.0398</td>
<td>103331.3</td>
<td>31602</td>
<td>23.17161</td>
<td>125.5966</td>
<td>648.6</td>
<td>81435.62</td>
<td>53180.44</td>
</tr>
<tr>
<td>66</td>
<td>1.8717</td>
<td>94.101</td>
<td>4269.72</td>
<td>88.0398</td>
<td>79916.35</td>
<td>20000</td>
<td>25.03617</td>
<td>95.9727</td>
<td>648.6</td>
<td>62247.09</td>
<td>43275.38</td>
</tr>
</tbody>
</table>

### HRSG (ONE SET)

<table>
<thead>
<tr>
<th>LOAD (%)</th>
<th>m(Exhaust gas) (kg/s)</th>
<th>h(Inlet) (kJ/kg)</th>
<th>h(LHP) (kJ/kg)</th>
<th>m(Cond) (kg/s)</th>
<th>h(LHP) (kJ/kg)</th>
<th>m(Sat) (kg/s)</th>
<th>h(Sat) (kJ/kg)</th>
<th>W(ST) (kW)</th>
<th>η(HRSG) (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>100</td>
<td>125.566</td>
<td>648.6</td>
<td>161.815</td>
<td>58007.66</td>
<td>17.084</td>
<td>441.67</td>
<td>33321.1</td>
<td>48380.11</td>
<td>17805</td>
</tr>
<tr>
<td>100</td>
<td>125</td>
<td>648.6</td>
<td>162.275</td>
<td>60290.63</td>
<td>17.717</td>
<td>441.67</td>
<td>33321.1</td>
<td>51203.75</td>
<td>1847</td>
</tr>
<tr>
<td>66</td>
<td>95.9727</td>
<td>648.6</td>
<td>161.815</td>
<td>44798.62</td>
<td>13.611</td>
<td>441.67</td>
<td>3349.3</td>
<td>39575.75</td>
<td>16665</td>
</tr>
</tbody>
</table>

### STEAM TURBINE SET (COMBINED by 2 HRSGs)

<table>
<thead>
<tr>
<th>LOAD (%)</th>
<th>Hi (kJ/s)</th>
<th>Hi steam</th>
<th>Lp steam</th>
<th>ΔHi (kJ/kg)</th>
<th>W(ST) (kW)</th>
<th>η Rankine (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>100</td>
<td>113351.2</td>
<td>9739.335</td>
<td>123590.5</td>
<td>32720</td>
<td>26.47452</td>
<td></td>
</tr>
<tr>
<td>100</td>
<td>116318.6</td>
<td>10111.96</td>
<td>128161.6</td>
<td>32450</td>
<td>25.31566</td>
<td></td>
</tr>
<tr>
<td>66</td>
<td>91174.64</td>
<td>8683.102</td>
<td>100157.7</td>
<td>25100</td>
<td>25.08047</td>
<td></td>
</tr>
</tbody>
</table>
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

$$
\eta_i = \frac{(W_E + Q_P) \times 100}{E_f}
$$

(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 $m_{SH}(h - h_c)_H$ plus low-pressure steam $m_{SL}(h - h_c)_L$. 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 $\sum j\Delta S_j \geq 0$ 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 $\sum \Delta \Psi$ 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),
\[
R = \frac{W_{\text{GT}}}{Q_p} \quad \text{(6-2)}
\]

and for combined cycle,
\[
R = \frac{W_E}{Q'_p} \quad \text{(6-3)}
\]

Where \( Q'_p = Q_p - W_{sr} \). 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

\[
\eta_E = \frac{(W_E + \Delta \psi) \times 100}{\psi_f} \quad \text{(6-4)}
\]
Fig. (6-1). T–S diagram of combined cycle plant.

The exergy factor of generated steam/water ($\varepsilon_s$) and the exergy factor of fuel input ($\psi_f$) can be expressed as follows:

$$\varepsilon_s = \frac{\Delta \psi}{Q_p}$$ (6-5)

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

Substituting in equation (6-4)
\[
\eta_{II} = \frac{W_E + \varepsilon_s Q_p + Q_p - Q_{p}}{\varepsilon_f E_f} = \frac{W_E + Q_p + (\varepsilon_s - 1)Q_p}{\varepsilon_f E_f} \\
\overset{\text{(6-7)}}{= \eta_I \left[ 1 + \frac{\varepsilon_f}{\eta_I} \frac{(\varepsilon_s - 1)Q_p}{\varepsilon_f E_f} \right] = \eta_I \left[ \frac{W_E + \Delta \psi}{\eta_I E_f} \right]} \\
\overset{\text{(6-7)}}{= \eta_I \left[ \frac{R + \varepsilon_s}{R + 1} \right]}
\]

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 \((\eta_{GT})\) 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 (\(\eta\)) 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 (\(\eta_{cc}\)) is 44.27% \([95924 \text{ kW}/(216662.6 \text{ kJ/s})]\) 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 (\(\eta_{ul}\)) at 100% load is 2.44% higher than at 66% load.
TABLE (6-1): EXERGY DESTRUCTION FOR GARRI 1 COMPONENTS.

