Energy and Exergy Analysis of a Combined Cycle Power Plant

By

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It is hereby declared that this thesis or any part of this thesis has not been submitted elsewhere for any award or any degree or any diploma.

October,2020

MD. MEZBA UDDIN

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ABSTRACT

In Bangladesh, the primary fuel is natural gas, predicted to be exhausted by 2025, and the availability of other fuels is also limited. Therefore, it is important to ensure proper utilization of available natural resources, which can be achieved by performing energy analysis and the exergy and exergoeconomic analysis. A detailed energetic, exergetic and exergoeconomic analysis of the Fenchuganj Combined Cycle Power Plant has been carried out in this thesis to investigate the possible scope of improvement of plant’s performance. Basic thermodynamic properties of the systems have been determined by energy analysis utilizing main operating conditions. Exergy destructions within the system and exergy losses to the environment were investigated to determine thermodynamic inefficiencies in the design and guide future plant improvements. Exergoeconomic analysis was carried out to assess the cost-effectiveness of individual components. Among the different approaches for thermoeconomic analysis in literature, Specific Exergy Costing (SPECO) method was applied to calculate each product, Per unit fuel cost and unit total generation cost. This thesis has determined the comparison of the configurations in terms of performance assessment parameters and costs per unit of exergy. The average range of thermal efficiency of the gas turbine is found 30.23%, the overall efficiency of the plant in simple cycle operation is 30.46%, and the plant's combined cycle efficiency is 43.21%. The average Heat Rate in simple cycle operation is 11,641.32 kJ/kWh, and the average Heat Rate in combined cycle operation is 8,369.55 kJ/kWh. The exergoeconomic analysis results indicated that the combustion chamber was the most cost-effective component, which a low cost of capital investment can improve. Comparing the Fenchuganj Combined Cycle Power Plant with other plants also represents that the combustion chamber is the plant's highest exergy destruction component. The heat energy loss in the combustion chamber decreases with an increase in air mass flow rate. This implies that a high mass flow rate of air can minimize the combustion chamber's energy losses as this would introduce more air for combustion. The component with the highest exergy improvement potential is the combustion chamber.

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Chapter 1: Introduction

1.1 Introduction

Electricity plays an essential role in the economic growth through the sustainable structure and poverty eradication and security of any country. A reliable electricity supply is a vital issue for the world today. Future economic growth crucially depends on long-term electricity availability, affordable, available, and environmentally friendly. Security, climate change, and public health are closely interrelated with electricity.

Nowadays, efficient usage of current energy resources is pretty significant because, with the continuously increasing population and intense industrialization rates demanding too much energy, our resources will be insufficient to meet this demand in a near-future. Thus, finding new energy resources and using present resources more efficiently are the critical concepts of the new century.

To achieve overall economic development and upgrade the socio-economic condition and alleviate the poverty of Bangladesh, the pace of power sector development has to be accelerated. The government needs to confirm a sustainable and quality supply of power to the country to maintain this growth rate in the power sector.

In line with this aspect, the Bangladesh Government designed an extensive power generation plan to create sustainable growth in the power sector and its economy's overall development. The Government of the People's Republic of Bangladesh (GOB) has taken a goal to provide affordable and reliable electricity to all Bangladesh citizens by 2020. GOB is currently working to provide sustainable electricity for the population by 2021, and the installed capacity of generation will be 24000 MW.

Power plants, much like other businesses, are first and foremost driven by economics. Electricity need is increasing, whereas the available generation also increases against demand. In the public sector, many generation units have become very old and have been operating at a reduced capacity.

Other issues are that the world is not energy-efficient, so we need to ensure optimum usage and savings of energy. Its impact daily is high. One way to use energy resources more efficient is to produce electrical energy from combined cycle power plants, which is one of the most popular energy conversion systems used around the world today. Using the Combined Cycle, it’s possible to directly reduce fuel consumption to the other electricity generation method.

In Bangladesh, the primary fuel is natural gas, predicted to be exhausted by 2025, and the availability of other fuels is also limited. For those reasons, the continued advancement and optimization of power plants is necessary. The power plant's performance can be enhanced by improving the overall system design, individual component performance, and plant operation management.

A comprehensive analysis should conduct to identify the areas with the most potential for improvement. Improving the methods of analysis can translate to further optimization of the systems. To maintain the efficient operating conditions of the plants, performing performance analysis is a requirement. Conservation of energy is a key to saving excessive bills and building up needless domestic and industrial use, and these analyses are focused mainly on the first thermodynamics rule. More recently, power plant performance examines from thermodynamics' second law, where exergy use as a performance parameter.

The energy demand is growing; optimization of energy conversion systems is indispensable. Exergy analysis usually predicts the thermodynamic performance of an energy system and the efficiency of the system's components. The exergy analysis of a system is closely related to its economic indices. It is an exergy that is paid for and not the energy that is thought to be.

Exergoeconomics is a branch of engineering science that combines exergy analysis and economic principles. This analysis enables the determination of exergy loss and destruction cost with these two concepts. In the last few decades, extensive studies have shown the importance of exergy analysis and its economic value.

For proper utilization of natural resources, it's crucial to perform energy analysis and the exergy and exergoeconomic analysis of a plant. Not only for the improvement of the plant's performance but also to reduce the excessive consumption of natural resources and keep the environment balanced.

In this thesis, a detailed study based on the energy, exergy, and exergoeconomic analyses of a combined cycle power plant of Bangladesh Power Development Board located at Fenchuganj, Bangladesh, has been carried out.

1.2 Power Sector Scenario in Bangladesh

Bangladesh's power sector is a proud sector of the country, with an installed power generation capacity of 23,548 MW as of July 2020 (including captive and renewable energy). The power sector of Bangladesh is one of the booming sectors of the country. Now 99% of the total population has access to electricity. The largest energy consumers in Bangladesh are the residential sector, followed by industries, commercial and agricultural sectors.

A few years back, high system loss, low plant efficiency, erratic power supply, shortages of funds for power plant maintenance, and absence of a new power generation plan were the big problems in Bangladesh's power sector, but now the scenario is entirely different.

The government has given top priority to power sector development and has committed to providing electricity to all citizens across the country by 2021. To achieve this goal Government has undertaken several reform measures; some of them have already been implemented.

The Electricity Directorate was established in 1948 to plan and improve the power supply situation of the country. Considering the increasing demand for electricity and its importance in agriculture & industry, "Water & Power Development Authority" (WAPDA) was created in 1959. Later the "WAPDA" was divided into two parts, namely "Bangladesh Power Development Board" & "Bangladesh Water Development Board" by the Presidential Order 59 (PO-59) of 31st May 1972. As a result, Bangladesh Power Development Board was entrusted with the Operation, Maintenance, and Development of Generation, Transmission & Distribution facilities of electricity throughout the country.

As part of reform and restructuring, the transmission was vertically separated. A subsidiary of BPDB and distribution was horizontally split to create new distribution entities in the capital city (DPDC & DESCO) and rural areas (REB). Further, some generation and urban distribution companies were created as a subsidiary of BPDB. The subsidiaries of BPDB are:

- Ashuganj Power Station Company Ltd. (APSCL)
- Electricity Generation Company of Bangladesh Ltd. (EGCB)

- North West Power Generation Company Ltd. (NWPGCL)
- Power Grid Company of Bangladesh Ltd. (PGCB)
- West Zone Power Distribution Company Ltd. (WZPDCL)
- Northern Electricity Supply company Ltd. (NESCO)

BPDB is under the Power Division of the Ministry of Power, Energy and Mineral Resources, Government of Bangladesh. Key responsibilities of the Board are:

- Generation of electricity from its Power Plants.
- Power purchase from Public & Private Generation companies as a single buyer.
- Bulk sales of electricity to Utilities as a single buyer.
- Retail sales of electricity within its Four Distribution Zones.
- Preparation of Generation and Distribution Expansion Plan.
- Implementation of Generation & Distribution Projects as approved by the Government.

1.2.1 Present Power Generation Scenario in Bangladesh

Now Bangladesh has shown implausible achievement in the power sector. The Government's target has been implemented successfully and has even achieved a higher level of economic growth.

Electricity Demand is growing day by day. To mitigate the demand-supply gap, an aggressive plan is prepared by the Government for new generation addition. The highest peak generation till now was 12,893 MW on the date of 29 May 2019. The total energy generation was 70,533 GWh, 17.66%, and 12.53% in 2019 to 2020 fiscal year, which is higher than the previous year.

Distribution system loss is reduced gradually, and continuous improvement in the distribution sector is going on.

The Present Power Generation Scenario in Bangladesh shown in table 1.1 below:

Table 1.1: Power Generation at a Glance (2019-2020)

Total Power Generation Facility	:	138 Nos.
Generation Capacity	:	23,548 MW
Deaerated Capacity	:	19107 MW
Maximum Generation	:	12893 MW (29 May, 2019)
Total Consumers	:	34.3 Million
Transmission Line	:	11,650 Circuit Kilomiter (Ckt. Km)
Distribution Line	:	5,32,000 km
Power Import	:	1160 MW
Per Capita Generation	:	510 KWh
Access to Electricity	:	99%

(ref: Annual report of 2019-2020 of Power Division)

Power Division and Bangladesh Power Development Board are on the right track to achieve the goal of providing uninterrupted, quality, and reliable power to all citizens of the country by 2021 through integrated development of power generation, transmission and distribution system.

Our Prime minister's aim is to give 100% power accessibility to all the people of our country. And we have a slogan for that- "Sheikh Hasinar uddog ghore ghore biddut."