<table>
<thead>
<tr>
<th>Equipment</th>
<th>unit</th>
<th>EXERGY DESTRUCTION</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>100% LDO</td>
</tr>
<tr>
<td>HRSG</td>
<td>kJ/s</td>
<td>22601.42</td>
</tr>
<tr>
<td>S.T</td>
<td>kJ/s</td>
<td>10042.14</td>
</tr>
<tr>
<td>CONDENSE</td>
<td>kJ/s</td>
<td>4428.03</td>
</tr>
<tr>
<td>C.P</td>
<td>kJ/s</td>
<td>329.384</td>
</tr>
<tr>
<td>H.P.F.W.P</td>
<td>kJ/s</td>
<td>70.582</td>
</tr>
<tr>
<td>L.P.F.W.P</td>
<td>kJ/s</td>
<td>0.89167</td>
</tr>
<tr>
<td>GT</td>
<td>kJ/s</td>
<td>67578.86</td>
</tr>
</tbody>
</table>

6.3. THERMODYNAMICS ANALYSIS

The first law efficiency of simple cycle $\eta_1 = 78.26\%$ 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 $\eta_2 = 48.93\%$, where in the heat to power ratio ($R$) is equal to 0.5942 which is calculated by Eq. (6-2) and $\varepsilon_s = 0.4026$ 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 $Q_p$ to obtain $Q'_p$, which is equal to 73640.88 kJ/s. The second law efficiency of combined cycle $\eta_2 = 64.04\%$, wherein R is equal to $W_{\varepsilon}/Q'_p = 1.3026$ 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 $C_p$ (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
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.

Figure 6-5: Gas and steam temperature profile of HRSG for 66% load LDO
The pinch point for the HP portion is \( \Delta TH = 8.6^\circ C \) (295-286.4 \(^\circ\)C). This pinch point is an important parameter affecting the thermal performance of the system.

If the pinch point of an HRSG is changed to 10, 20 or 30 \(^\circ\)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.
References


[14] Deng-Chern Sue- Chia-Chin Chuang, Engineering design and exergy analysis for combustion gas turbine based power generation system.