1.2.2 Fuel based Power Plant Scenario of Bangladesh

Natural gas is currently the primary fuel for power generation in Bangladesh, but this finite resource won't last forever. Recognizing the importance of primary fuel for power generation, the government is diversifying the fuel mix by gradually shifting to coal, LNG, and other available fuel besides gas. To ensure energy security, the government has prepared Power Sector Master Plan-2016 considering gas, coal, LNG, liquid fuel, dual-fuel, nuclear, and renewable energy resources. The government has also taken initiatives to import power from neighboring countries. The present energy generation scenario with various types of fuels is presented in table 1.2 below:

Table 1.2: Fuel based Energy Generation Capacity (On February 2020)

Fuel Type	Capacity(Unit)	Total(%)
Coal	444	2.32
Gas	10261	53.70
HFO	5206	27.25
HSD	1771	9.27
Hydro	230	1.20
Imported	1160	6.07
Solar	35	0.18
Total	**19107**	**100**

(ref: Annual report of 2019-2020 of Bangladesh Power Development Board)

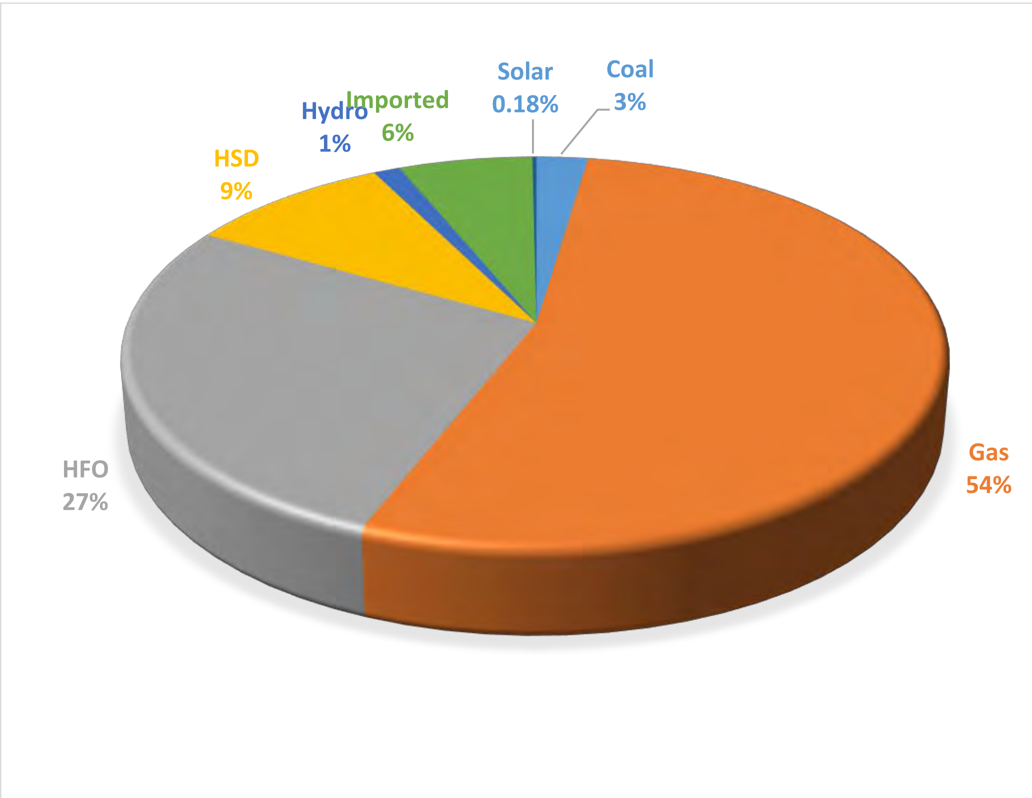

Figure 1.1: Net Energy Generation based on fuel use on February 2020.

1.2.3 Technology based Power Plant Scenario of Bangladesh

The present energy generation scenario with the basis of Technology based is presented below:

Table 1.3: Technology based Energy Generation Capacity (On February 2020)

Technology	Installed Capacity (MW)
Gas Turbine	1607
Reciprocating Engine	7372
Steam Turbine	2344
Combined Cycle	6364
Hydro	230
Solar	30
Power Import	1160
Total	**19107**

(ref: Annual report of 2019-2020 of Bangladesh Power Development Board)

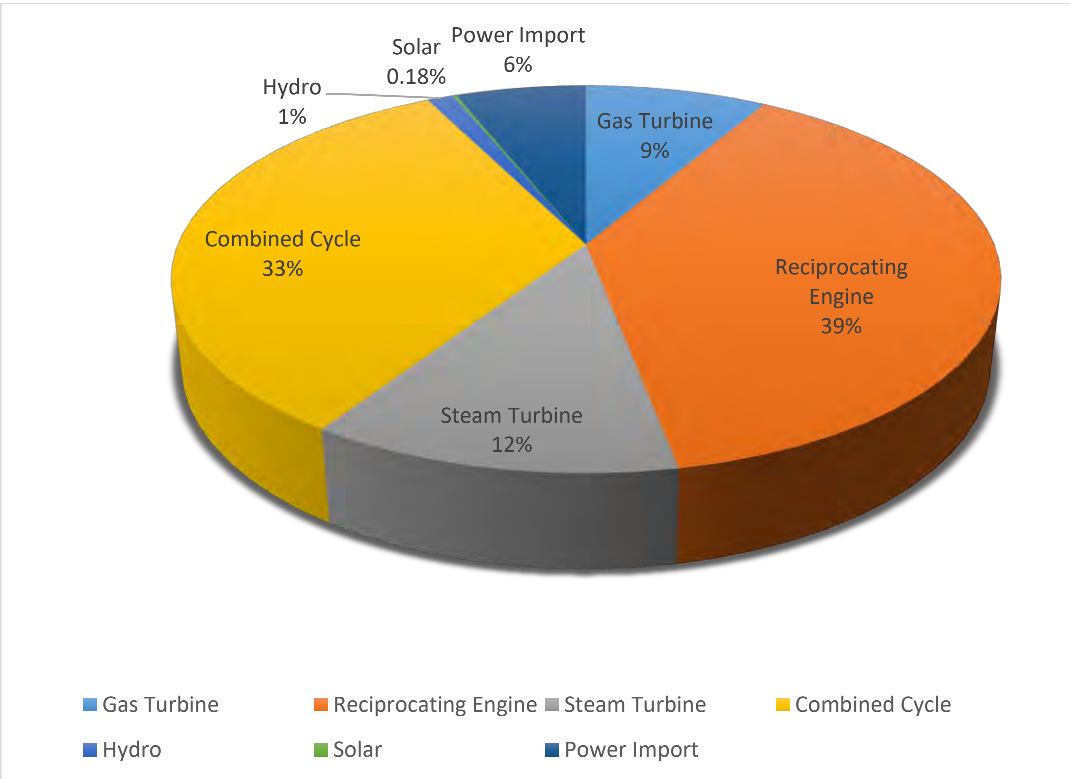

Figure 1.2: Net Energy Generation based on Technology based on February 2020.

1.3 Background and the present state of Energy and Exergy Analysis

The present situation in terms of energy efficiency is not very encouraging. There are serious shortcomings in energy utilization which are posing great concerns to the economic and environmental situation of the country. A systematic approach for decision-making in the area of energy management, energy audit is the key. It tries to balance the total energy inputs with its use, and serves to identify all other energy sources in a facility. It quantifies energy usage according to its separate quantity. Industrially energy audit is most important and feasible tool in defining and pursuing comprehensive energy management program.

A Combined Cycle Power Plant couples two power cycles such that the energy discharged by heat from one cycle is used partly or wholly as the input to the power cycle. This is how a combined cycle power plant works to produce electricity and captures waste heat from the gas turbine to increase efficiency and electrical output. The specific objective of energy audit of a thermal power plant is to improve the heat rate and reduce auxiliary power consumption of the power plant. Traditional energy audits are broad in scope and usually examine all energy utilizing systems based on the energy quantity. Energy has both quantity and quality. Quality aspect of energy is examined by the Exergy analysis which determines the magnitude and direction of irreversible processes in a system and thereby provides an indicator that points the direction in which engineers should concentrate their efforts to improve the performance of the considered system. Such a detailed energy analysis has not been conducted before for a combined cycle power plant in Bangladesh.

1.4 Definitions

Whenever the following terms appear in this thesis, they shall have the meanings given to such terms below:

Approach temperature: Approach temperature is defined as the difference between the saturation temperature and water temperature entering evaporator.

Circuit kilometer: means one kilometer of electrical transmission or distribution circuitry including all necessary conductors, insulators and supporting structures required to provide a complete circuit or double circuit.

Cooling tower effectiveness (in percentage) is the ratio of CTrange, to the ideal range, i.e., difference between cooling water inlet temperature and ambient wet bulb temperature, or in other words it is = Range / (Range + Approach).

Cooling tower bleed-off/blowdown: is the flushing of a portion of high mineral concentration cooling tower system water down the drain, while simultaneously replacing it with fresh water. This process dilutes the system water mineral concentrations that steadily increase due to water evaporation.

Condenser Effectiveness: ratio of actual heat transferred to maximum possible heat that can be transferred.

Condenser Efficiency: The ratio of temperature rises of cooling water to the maximum possible temperature rise (i.e. the vacuum temp. minus inlet cooling water temperature).

Cycles of Concentration: is the number of times the concentration of total dissolved solids (TDS) in cooling tower water is multiplied relative to the TDS in the makeup water. Because evaporation of pure water leaves dissolved solids behind in the system water, TDS increases over time as the tower operates.

Density: Mass per unit volume is called density. Density of a material changes with the change in temperature and pressures. The unit of density is kg/m^3.

Dependable Capacity: means at any given time the net amount of capacity of the facility (adjusted to Reference Site Conditions in accordance with Section 13.1 and Power Factor Adjustment), expressed in kW, as determined by the most recent Dependable Capacity Test.

Drift losses: Drift losses refer to the amount of total tower water flow escaping the cooling tower as droplets of water, in some towers it like being rained on, unlike evaporated water that leaves the tower with the discharged air, when water evaporates it drops out all minerals and dissolved solids that are left behind causing the TDS, (total dissolved solids) of the remaining water to increase.

Energy: bears the meaning ascribed thereto in Chapter 5.1.

Evaporation Loss: The loss of a stored volatile liquid component or mixture by evaporation; controlled by temperature, pressure, and the presence or absence of vapor-recovery systems.

Exergy: bears the meaning ascribed thereto in Chapter 6.1.

FAD: is the amount of free air drawn into the compressor that is actually delivered by the air compressor at its compressed air outlet.

Heat rate: The amount of heat input in kJ/hr for each kWh of electricity produced. used to indicate the power plant efficiency. The heat rate is the inverse of the efficiency: a lower heat rate is better.

Isothermal power: is the least power required to compress air assuming isothermal (constant temperature) compression conditions.

Log- Mean Temperature Differences: The log mean temperature difference (LMTD) is used to determine the temperature driving force for heat transfer in flow systems, most notably in heat exchangers. The LMTD is a logarithmic average of the temperature difference between the hot and cold streams at each end of the exchanger. The larger the LMTD, the more heat is transferred.

Pinch point: Pinch point is defined as the difference between the temperature of exhaust gas exiting the evaporator and saturation temperature of the steam corresponding to the related pressure level.

Specific Heat: The specific heat is the amount of heat per unit mass required to raise the temperature by one degree Celsius. The fluid having higher specific heat ensures higher rate of heat transfer. The unit of specific heat is J/kg.K.

Steam Purity: Steam purity can be maintained by using a correct feed water treatment operation and an adequate blowdown procedure.

Steam Rate (S.R): The capacity of a steam plant is often expressed in terms of steam rate or specific steam consumption. It is defined as the steam flow rate (kg.s) required to produce unit shat output.

$$\text{Steam Rate (S.R)} = \frac{m_s}{W_{net}}$$

Where;

m_s = rate of steam flow (kg/s);

W_{net} = power developed (kW).

Volume rate of flow: is the actual volume of gas, compressed and delivered at the standard discharge point, referred to conditions of total temperature, total pressure and composition prevailing at the standard inlet point.

Windage loss: is the reduction in efficiency due to windage forces.

1.5 Motivation for the Study

The role of electricity in modern society is so inherent that we do not even think about it. It is self-evident that all the appliances at home and the office, all the communications, heating, air-conditioning, lighting, many modes of transportation, etc. But they would not if there was no electricity.

Another critical issue is the reserves for fossil fuels are limited, and, what is more important, it is impossible to make use of all of those reserves. They will not be exhausted for a few decades at the current rate, but they will become a more and more marginal source of energy. Even before that, they were using fossil fuels to increase substantially because of the higher costs for acquiring them and the carbon taxes levied on burning them.

There is a limit for the available amount of many non-combustion and renewable energy sources, too. The solution can't be just increasing the production of electricity with a potential increase in the severity of the problems. Whatever the method, there is no possibility for exponential growth. There are limits set by technology, economics, and resources.

An obvious solution is to use energy more efficiently. The energy efficiency of electric appliances has vastly improved. With the same amount of energy- or electricity – their output is much larger. Because of the rising standard of living, the total number of appliances in the world will inevitably and continuously increases, which would also help decrease the need for expanded production of electricity.

Energy efficiency and energy conservation are also helping us to lessen the use of electricity. With the efficient use of electricity, it is also possible to make modern society less dependent on fossil fuels. The efficient use of electricity would have an enormous impact on the diminishing of CO2 into the atmosphere, which is essential in slowing down climate change and global warming.

Presently in Bangladesh, 138 nos. Power plants are under operation to produce electricity continuously. And the Installed generation capacity is 23,548 MW, and the recent deaerated Capacity is 19,107 MW. But till now, the maximum generation capacity is 12,893 MW. The gap between the maximum generation and deaerated Capacity is enormous. Most of the time, some plants shut down due to a lack of fuel (gaseous fuel), machine problems in plants, low dispatch requirements, etc.

Reducing the gap between the installed capacity and maximum generating capacity requires high time to focus on power plants' energy and exergy efficiency. And make the plant available efficiently and ensure sustainable electricity for all.

Therefore, in this study, a detailed study based on the energy, exergy, and exergoeconomic analyses of a combined cycle power plant is carried out. Basic thermodynamic properties of the systems determined by energy analysis utilizing main operation conditions. Exergy destructions within the system and exergy losses to the environment investigated to determine thermodynamic inefficiencies in the design and to assist in guiding future improvements in the plant. Exergoeconomic analysis was done to determine the cost-effectiveness of improving individual components. Among the different approaches for thermoeconomic analysis in literature, SPECO method is applied. Hence, cost of each product is calculated. And, also find out the Per Unit Fuel cost and per unit total generation cost is also calculated. Comparison of the configurations in terms of performance assessment parameters and costs per unit of exergy are also given in this thesis.

1.6 Objectives & Possible Outcomes

The main objectives of this thesis are as follows:

a. To study the operational procedure and characteristics of the combined cycle thermal power plant (CCPP) and develop a mathematical formulation for mass and energy flow rates at steady-state conditions to match with plant operational data.
b. To measure the loss of energy to the environment from a combined cycle power plant.
c. To carry out the energy and exergy analyses of the plant and draw the energy and exergy flow diagram.
d. To compare the plant performance based on the energy and exergy viewpoint.
e. Further, a thermodynamic analysis of the components of the plant will be carried out to understand the methodology of thermoeconomic evaluation and to identify the scopes for the energy and exergy performance improvement of these components to utilize the waste energy and its applicability in designing a cost-effective thermal system.

1.7 Organization of the thesis

The study presented in this dissertation has addressed the Energy, Exergy, and Exergoeconomic analysis of a combined cycle power plant. The dissertation has organized as follows:

Chapter 1 presents a general discussion on Energy, Exergy, and Exergoeconomics, the Power sector scenario in Bangladesh. Background and the present state of Energy and Exergy Analysis have been discussed. The motivation for this work and the objective of this dissertation have also been discussed.

Chapter 2 presents a detailed literature review for a better conception in previous studies and energy, exergy and exergoeconomic analysis method of combined cycle power plants that have carried out by different researchers across the world. The applications of this analysis in other sectors are also discussed.

In chapter 3, a small description of the background of a combined cycle power plant and the Configuration of all the main equipment's has been presented here. I also discussed in that chapter how this main equipment's are worked in combined cycle power plant & their effectiveness.

In chapter 4, the analysis methodology of this work has been discussed. The purpose of using this process in this work have been discussed here.

In chapter 5, discuss the energy analysis of the fenchuganj combined cycle power plant in detail and its application to case studies. Here, Performance assessment parameters related to energy concepts are described and analyzed for their change with steam demand, the pressure of high-pressure steam drum, and pinch point. The results of the energy analysis of this plant are also discussed in this section along with some recommendations.

In chapter 6, discuss the exergy analysis. For exergy analysis includes the formulation of exergy terms, exergy destructions within the plant, exergy losses to the environment, exergetic efficiency of the plants, and ratios related to exergy destruction and exergy loss.

In chapter 7, discuss with exergoeconomic/thermoeconomic analysis methodology of the plant and its application to case studies. Engineering economics analysis is studied. The general methodology used in such analysis is given. The results of these analyses are discussed in this chapter.

Finally, conclusions of this work and recommendation for future work have been discussed in chapter 8.

Chapter 2: LITERATURE REVIEW

The following literature review focuses on the energy, exergy and exergoeconomic analyses. A representative collection of works and their related findings are presented, with greater emphasis on papers from the past decade.

2.1 Conventional Energy Analysis

To reduce energy costs, lead to higher performance and save energy, Energy Audit is a powerful tool for exposure operational and equipment improvements. Sometimes, the energy audit is also called an “energy assessment” or “energy study”.

Energy audits can be done as a stand-alone effort but may be conducted as part of a larger analysis across an owner’s entire group. The purpose of an energy audit is to find out how, when, where and why energy is used.

The energy audit is also used to identify opportunities to improve efficiency. Energy auditing services are offered by engineering firms, energy services companies and energy consultants. The energy auditors do the audit process. The first thing energy auditor needs to be aware of end user expectations and then audit starts with an analysis of historical and current utility data. This sets the stage for an onsite inspection. The most important outcome of an energy audit is a list of recommended energy efficiency measures (EEMs). Energy audit serves the purpose of identifying energy usage within a facility, process or equipment, and then identifies opportunities for conservation, called energy conservation measures (ECMs). Audit provides the most accurate picture of energy savings opportunities.

Many researchers have done Energy analysis in past decade. Few energy analysis of thermal power plant is discussed below:

Cropper Paul A. et al. (1991) [1] discussed the audit's objective was to make a comprehensive study of the power plant operations and programs relating to the performance of the generating units and to identify areas of potential improvement. The results indicated that Heat rate improvements had been achieved, and an already high level of reliability had even further improved.

Bhansali V.K. et al. (1995) [2] presented the energy conservation was cost effective with a short payback period and modest investment. The results indicated that the energy conservation should be developed as a mass movement like family planning, literacy drive etc.

Khan Atif Zaman (1996) [3] discussed the application of the ECON techniques by which electrical energy could be saved and made cost efficient from the industrial perspective were presented for a sheet-glass industry in a developing country. The results indicated that recommendations if applied to any similar industry in other developing countries may also lead to very reasonable cost savings.

Babu N. Sundar et al. (1999) [4] described the Government has given higher priority for the power development projects, the Indian Power sector was struggling with formidable difficulties of meeting the heavy demands of electricity due to higher amount of power losses and energy thefts. The results indicated that their main functions for economic and emission controlled operation of the Power sector had been elaborately analyzed energy. This is achieved by raising the steam in the boilers, expanding it through turbine.

Hogg B. W. et al. (1999) [5] presented reviewing the performance monitoring practice of a typical 200 MW oil/gas-fired thermal power plant at Ballylumford, N. Ireland, results indicated that The possible best efficiency values of plant components, needed for comparative performance evaluation, were more reliably and accurately obtainable through neural network performance models.

Bathaee S.MT et al. (2000) [6] described the NEKA steam power plant's auxiliary service system in the North of Iran as analyzed from the reliability point of view. The results indicated that the applied indexes in the auxiliary service system of power plants reliability calculations. Theoretically, some of these systems could model, and the reliability calculation could be done in a simple form by the cut-set method.

Bose Bimal K. et al. (2000) [7] discussed global energy generation scenarios and the related environmental pollution problem. The results indicated that Electric/hybrid vehicle technology that played an essential role in fuel-saving and pollution control had been reviewed in this context.

Sabooh Dr. Y. (2000) [8] presented the model validated by experimenting with an EAF transformer in Ahvaz Steel Making Plant. The analysis results indicated that the simulation model could control the transformer's hot spot temperature. It provides an appropriate means of increasing the EAF transformer's reliability and preventing further damages to copper windings.

Najjar (2001) [9] described that a gas turbine engine's efficiency is relatively low at the design point, and it deteriorates further at part load and off-design high ambient temperatures. His work comprises of the study of adding an inlet air pre-cooler driven by the tail-end heat recovered from the engine exhaust gases. A heat recovery boiler was used to recover the exhaust heat partly. This combined system's performance, namely power, efficiency, and specific fuel consumption, was studied and compared with the simple cycle. This parametric study's variables were mainly compressor pressure ratio, turbine inlet temperature, and ambient temperature. Results show that the combined system achieves gains in power. The performance of the combined system showed less sensitivity to variations in operating variables. Thermoeconomic evaluation shows that the combined system is viable.