# List of contents

<table>
<thead>
<tr>
<th>Subject</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>ACKNOLEDGEMENT</td>
<td>II</td>
</tr>
<tr>
<td>ABSTRACT</td>
<td>III</td>
</tr>
<tr>
<td>LIST OF CONTENTS</td>
<td>V</td>
</tr>
<tr>
<td>LIST OF FIGURE</td>
<td>VIII</td>
</tr>
<tr>
<td>LIST OF TABLES</td>
<td>IX</td>
</tr>
<tr>
<td>ABBREVIATIONS</td>
<td>X</td>
</tr>
<tr>
<td>Nomenclature</td>
<td>XI</td>
</tr>
<tr>
<td>CHAPTER ONE</td>
<td>1</td>
</tr>
<tr>
<td>1.1. ELECTRICITY SUPPLY IN SUDAN</td>
<td>2</td>
</tr>
<tr>
<td>1.2. ELECTRICAL SYSTEM IN SUDAN</td>
<td>4</td>
</tr>
<tr>
<td>1.2.1. NEC Short (2005-2010) and Long (2005-2030) Term Plans.</td>
<td>6</td>
</tr>
<tr>
<td>1.3. SECOND-LAW OF THERMODYNAMIC ASPECTS OF DAILY LIFE</td>
<td>9</td>
</tr>
<tr>
<td>1.4. OBJECTIVES OF STUDY</td>
<td>12</td>
</tr>
<tr>
<td>CHAPTER TWO</td>
<td>13</td>
</tr>
<tr>
<td>LITERATURE REVIEW</td>
<td>13</td>
</tr>
<tr>
<td>CHAPTER THREE</td>
<td>27</td>
</tr>
<tr>
<td>FUNDAMENTALS OF EXERGY ANALYSIS</td>
<td>27</td>
</tr>
<tr>
<td>3.1 EXERGY: A MEASURE OF WORK POTENTIAL</td>
<td>28</td>
</tr>
<tr>
<td>3.2 EXERGY WORK POTENTIAL OF ENERGY</td>
<td>28</td>
</tr>
<tr>
<td>3.3 REVERSIBLE WORK AND IRREVERSIBILITY</td>
<td>31</td>
</tr>
<tr>
<td>3.4 SECOND-LAW EFFICIENCY, $\eta_{II}$</td>
<td>33</td>
</tr>
<tr>
<td>3.5 EXERGY CHANGE OF A SYSTEM</td>
<td>37</td>
</tr>
<tr>
<td>3.5.1.1 Exergy of a Fixed Mass:</td>
<td>38</td>
</tr>
<tr>
<td>3.5.1.2 Exergy of a Flow Stream: Flow (or Stream Exergy</td>
<td>42</td>
</tr>
<tr>
<td>3.6 THE DECREASE OF EXERGY PRINCIPLE AND EXERGY DESTRUCTION</td>
<td>44</td>
</tr>
<tr>
<td>3.6.1 Exergy Destruction</td>
<td>46</td>
</tr>
<tr>
<td>3.7 EXERGY BALANCE: CLOSED SYSTEMS</td>
<td>48</td>
</tr>
<tr>
<td>3.8 EXERGY BALANCE: CONTROL VOLUMES</td>
<td>52</td>
</tr>
<tr>
<td>3.8.1 Exergy Balance for Steady-Flow System</td>
<td>53</td>
</tr>
<tr>
<td>3.8.2 Reversible Work</td>
<td>54</td>
</tr>
<tr>
<td>3.8.3 Second-Law Efficiency of Steady-Flow Devices,</td>
<td>55</td>
</tr>
<tr>
<td>CHAPTER FOUR</td>
<td>58</td>
</tr>
<tr>
<td>GARRI 1 POWER PLANT</td>
<td>58</td>
</tr>
<tr>
<td>4.1. SITE LOCATION</td>
<td>59</td>
</tr>
<tr>
<td>4.2. WEATHER CONDITIONS</td>
<td>59</td>
</tr>
</tbody>
</table>
4.3. DESIGN OPERATING CONDITIONS ................................. 59
4.4. PLANT CONFIGURATIONS ............................................ 60
4.5. OPERATING SCHEME OF THE CCPP ......................... 60
4.6. GAS TURBINE WITH EXHAUST GAS SYSTEM .......... 63
   4.6.1 Gas Turbine ......................................................... 63
   4.6.2 Exhaust Gas System .............................................. 65
4.7. WATER STEAM CYCLE ............................................... 65
4.8. HRSG AND ITS AUXILIARY SYSTEM ......................... 66
4.9. THE MAIN STEAM AND BYPASS SYSTEM .................. 66
4.10. DEAERATOR HEATING STEAM SYSTEM .................... 66
4.11. BOILER FEED-WATER SYSTEM ................................. 67
4.12. CONDENSATE SYSTEM .............................................. 67
4.13. CIRCULATING COOLING WATER SYSTEM ................. 68
4.14. AUXILIARY SYSTEM .................................................. 68
4.19. HEAT RECOVERY STEAM GENERATOR (HRSG) .......... 69
   4.19.1 General ............................................................. 69
   4.19.2 Type of Boiler ................................................... 69
   4.19.3 Boiler Performance Features ................................. 69
   4.19.4 Boiler Inlet Duct, Main Stack and Duct Accessories 70
4.20. STEAM TURBINE ....................................................... 71
CHAPTER FIVE ................................................................. 78
EXERGY ANALYSIS FOR GARRI 1 POWER PLANT USING EXCEL ................................................................. 78
5.1. EXERGY ANALYSIS FOR STEAM CYCLE (TABLES (5-1,2,3)) ............................................................. 79
   5.1.1. HRSG: ............................................................... 79
   5.1.2. Steam turbine (ST): ............................................. 80
   5.1.3. Condenser ........................................................ 80
   5.1.4. Feeding system (pumps) ...................................... 81
5.2. EXERGY ANALYSIS OF MAIN EQUIPMENT FOR GARRI
    POWER PLANT (TABLE (5-4)) ........................................ 81
   5.2.1. Gas turbine one set .......................................... 81
   5.2.2. HRSG one set ................................................ 83
   5.2.3. Steam turbine set (combined by 2 HRSGs) ............. 84
CHAPTER SIX ................................................................. 89
RESULTS AND DISCUSSION .................................................. 89
6.1. EXERGY ANALYSIS OF CCPP ....................................... 90
   6.1.1. Brayton cycle theoretical efficiency .................... 94
   6.1.2. Rankine cycle theoretical efficiency .................. 94
6.2. EFFECT OF PARTIAL LOAD ON CCPP EFFICIENCY ........ 95
6.3. THERMODYNAMICS ANALYSIS .................................. 96
6.4. EVALUATION OF TEMPERATURE AND HEAT
    ABSORPTION FOR HRSG ........................................... 98
6.5. EFFECT OF PINCH POINT ON PLANT PERFORMANCE ... 101
CHAPTER SEVEN .............................................................................. 103
CONCLUSION AND RECOMMENDATION .................................... 103
7.1. CONCLUSION .......................................................................... 104
7.2. RECOMMENDATION .............................................................. 105
References .................................................................................................. 106