Lee Wei-Jen et al. (2002) [10] discussed energy management and energy conservation for motors, systems, and electrical equipment. The results indicated that electric motors constitute about 70% of energy consumption. Special attention was given to their types, design, characteristics, applications, sizing, and utilization.

Han Pu et al. (2003) [11] discussed the PFC algorithm for first-order plus dead time system was provided, another novel PFC algorithm based on Finite Impulse Response (FIR) model was also presented. The results indicated that the algorithm had no difference due to the variety of set points and output disturbances.

Changliang Liu et al. (2004) [12] presented a genetic algorithm introduced to identify the transfer functions and parameters non-linear dynamic model of the thermal process. The results indicated that the genetic algorithm-based identification method was reasonable, and satisfactory results could be obtained with it.

Ameri and Hejazi (2004) [13] observed that the ambient temperature variation causes a loss of 20% of the 170 gas turbine units' rated capacity in Iran. They studied five gas turbines, where the difference between the ambient temperature and the ISO conditions was 11.8 °C. They found that for each 1 °C increase in ambient temperature, the power output was decreased by 0.74%, and they suggested cooling the compressor's intake-air temperature to improve the gas turbine cycle efficiency.

Kim Hoyol et al. (2006) [14] discussed the control strategy for the new thermal power plant having the steam condition of 3769psi and 1130F, and the rated utility output of 1000MW was developed. This paper showed the system structure and information flow scheme. This paper also showed the control strategy strictly following the hierarchical structure of the integrated control and management system.

Boonnasa et al. (2006) [15] studied the performance improvement of an existing combined cycle power plant located in Bangkok that consisted of two gas turbines (110.76MW each) and 115.14MW steam turbines in ISO conditions. The plant used an absorption chiller to cool one of the two gas turbine's intake-air to 15°C, in addition to having a thermal energy storage tank that stored the sensible heat of the chilled water to meet the varying daily cooling load. Low-pressure steam from a heat recovery steam generator was used to drive the absorption chiller needed to meet a maximum load of 7049.58kW with thermal heat storage. As a result, the cooled gas turbine's power output increased by 10%, improving the CCPP total power output by 6.24%. Economically, the study found that due to the low initial investment cost of retrofitting the absorption chiller, the internal rate of return was 40%, and the payback period was just 3.81 years. However, the authors also reported a reduction of 2.85% in the steam turbine power output, which was due to powering the absorption chiller directly from the HRSG unit steam that was powering the steam turbine. This reduction in the steam turbine power output could have been avoided if they had used a boiler that utilized the waste heat energy from the stack after the HRSG unit.

Gupta J.B. (2006-07) [16] discussed that thermal energy is the primary source of power generation itself, showing the importance of thermal power generation in India.

Rajput R.K. (2006-07) [17] presented a steam power plant that converts fossil fuel's chemical energy into mechanical.

Guo Ying et al. (2007) [18] proposed a multi-objective optimizing control method, including two optimization procedures. The results had demonstrated the satisfactory performance of this novel multi-objective optimizing control strategy. One future research topic was to develop a more efficient multiple fuzzy objectives optimizing predictive controller for real-time implementation.

Hosseini et al. (2007) [19] indicated that the gas turbine compressor is designed for constant air volume flow, making the electric power output depending on the ambient temperature through the specific mass flow rate. They added that the ambient temperature increase also decreases the compressor's output pressure, which reduces the gas turbine cycle efficiency, while the increase in the air density reduces the gas turbine's heat rate and increases its specific fuel consumption. They stated that for each 1°C increase in the ambient air temperature, the gas turbine's electric power output decreases by 0.5% to 0.9%, and by 0.27% for a combined cycle.

Xiang and Chen (2007) [20] discussed the combined cycle with three-pressure HRSG, equipped with the GE PG9351FA gas turbine. They maximized the combined cycle efficiency through the optimization of the HRSG operating parameters by minimizing exergy losses. Moreover, they highlighted the influence of the HRSG inlet gas temperature on the bottoming cycle efficiency. They studied the influence of HRSG inlet gas temperature on the steam bottoming cycle efficiency. Their result shows that increasing the HRSG inlet temperature has less improvement to steam cycle efficiency when it is over 590ºC.

P.S.P.C.L. manual (2007) [21] presented thermodynamics as the essential subject of thermal Engineering. It deals with gases and water vapors' behavior when subjected to varying temperatures pressure. In the thermal plant, heat energy is converted into mechanical energy, further converted into electrical energy.

Guo Tieqiao et al. (2008) [22] described continuous Emissions Monitoring Systems (CEMS) permanently installed for selected processes require initial and periodic quality assurance audit tests to ensure that they were functioning correctly and reporting emissions accurately. Results indicated that facility would perform continuing quality assurance (Q.A.) on the CEMS.

Bera S. C. et al. (2008) [23] presented the typical I.D. fans' performances in the steam generation unit of naphtha, and natural gas-based captive power plant had been studied. It had been a considerable part of the losses in the I.D. fan was due to the over design than the current requirement, and the older design was another cause of low efficiency. So from this performance calculation of an I.D. fan a better design of a fan had been proposed to improve the plant efficiency and save the energy for global interest.

Kamalapur G D et al. (2009) [24] presented the achievements and further challenges of electrical energy conservation in the Indian context. The results indicated that the Energy Conservation Act- 2001 by the Government of India was the first step in this direction and had given encouraging results.

Jiang Tie-Liu et al. (2009) [25] discussed the energy audit concept, and the purpose was analyzed. The significance of energy audit in coal-fired power plant to energy-saving and emission-reducing was introduced by analyzing the current utilization situation of energy in China's coal-fired power plant. Energy audit contents such as energy management audit, energy utilization audit, and energy-saving potential analysis showed.

Qingsheng Bi et al. (2009) [26] presented two modes of heat pump heat regenerative of the system and established mathematical models for energy saving critical points. Through comparative study and analysis in the actual case, the results demonstrated that only when the Coefficient of heat pump performance (C.O.P.) was larger than the Coefficient of performance of heat pump of energy-saving critical points, the thermal system was energy saving.

Khaliq (2009) [27] reported in their detailed analyses that the combustion chamber's exergy destruction increases with the cycle temperature ratio, and the second-law efficiency of the primary combustor behaves in reverse from the second-law analysis. Increasing the maximum cycle temperature gives a significant improvement in both efficiency and specific work-output. The study also concludes that the efficiency reduces rapidly with a reduction in the TIT.

Evonik Industries (2009) [28] Energy efficiency is vital for sustainable development. Energy conservation Act of 2001 provides a legal framework.

Mohagheghi and Shayegan (2009) [29] performed the thermodynamic optimization of design variables and heat exchangers layout in a heat recovery steam generator HRSG for combined cycle gas turbine CCGT using a genetic algorithm. Their method was introduced for modeling the steam cycle in advanced combined cycles by organizing the non-linear equations and their simultaneous used solutions with numerical methods. In addition to the optimization of the recovery boiler's design variables, they performed the distribution of heat exchangers among different sections and optimized their layouts in HRSGs. A standard gas turbine was assumed, and then outlet gas stream conditions (mass flow rate, temperature, and chemical composition of gas stream) were considered the inlet parameters for the recovery boiler model. From the optimization process, maximum output power from a steam cycle for different HRSGs was then analyzed.

Tyagi and Khan (2010) [30] studied the effects of gas turbine exhaust temperature, stack temperature, and ambient temperature on the combined cycle power plant's overall efficiency, keeping the gas turbine efficiency and steam turbine efficiency constant. They concluded that the stack temperature should be minimum and gas turbine exhaust temperature should be maximum. Out of these three variables, i.e., turbine exhaust temperature, stack temperature, and ambient temperature, the dominating factor of increasing the combined cycle power plant's overall efficiency is the stack temperature.

Cao Lihua et al. (2010) [31] discussed how to analyze the reasons and sites that cause the standard coal consumption rate for generating increase, and provided the basis to power plants for their overhauling exploiting energy savings potential. The results indicated that the plant had tremendous energy-saving potential by reducing the condenser pressure and improving the condenser vacuum.

Gao Han et al. (2010) [32] presented the main factors of fly ash carbon content was analyzed to utilized B.P. neural networks for dynamic fitting fly ash carbon content value. Besides, the new method was checked up according to some coal-fired boiler's online data, which indicated the method could satisfy the request of practical application.

Chai Yuman et al. (2010) [33] described the calculation and analysis of energy destruction and energy efficiency was carried on for the specific working conditions based on the calculation of thermodynamic parameters and the energy destruction of the components was also conducted. The results indicated that the boiler's energy destruction was the largest in the system, which accounted for more than 90% of the system's energy destruction, so the boiler was the primary source of the irreversible loss of the power plant.

Woudstra et al. (2010) [34] performed the thermodynamic evaluation of combined cycle plants with the same gas turbine and different steam bottoming cycles. The evaluation showed that the increasing number of steam generation pressure levels would reduce the losses due to heat transfer in the HRSG and the exergy loss due to the exhaust of flue gas to the stack. Among the investigated configurations for the bottoming cycle, triple pressure reheat was the best option from an exergy point of view.

Bentarzi H. et al. (2011) [35] presented a new approach to controlling the steam turbine of a thermal power plant using a distributed controlled system with fuzzy logic technique. The results indicated that it was not generally suitable for non-linear, time delay, high order, and complex systems.

Ding Jinliang et al. (2011) [36] discussed the research results of modeling and simulation of thermal power units were reviewed. Several standard models for the research of thermal power control systems were analyzed, including simplified turbine and furnace models for the unit coordinated control system (C.C.S.) and local equipment models. The challenges of modeling and simulation of thermal power plant researches in the future were discussed.

Sanjay (2011) [37] stated that most of the parameter that affects cycle performance is the turbine inlet temperature TIT. The TIT should be kept on the higher side because the exergy destruction is higher at lower values.

T.E.R.I. (2011) [38] discussed the structure of the energy audit report is governed basically by the directives issued. The energy audit reports are details of energy consumption, their costs, and specific energy consumption institutional arrangement for promotion of energy efficiency.

Jankes Goran et al. (2012) [39] presented characteristics of energy consumption in the industrial sector of Serbia, the methodology and results of energy audits (E.A.) performed in industrial sites, and potentials for energy efficiency (E.E.) improvements. The results indicated that the present state of energy systems in the Serbian industry could be characterized by significant technological out-of- date, low E.E., and low level of environmental protection.

Ibrahim and Rahman (2012) [40] performed a parametric thermodynamic analysis of a combined cycle gas turbine. They investigated the effect of operating parameters, compression ratio, gas-turbine peak temperature ratio, isentropic compressor, and efficiency and air-fuel ratio on the overall plant performance. Their results show that the compression ratios, air to fuel ratio, and isentropic efficiencies are strongly influenced by the overall thermal efficiency of the combined cycle gas turbine power plant. The overall thermal efficiency increases with compression ratio as well as isentropic compressor and turbine efficiency. However, the variation of overall thermal efficiency is minor at the lower compression ratio, while it is very significant at the higher compression ratio for both the isentropic compressor and turbine efficiency. The combined cycle gas turbine's overall efficiencies are much higher than the efficiencies of gas turbine plants. The efficiency quoted range is about 61%. The overall thermal efficiency increases and total power output decreases linearly with the compression ratio's increase with constant turbine inlet temperature. The overall peak efficiency occurs at the higher compression ratio with the higher cycle peak temperature ratio and higher isentropic compressor and turbine efficiencies.

Mansouri et al. (2012) [41] investigated the effect of pressure levels of steam generation at heat recovery steam generator HRSG on the energetic and exergetic efficiency of HRSG, bottoming cycle and combined cycle power plants, as well as the effect of HRSG (heat recovery steam generator) pressure levels on exergy destruction at HRSG and other main components of the bottoming cycle. Their result shows that an increase in steam generation pressure levels at HRSG leads to an increase in the exergy efficiency of HRSG and CCPP increase respectively. Besides, an increase in pressure levels at HRSG decreases the exergy destruction due to heat transfer in HRSG: the exergetic efficiency of HRSG increases with an increase in pressure levels of steam generation and adding reheat to the cycle.

Abdul Rahman Almutairi et al. (2015) [42] presented the Energetic and exergetic analyses using operational data for Sabiya, a combined cycle power plant (CCPP) with an advanced triple pressure reheat heat recovery steam generator (HRSG). A detailed energetic and exergetic analysis of a 2000 MW CCPP has been carried out, based on a very advanced HRSG. The thermodynamic data were extracted from the CCPP model, developed by the IPSEpro software, and validated with the manufacturer’s published data, and it presented a high level of compatibility. The results confirmed that 60.9% of the total exergy destruction occurs in the combustion chamber, which constitutes the primary source of irreversibilities within a system, with high exergy destruction attributable to three reasons: (i.) combined diffusion/fuel oxidation; (ii.) internal energy exchange–heat transfer; and (iii.) the mixing process.

Inefficiency in the combustor can be reduced by improving the combustion process, adding an air preheater, and reducing the air-to-fuel ratio. The steam turbine constitutes the lowest source of irreversibility due to the reheating system and the working fluid. The variation in ambient temperature significantly affects the CCPP's exergetic efficiency and net power output. Also, recommend using a cooling system at the point of intake of the gas turbine engine to achieve high power output and exergetic efficiency. The reheat gas turbine engine, with a high-pressure ratio, will achieve substantial improvement in the CCPP due to a reduction in the fuel consumption, producing more power and maintaining the exhaust temperature at a high level, which is compatible with the triple pressure reheat HRSG. Adjusting the steam pressure at a high value and the condenser pressure at a low value can be proposed to improve the CCPP's efficiency. Regulating the HP pinch temperature at a low level will augment the CCPP's efficiency, while a high approach temperature is always preferable to recover more energy from the exhaust gases from the gas turbine.

H. Sreedharan et al. (2016) [43] presented the Energy and Exergy Analysis on 350MW Combined Cycle Power Plant. This paper discussed the variation of exergetic efficiency of the gas turbine for different operating parameters. The efficiency varies from 33.8 to 28.7, and combined cycle thermal efficiency varies between 42 and 55 as various parameters vary.

Abdallah Haouam et al. (2019) [44] presented the calculation and the analysis of the thermal performance of the components of an MS 7001 type gas turbine with a nominal power of 87 MW using the concept of exergy. The exergy analysis is carried out by applying the equilibrium equations obtained from the general definitions of the thermodynamic processes' irreversibility and the data provided by the manufacturer. The results show that the combustion chamber's efficiency is the lowest compared to that of the compressor and the expansion in the turbine because it is where the highest exergy destruction occurs. The total exergy destruction of the 84.55MW gas turbine is 53.51 MW, and its efficiency is 32.44% in the operating conditions. The change in ambient temperature has a direct impact on the exergy performance of the gas turbine. Exergy destruction increases with air increase in temperature while the exergy efficiency decreases.

2.2 Conventional Exergy Analysis

A basic definition of exergy, also referred to as work potential and available work, is the potential of a substance to do work. While in thermodynamic equilibrium with its surroundings, a substance has no potential to do work, and therefore its exergy is zero.

Fiasch and Giampaolo (1998) [45] investigated an exergy analysis of the semi-closed gas turbine combined cycle. They concluded that combustion, heat recovery steam generator, water injection/mixing, and water recovery system are the primary sources of the losses, representing globally more than 80% of the overall exergy destruction.

Song et al. (**2002) [46]** studied the exergetic performance of a CT and had several interesting results. They found the largest exergy destruction was due to chemical reactions in the combustor. The variable inlet guide vanes (IGV), despite being used to enhance part load performance, caused a significant increase in exergy destruction at the first compressor row, and a decrease in overall compressor efficiency. Also, turbine blade cooling air accounted for over half the total exergy destruction in cooled stages, significantly affecting overall turbine efficiency.

Exergy Analysis is the job of the power plant to extract as much of the inlet substances' work potential, and convert it to useable energy.

Giannantoni et al. (2005) [47] proposed the upgrade of a steam power plant to CCPP using a five-part approach: energy analysis, conventional exergy analysis, thermoeconomic analysis, environmental evaluation, and economic evaluation. They found the energy analysis to be the most helpful, however that is likely no longer the case, as exergy and thermoeconomic analysis methods have improved in the years since.

Cihan et al. (2006) [48] carried out energy and exergy analyses for a combined cycle in Turkey and suggested modifications to decrease the exergy destruction in CCPPs. Their results showed that combustion chambers, gas turbines, and HRSGs are the primary sources of irreversibilities, representing over 85% of the overall exergy losses.

Sciubba and Wall (2007) [49] defined exergy as the maximum theoretical useful work obtained if a system S is brought into thermodynamic equilibrium with the environment by means of processes in which S interacts only with this environment. Unlike an energy based analysis, an exergy analysis allows one to identify the location, magnitude, and origin of thermodynamic inefficiencies. This additional information is useful for improving the efficiency and cost-effectiveness of a system, and providing means of comparing multiple systems.

The idea of available work can be traced back almost two hundred years. According to Sciubba and Wall (2007), Carnot (1824) first stated that the work that can be extracted from a heat engine is proportional to the temperature difference between the hot and the cold reservoir. However, Clapeyron (1832, 1834), Rankine (1851), Thomson (1852), and Clausius (1850, 1867) all contributed to the establishment of the principle.

The earliest mention of the term available work was by Gibbs in 1873. Gibbs modeled an arbitrary substance as a triangular surface. He said that the surface is made up of three points, the properties of the substance: volume, energy, and entropy. The surface is acted upon by the two other properties, temperature and pressure. When the substance is in thermodynamic equilibrium with its surroundings, the triangular surface is tangent to the plane; if it is not, then it is shifted, and its coordinates change. The movement is parallel to the respective axis of each property. The coordinates for which the surface is tangent to the plane, and the substance is in thermodynamic equilibrium with its surroundings, will be referred to as at the dead state. Exergy is basically a measure of the distance between the coordinates of the old and new vertices of the triangular surface. Therefore, the work potential equals the sum of these three distances, with the sign of the entropy term being negative.

Butcher and Reddy (2007) [50] carried out an exergy analysis for waste heat recovery based power generation system. The waste heat recovery power generation systems' performance based on second law analysis was investigated for various operating conditions. The temperature profiles across the heat recovery steam generator (HRSG), network output, second law efficiency, and entropy generation number was simulated for various operating conditions. The variation in specific heat with exhaust gas composition and the temperature was accounted for in the analysis and results. The effect of pinch point on the performance of HRSG, entropy generation rate, and second law efficiency investigated. The researchers found that the HRSG and power generation system's second law efficiency decreases with increasing pinch points. The first and second law efficiency of the power generation system varies with exhaust gas composition and oxygen content in the gas. The results contribute further information about the role of gas composition, specific heat, and pinch point influence on the performance of a waste heat recovery based power generation system (based on the first and second law of thermodynamics).

Abusoglu and Kanoglu (2008) [51] applied the exergy destruction method to the diesel engine powered cogeneration systems generating electricity and steam. They defined the fuel and product in terms of exergy flow for each component of the system and then calculated the exergetic efficiency. It was observed that the total exergy destruction in the engine was mostly due to the highly irreversible combustion process in the engine, heat losses from the engine, and friction.

Kamate and Gangavati (2009) [52] analyzed cogeneration power plants in sugar industries through the exergy destruction method for various steam inlet conditions. The result shows that, at optimal steam inlet conditions of 61 bar and 475C, the backpressure steam turbine cogeneration plant performs with energy and exergy efficiency of 0.863 and 0.307, while the condensing steam turbine plant performs with energy and exergy efficiency of 0.682 and 0.26, respectively. The boiler is the least efficient component, and the turbine is the most efficient component of the plant.

Aljundi (2009) [53] studied energy and exergy analysis of a steam power plant in Jordan using the exergy destruction method. A component-wise modeling and a detailed break-up of energy and exergy losses estimated the performance of the plant. The modeling shows that the thermal efficiency (26%) is low compared to modern power plants because this efficiency not based on the specific heat input to the steam; instead, it was based on the lower heating value of the fuel to incorporate the losses occurring in the furnace-boiler system due to energy lost with hot gases, incomplete combustion, Etc. The maximum exergy destruction is in the boiler and maximum exergy loss in the condenser.

Gulen and Smith (2010) [54] derived a simple method for estimating the Rankine bottoming cycle output directly from the CT exhaust exergy. In a separate paper, Gulen and Joseph said that a combined cycle plant will run under boundary conditions significantly different from those under which individual components are designed. They also presented a simple, generalized, physics-based calculation method to estimate off- design performance.

Aklilu and Gilani (2010) [55] performed a conventional exergy analysis on a cogeneration plant running at varying part loads. They found the exergy destruction rate of the CC to be 50.6% to 63.7% of the overall system destruction rate, making it the largest source of destruction. They attributed the majority of destruction the chemical reaction and mixing of dissimilar fluid streams.

Bracco and Silvia (2010) [56] studied a combined cycle power plant with a single level heat recovery steam generator HRSG. They developed a mathematical model to determine the optimal steam pressure values in the HRSG according to different objective functions (in the HRSG for a given gas turbine). Their work reports numerical results for the combined cycle power plant considering four different gas turbines. The optimization approach focused on studying the heat transfer between the steam and the exhaust gas in the HRSG, based on an exergetic analysis. They present the comparison among different objective functions that refer to the HRSG specifically or the whole bottoming cycle. In their mathematical model, they considered the presence of specific constraints for the power plant's operating parameters, the most important constraints that were considered refer to the steam quality at the turbine outlet, the HRSG outlet exhaust gas temperature, and the steam turbine blade height. In their work, a parametric analysis was also performed to evaluate the gas temperature's influence at the HRSG inlet and the pinch point temperature difference on the considered objective functions.

Çoplan (2011) [57] applied a conventional exergy and thermoeconomic analysis to a combined-cycle cogeneration plant, using the SPECO method and CGAM problem. De Sa and Al Zubaidy investigated the effect of ambient temperatures on the energetic performance of a gas turbine. They found that for every degree Kelvin rise in compressor inlet temperature, there was a 0.07% drop in combustion turbine (CT) thermal efficiency.

Kaushik et al. (2011) [58] conducted an in-depth review of the energy and conventional exergy analysis as applied to coal and gas power plants, noting the method of splitting the exergy destruction into endogenous and exogenous components as a potential improvement.

Acir et al. (2012) [59] looked at the effect of varying dead state temperatures on energetic and exergetic efficiencies in a thermal power plant. They found the exergy efficiency to decrease by ~8% for an ~8% increase in dead state temperature (278K to 303K).

Kaviri et al. (2012) [60] show that an increase in the compressor pressure ratio decreases the cost of exergy destruction. The reason is that by increasing the compressor ratio, the outlet temperature increases as well. Therefore, the temperature difference decreases. Because the cost of exergy destruction is a direct function of exergy destruction, it leads to a decrease in the cost of exergy destruction. As the compression ratio increases, the air exiting the compressors is hotter; therefore, less fuel is required (lowering the air-fuel ratio) to reach the desired turbine inlet temperature in a fixed gas flow to the gas turbine. The work required in the compressor and the gas turbine's power output steadily increases with compression ratio and then causes decreases in the exhaust gases temperature. This lower gas temperature causes less steam to be produced in the HRSG, therefore lowering the steam cycle's outputs. It is noticed that the total power output increases with the compression ratio. However, the total power output variation is minor at the lower compression ratio, while it is significant at the higher compression ratio for all gas turbine configurations.

2.3 Advanced Exergy Analysis

Petrakopoulou et al. (2012) [61] used conventional and advanced exergy methods to conduct an environmental evaluation of a power plant. They found that as result of being the largest source of exergy destruction, the combustion chamber has the most environmental impact, and 68% of the exergy destruction is unavoidable. Petrakopoulou et al. applied conventional and advanced exergy methods to a combined cycle power plant and found the combustion chamber to be the largest source of irreversibility in the system. They determined the combustion chamber should be improved first, followed by the expander and air compressor.

Wang et al. (2012) [62] applied conventional and advanced exergy methods to a supercritical power plant. From their results of splitting the exergy destruction, they suggested the generator and steam turbines should be the first components to be improved.

Wadhah Hussein Abdul Razzaq Al- Doori (2012) [63] presented the exergy analysis of a Baiji plant with a gas-turbine of capacity 159-MW with effect cycle temperatures. Each component of the system is tested by the laws of mass and energy conversion. The aspects under consideration were the quantitative exergy balance for the entire system and each component, respectively. At different temperatures, the rate of the irreversibility of system components, efficiency of exergy, and efficiency flaws were highlighted for each component and the whole plant. The exergy flow of material is classified into thermal, mechanical, and chemical exergy in this study and a stream of entropy-production. Fuel oil of a low heating value of 42.9 MJ/kg was used as the fuel. The rate of exergy destruction in the turbine was around 5.4%, whereas that in the combustion chamber was about 36.4%. When a 14°C rise in the temperature, then the combustion chamber's exergy efficiency is 45.43%, and the turbine is 68.4%. According to the study results, the combustion chamber and turbine are the chief means of irreversibilities in the plant. It was also identified that the exergetic efficiency and the exergy destruction are considerably dependent on the alterations in the turbine inlet temperature.

V. Tara Chand et al. (2013) [64] presented the gas turbine power plant's exergy analysis. The exergy analysis conducts the mass, energy, and exergy balance of each component in the gas turbine plant. Parametric analysis of the influence of various factors, namely compression ratio, compressor inlet air temperature (AT), and turbine inlet temperature (TIT), on each component of gas turbine plant's irreversibility out. Due to the combustion chamber and gas turbine plant's irreversibilities finely increases and then decreases with pressure ratio. Compressor inlet temperature has an insignificant effect on gas turbine plant components. Irreversibilities of combustion chamber and gas turbine plant decreases with turbine inlet temperature increases.

Mousafarash and Ameri's (2013) [65] study consists of exergy analysis of a typical GT power plant, analysis of system performance at different ambient temperatures and partial loads, and exergo-economic analysis of the gas turbine power plant. Their study results reveal that the highest exergy destruction occurs in the combustion chamber, where the large temperature difference is the primary source of irreversibility. The effects of the gas turbine load variations and ambient temperature were investigated to see how system performance changes: the gas turbine was significantly affected by the ambient temperature, which led to a decrease in net power output. The load variation of the gas turbine showed that a reduction in gas turbine load resulted in a decrease in the cycle's exergy efficiency and all the components. They conducted an exergo-economic analysis to determine the cost of exergy destruction in each component and determine fuel cost. The results show that the combustion chamber has the highest cost of exergy destruction.

Morosuk and Tsatsaronis (2013) [66] outlined the strengths and limitations of the advanced exergy analysis. They presented three important questions that should be answered in order to thermodynamically improve an energy conversion system:

1. What is the maximum possible decrease of the exergy destruction within each system component?
2. How will reducing the exergy destruction of the k-th component affect the exergy destruction of the other components in the system?
3. Are there any other ways to restructure the system so that the exergy destruction of the k-th component, or more importantly, within the overall system can be reduced?

These questions are not addressed in a conventional exergy analysis, however, they are addressed in certain advanced exergy analyses that split the exergy destruction term. The limitations of advanced exergy analysis include arbitrariness of some values, a need to simulate unique processes, and the requirement of many simulations.

Sayed A. Abdel-Moneim and Khaled M. Hossin (2013) [67] discussed the Exergy Analysis of a combined gas/ steam turbine cycle with a supercharged boiler. A combination of a primary gas turbine and steam cycle with both a supercharged boiler (SB) and a heat recovery boiler (HRB) investigated. The effects of the inlet temperature of the gas turbine, the excess air factor, and the compressor pressure ratio on the supercharged boiler combined cycle (SBCC) performance were performed. The results indicated that the SBCC gives output power up to 2.1 times that of the conventional combined cycle compared to the same values of the operating parameters. However, the SBCC efficiency was found to be lower than the conventional combined cycle. The exergy analysis showed an advantage of SBCC over the conventional combined cycle. The largest values of the output power for the SBCC are predicted at a minimum excess air factor and a maximum turbine inlet temperature. For a turbine inlet temperature of 1300oC, optimum compressor pressure ratios that give maximum efficiencies are predicted for the SBCC. While, for the conventional cycle, the efficiency is continuously increased with the compressor pressure ratio. Higher values for the second-law efficiency were found for SBCC compared with that for the conventional combined cycle. An enhancement ranging from 9.5% to 18.5% in the second-law efficiency for SBCC was found compared with that for a conventional cycle.

Padma Dhar Garg et al. (2013) [68] presented the Exergy and Efficiency Analysis of Combined Cycle Power Plant of NTPC (National Thermal Power Corporation) Dadri India. The performance of a combined cycle cogeneration configuration based on energy and exergy analysis approaches. Here demonstrated that a combined cycle cogeneration unit operates more efficiently and produces less carbon dioxide than two separate power production and process heat systems. The exergy analysis identifies the sources of irreversibility in the system and aids in evaluating losses and outputs by examining their quality. The results pinpoint that more exergy losses occurred in the gas turbine combustion chamber reaching 35% of the total exergy losses, while the exergy losses in the other plant components are between 7% and 21% of the total exergy losses at 1400°C turbine inlet temperature and pressure ratio 10. The first law efficiency is around 45%, and that associated with second law is 24%. NTPC Dadri can use 45% of its energy source to convert to real work. System losses and irreversibility consume the rest. Heat recovery steam generators are the primary equipment for a combined cycle power plant. More exergy losses occur in the combustion chamber due to combustion irreversibility and this must be reduced with the aid of new advances in the technology. At higher turbine inlet temperature and lower pressure ratio, the combustion chamber's exergy losses, heat recovery steam generator, and the steam turbine are higher. At lower turbine inlet temperature and higher pressure ratio the gas turbine's exergy losses are higher.

G. MEMON et al. (2013) [69] discussed the exergy analysis 144 MW combined cycle power plant kotri Pakistan. This research states that the exergy analysis is a more useful thermodynamic tool than energy analysis for the power plant's performance assessment. The exergy destruction models are used to assess the losses that occurred in the power plant's main components. The results indicate that the total exergy destruction of the power plant is around 288.5 MW; combustion chambers contribute a significant share of 168 MW (58.2%), followed by heat recovery steam generators (HRSGs) with 43 M.W. (14.8%), gas turbines (10%), air compressors (5%), and stack gas (4%). The power plant's energy efficiency and exergy efficiency are calculated as 34.41 and 33.40%, respectively. Arrangements proposed to reduce the exergy destruction rates in combustion chambers and HRSGs are higher gas turbine inlet temperature and optimizing the HRSGs operating parameters.

Mukesh Gupta and Raj Kumar (2015) [70] presented the Exergy Based Analysis of an Open Cycle Gas Turbine Power Plant, and a 25 MW open cycle GTPP has considered. Exergy destruction and the effect of thermodynamic variables on the exergy destruction in various components have been analyzed. Finally, equations have developed, which correlate the exergy destruction in different components as a function of the thermodynamic variables under consideration. The combustion chamber has the maximum exergy destruction, followed by the gas turbine and compressor. The combustion chamber is the least efficient among the three components from an exergetic viewpoint. Further, the effect of two thermodynamic variables, (1) Compressor pressure ratio and (2) Inlet air temperature, on the exergy destruction in different components have been studied in detail. The maximum effect of variation in compressor pressure ratio is felt in the gas turbine and is felt least in the compressor. The maximum effect of inlet air temperature felt in the combustion chamber and least in the gas turbine.

Siddig Abuelgasim Abbas Mohieldein (2016) [71] presented the Exergy Analysis of Combined-cycle Power Plant (GARRI"2" 180 MW). Exergy analysis has been carried out analytically for Garri"2" to evaluate exergetic efficiency and exergy destruction of each part; exergy balance and entropy generation calculated to achieve it. The results show that the combustion chamber's exergy efficiency is much lower than the efficiency of other combined cycle components due to its high irreversibility; it represents 63% of destruction for the whole Garri"2". The second significant exergy losses are in the gas and steam turbines, respectively. Optimization of these equipment has an essential role in reducing the exergy losses of the total combined cycle. This plant achieved thermal and exergetic efficiencies (38%, 49%) respectively; that means 51% from entered exergy destroyed due to irreversibilities and escape exergy to an ambient and cooling tower.

B.V. Reddy and K. Mohamed (2016) [72] discussed the exergy analysis of natural gas-fired combined cycle power generation unit to investigate the effect of gas turbine inlet temperature and pressure ratio on exergetic efficiency for the plant and exergy destruction/losses for the components. For a fixed gas turbine inlet temperature, an optimum pressure ratio exists where the exergy destruction is minimum. The exergy analysis provides the details on exergetic efficiency, exergy loss, and destruction for individual components in the plant and the overall plant and their variation with operating parameters. The gas turbine main component combustion chamber (CC1) is the primary source of exergy destruction rate. In the main combustion chamber, the exergy destruction rate in the HRSG and steam cycle components (SET2) drops with increasing pressure ratio. Simultaneously, the higher pressure ratio results in an increase in the exergy destruction rate in the gas turbine cycle components and the reheat combustion chamber (CC2). The combined cycle network output increases with higher gas turbine inlet temperatures for the same pressure ratio. The exergy destruction rate in the combustion chambers and the gas turbine cycle components (SET1) reduces with higher gas turbine inlet temperature. For the bottoming steam cycle components (SET2) and the HRSG, the increase in GTTI results in a higher rate of exergy destruction/loss. Using the exergy destruction rate plots, the main combustion chamber's minimum exergy destruction rate (CC1) occurs at the highest pressure ratio and gas turbine inlet temperature. In the reheat combustion chamber (CC2) case, the exergy destruction rate drops with lowering the pressure ratio and increasing the gas turbine inlet temperature. The exergy destruction rate in the main combustion chamber (CC1) is significant, and setting the conditions to lower the exergy destruction rate in the main combustion chamber, results in an overall drop in the combustion chambers total exergy destruction rate. In the HRSG, the maximum total exergy destruction rate and losses occur at the lowest pressure and highest gas turbine inlet temperature. At low gas turbine inlet temperatures (<900°C), they increase the pressure ratio, resulting in a significant increase in the combined cycle thermal efficiency, with higher supplementary firing fuel rates.

Divya Prakash and Onkar Singh (2020) [73] presented the Exergy analysis of a combined cycle power plant with carbon capture utilization. This paper considers retrofitting an existing combined cycle power plant arrangement to get carbon dioxide from gas turbine exhaust and convert it into methane to run additional gas turbine for power augmentation. Here exergy analysis of the considered combined cycle estimates the exergy efficiency and exergy destruction of significant components. The study shows the comparative results for different power plants with and without carbon capture arrangement. The highest exergy destruction occurs in the combustion chamber at 12.7 cycle pressure ratio, 1400 K turbine inlet temperature while it is minimum at 8.7 cycle pressure and 1475 K turbine inlet temperatures. The highest exergy efficiency of 96.66% was obtained for the gas turbine at 12.7 cycle pressure and 1475 K turbine inlet temperatures. These exergy estimations help in knowing the relative performance of different components of the cycle under consideration, which can be used for the future development of such systems.

2.4 Exergy Economics

According to Sciubba, the earliest idea of combining thermodynamics and economics was by Lotka in 1921, and the application of exergy analysis and engineering economics was proposed in the early 1960's. In Europe it was called exergo-economics (Rabek 1964, Szargut & Petela 1964, Baehr et al. 1965, Brodyanski 1965, Fratzscher 1965, Elsner 1965, Nitsch 1965, Bergmann & Schmidt 1967) and in the US it was called thermos-economics (Evans 1961, Tribus 1961, Tribus & Evans 1962, Evans & Tribus 1965, El-Sayed 1970).

With respect to the modern exergoeconomic analysis, Tsatsaronis, Lazzaretto, Diner, and Rosen are some of the main researchers. In 1984, Tsatsaronis first defined the term exergoeconomics as a more specific term for an exergy based thermoeconomic analysis. In 1985, Tsatsaronis and Winhold described a new method of exergy based thermoeconomic analysis, or exergoeconomic analysis, for power plants. This method is broken down into seven steps:

1. Conduct detailed mass, energy, and exergy balances of the plant.
2. Calculate the investment and operating costs for each plant component.
3. Calculate the cost of the exergy unit of each process flow stream.
4. Calculate the average exergy unit cost of fuel and products of each component.
5. Calculate the cost of the exergy losses in each component.
6. Interpret the results.
7. Conduct a sensitivity analysis and make recommendations.

This method of exergoeconomic analysis is useful for illustrating the sources of costs, providing a means of comparison between them, and calculating the optimum capital cost to exergy loss ratio for a given design. This method can be used to improve decisions concerning selection and optimization of process design, plant maintenance, and replacement of certain plant components.

Valero et al. (1994) [74] presented the application of four methods of thermoeconomic analysis and optimization, called the CGAM problem, an acronym for the principle researchers' first names. The CGAM problem consists of a physical model, thermodynamic model, and economic model, applied to a small cogeneration plant.

Decision variables are chosen to be the pressure ratio, the air compressor efficiency, the gas turbine efficiency, the air temperature at the preheater exit, and the combustion gas temperatures at the turbine inlet. The aim of the CGAM problem is to unify the four thermoeconomic methodologies as each has specific fields of applications for which it provides proven and efficient solutions.

Kim et al. (1998) [75] introduced a modified productive structure analysis (MOPSA) method, where an exergy costing method is used without flow-stream cost calculations. For the entire system, a set of equations for the unit exergy costs obtained by assigning a unit exergy cost for each component's cost balance equation.

Kwon et al. (2001) [76] compared specific exergy cost methods and modified productive structure analysis methods by applying them to the CGAM problem.

Valdes et al. (2003) [77] showed a possible way to achieve a thermoeconomic optimization of combined cycle gas turbine power plants. The optimization was done using a genetic algorithm, tuned by applying it to a single pressure CCGT power plant. Once tuned, the optimization algorithm was used to evaluate more complex plants, with two and three pressure levels in the heat recovery steam generator. The variables considered for the optimization were the thermodynamic parameters that established the configuration of the HRSG. Two different objective functions were proposed: one minimizes the production cost per unit of output, and the other maximizes the annual cash flow. The results obtained with both functions were compared in order to find a better optimization strategy. The results show that it is possible to find an optimum for each design parameter. This optimum depends on the selected optimization strategy.

Casarosa et al. (2004) [78] minimized the total cost of the exergy losses of the HRSG for a combined cycle using the Simplex method. The objective function was the total installed cost of the HRSG and reduced the cost of the increased fuel consumption when the area of the HRSG.

Lazzaretto and Tsatsaronis (2006) [79] proposed defining and calculating exergetic efficiencies and exergy related costs in thermal systems. It was based on the SPECO. Separate forms of exergy and costs associated with these exergy streams were used to define exergetic efficiencies in a detailed manner. It was concluded that the SPECO was a powerful approach to express the validation of the calculated cost values.

Cziesla et al. (2006) [80] determined the avoidable thermodynamic inefficiencies and costs in an externally fired combined cycle power plant. They found that the largest avoidable cost of exergy destruction occurs in the combustion chamber, but that the component had a low potential for improvement.

Kelly (2008) [81] used the advanced exergy method to improve upon existing exergoeconomic analysis methods. Most notably, by calculating cost rates from the avoidable endogenous and exogenous exergy destruction rates of the major component, a realistically attainable potential savings can be found, as well as the source of the destruction. By having the avoidable costs associated with each component, and knowing whether they are due to the plant configuration or inefficiencies in the component itself, a more informed decision can be made with regard to system improvements.

Orhan and Dincer (2010) [82] studied the minimization cost of a copper–chlorine (Cu- Cl) thermo-chemical cycle for hydrogen production. The specific exergy costing method was used to determine changes in the cycle's design parameters, improving the overall system's cost-effectiveness. It was found that the cost rate of the exergy destruction took the values between \$1 and \$15 per kg hydrogen. The exergoeconomic factors were calculated between 0.5 and 0.02.

Behbahani-nia et al. (2010) [83] presented an exergy-based thermoeconomic method applied to find the optimum values of design parameters for a single pressure HRSG in combined cycle power plants. The design variables optimized in this work were pinch point and gas side velocity. Optimization was performed by being based on two different objective functions. The first function was the thermodynamic(the summation of exergy loss due to an outflow of hot gas escaping from the HRSG through stack and exergy destruction due to internal irreversibility inside the HRSG). The second function was a thermoeconomic objective function (the summation of exergy loss and destruction in terms of expenses, including the cost of fuel and electricity and the capital cost of HRSG). They investigated the effects of pinch point and gas-side velocity on the components of objective functions. The study concluded that a considerable amount of exergy is destroyed due to gas pressure drop, especially when the pinch point is very close to zero.

Ahmadi and Dincer (2011) [84] performed the thermodynamic analysis and thermoeconomic optimization of a dual pressure combined cycle power plant with a supplemental firing unit. They conducted exergy and exergoeconomic analyses for the power plant. The design parameters of this study were compressor pressure ratio, isentropic compressor efficiency, gas turbine isentropic efficiency, gas turbine inlet temperature, duct burner mass flow rate, high-pressure stream, low-pressure stream, high-pressure central steam temperature, low-pressure steam temperature, high-pressure pinch point temperature difference, low-pressure pinch point temperature difference, condenser pressure, steam turbine isentropic efficiency, and pump isentropic efficiency. They introduced an objective function, a new objective function, representing the plant's total cost (in terms of dollar per second), defined as the sum of the operating cost related to the fuel consumption and the capital investment for equipment purchase and maintenance costs. The optimum key variables were obtained by minimizing the objective function using a genetic algorithm. The optimum design parameters obtained for the plant showed a trade-off between the thermodynamic and economic optimal designs. Two factors were considered: the unit cost of fuel and the combined cycle power plant's net output power. They concluded that by increasing the fuel price, the optimized decision variables in the thermoeconomic design tends to reach those of the optimum thermodynamic design.

Ahmadi et al. (2011) [85] performed a comprehensive conventional exergy, exergoeconomic, and environmental impact analysis comparing several combined cycle power plants (CCPP). They found the combustion chamber to have the most significant exergy destruction and cost, and the amount of supplementary firing to be proportional to the CCPP exergy efficiency.

Kanoglu et al. (2011) [86] developed a methodology for calculating exergy flows, cost formation, and allocation within the high-temperature steam electrolysis system. They used specific exergy costing methodology while applying exergetic fuel and product approaches to obtain the cost balance equations. They examined exergy efficiency, exergy destruction rates, exergy loss–exergy destruction ratio, capital investment, operating, maintenance costs, and exergoeconomic factor. The capital investment cost, operating and maintenance costs, and the system's total cost were calculated as 422.2, 2.04, and 424.3 €/kWh, respectively. The cost distribution among the components was also determined. The exergetic costs of the steam were 0.000509, 0.000544, and 0.000574 €/kWh at the outdoor temperatures of 25 °C, 11°C, and −1°C, respectively.

Alus and Petrović (2012) [87] performed the optimization of a triple pressure CCGT. Their work's objective was to develop a new system for optimization of parameters for CCGT with a triple-pressure heat recovery steam generator. The thermodynamic optimization objective is to enhance the efficiency of the CCGT and maximize the power production in the steam cycle (steam turbine gross power). Improvement of the CCGT plants' efficiency was achieved through optimization of the operating parameters: the temperature difference between the gas and steam pinch point (PP) and the steam pressure in the HRSG. The thermoeconomic optimization aimed to minimize the production costs per unit of the generated electricity; optimization was to minimize the production cost of electricity in the CCGT power plant based on energetic and economic analysis.

Ghazi et al. (2012) [88] carried out a thermo-economic modeling and optimization method to obtain the optimum values of design parameters (high and low drum pressures, steam mass flow rates, high pressure and low-pressure pinch point temperature differences, and the duct burner fuel consumption flow rate) for a dual pressure HRSG. They performed the complete sensitivity analysis of changes in inlet gas temperature entering the HRSG and exergy unit cost. Total cost per unit of produced steam exergy was defined as the objective function. They found that the required heat transfer surface area (capital cost) increases at higher inlet gas enthalpy.

Naomi et al. (2013) [89] developed the thermodynamic model of a dual pressure HRSG coupled with a heavy-duty gas turbine. They investigated thermodynamic and thermoeconomic analyses to achieve the optimum operating parameters of a dual pressure heat recovery steam generator and computed exergy waste and exergy destruction for different pinch points. They discussed the effects of non-dimensional parameters on the HRSG performance. They also investigated optimum design of HRSG regarding financial considerations, and performed a sensitivity analysis.

Dincer and Rosen (2013) [90] presented the necessary equations for conducting a conventional exergy analysis of several systems, including combined cycle. They also explained two methods of exergoeconomic analysis: exergy, cost, energy, and mass analysis (EXCEM) and specific exergy cost analysis (SPECO).

D. I. Igbong and D.O. Fakorede (2014) [91] presented the exergoeconomic analysis of a 100 MW unit GE frame-9 gas turbine plant in ughelli, Nigeria**.** These optimizations were performed on engineering equation solver (EES) software to estimate the cost rate associated with all the exergy streams at cycle state points and the cost of the plant's final product, which is electricity. Two parameters were chosen as decision variables: turbine inlet temperature and compressor pressure ratio. The product's unit cost decreases to a minimum point as the decision variables increase, beyond which it increases with further increase in the parameters' value. The effect of the turbine inlet temperature and the pressure ratio on the product's unit cost was investigated. The result shows that the unit cost of production decreases to a minimum point as the decision variables increase, increasing with further increase in the parameters' value. The plant thermal efficiency is 31.05%, and overall exergy efficiency of 30.81%, identified the combustion chamber with the lowest exergetic efficiency of 54.05%, accounting for the component with the most immense total inlet exergy destruction value of 238.681 MW.

Oyedepo et al., Cogent Engineering (2015) [92] discussed the exergy analysis and exergy costing analysis and performance evaluation of selected gas turbine power plants in Nigeria. Exergy costing analysis and performance assessment from a thermodynamics point of view performed for 11 selected gas turbine power plants in Nigeria. The exergy analysis results show that the combustion chamber is the most significant exergy destructor in the selected power plants, which is due to the chemical reaction and the large temperature difference between the burners and working fluid. Moreover, the results show that an increase in GTIT leads to an increase in gas turbine exergy efficiency due to a rise in the turbine's output power and a decrease in the combustion chamber losses. Besides, by increasing GTIT, the gas turbine cost of exergy destruction can be decreased. The finding solidifies the concept that the combustion chamber's exergy loss is associated with the large temperature difference between the flame and the working fluid. Reducing this temperature difference reduces the exergy loss. Furthermore, cooling compressor inlet air allows the compression of more air per cycle, effectively increasing the gas turbine capacity. This study found that an increase in the GTIT of about 200 K could lead to a reduction of about 29% in the cost of exergy destruction.

Therefore, GTIT is the best option to improve cycle losses. From exergy costing analysis, the unit cost of electricity produced in the selected power plants varies from cents 1.99/kWh (N3.16/kWh) to cents 5.65/kWh (N8.98/kWh).

Shoaib Khan Mohammadi and Ahmad Reza Azimian (2015) [93] This paper deals with exergy and exergoeconomic analysis of a combined-cycle power plant with supplementary firing. A process simulation program, IPSE Pro, is used to model the combined-cycle power plant. Exergy and exergoeconomic analysis is performed by developing a Matlab code. Three configurations of the combined-cycle power plant are investigated. The effect of the configurations of a heat recovery steam generator (HRSG) and performance parameters such as fuel mass flow rate of duct burner and pressure ratio of the compressor on combined-cycle performance have been studied. Finally, the cost of generated power for the three configurations of the combined-cycle power plant is the duct burner in HRSG.

Karaali and Oztürk (2015) [94] introduced a simple method of thermoeconomic optimization and applied it to four cogeneration cycles with constant power output. They found that there is an optimum excess air flow rate for each cycle that gives the minimum electricity cost.

Edgar Vicente Torres Gonzalez et al. (2016) [95] presented an exergoeconomic analysis of the combined cycle power plant of three levels of pressure Tuxpan II located in Mexico. The plant comprises two identical modules confirmed by two gas turbines generating the required work and releasing the hot exhaust gases in two heat recovery steam generators. These components generate steam at three different pressure levels and produce additional work in one steam turbine. The productive structure of the considered system is used to visualize the cost formation process and the productive interaction between their components. The exergoeconomic analysis is pursued by 1) carrying out a systematic approach, based on the Fuel-Product methodology, in each component of the system; and 2) generating a set of equations, which allows computing the exergetic and exergoeconomic costs of each flow. The two gas turbines' thermal and exergetic efficiency delivering 278.4 MW is 35.16% and 41.90%, respectively. The computed thermal efficiency of the steam cycle, providing 80.96 MW, is 43.79%. The combined cycle power plant generates 359.36 MW with a thermal and exergetic efficiency of 47.27% and 54.10%, respectively. Moreover, there is no improvement in the gas turbine's performance when the current operating conditions are changed to the conditions of maximum output work because the costs of exergy destruction of the gas turbine increase by 2.20%, and its exergetic efficiency decreases by 0.49%.

Paul Otujevwe Ogbe et al. (2016) [96] presented the exergoeconomic evaluation of transcorp power plant ughelli. In this study, an exergoeconomic evaluation of two 100MW gas turbine units at Transcorp power plant Ughelli, Nigeria, to determine the location and magnitude of exergy destruction and the cost of exergy destruction associated with each component for the two GE frame-9 engines (GT16 and GT19 units) in the plant. The results obtained from exergoeconmic analysis show that the combustion chamber has the highest energy destruction cost of 2351.81$/h GT16, 2315.93$/h GT19 compared to the turbine with 277.36$/h GT16 and 274.46$/h GT19, also compressor with 556.31$/h GT16, 547.60$/h GT19. It shows the high level of irreversibility and degradation in the combustion chamber. The exergoeconomic analysis results from two plant units show that the combustion chamber has the highest exergy destruction compared to other components. The cost of exergy destruction of turbine and compressor is lower than the combustion chamber with the highest cost of exergy destruction. The exergonomic factor of turbine and compressor is far better than the combustion chamber with the lowest exergoeconomic factor's lowest value. To reduce the exergy destruction cost of the combustion chamber and efficiency is by increasing the capital investment costs, which will increase the exergoeconomic factor.

Awaludin Martin et al. (2016) [97] presented the exergy analysis of a 20 MW gas turbine power plant Pekanbaru-Indonesia. The first and second thermodynamics laws and the mass and energy conservation law are applied in each component. The results show that more exergy destruction occurred in the combustion chamber up to 71.03% or 21.98 MW. Meanwhile, the lowest exergy occurred in the compressor at 12.33% or 3.15 MW. According to the first law, the thermal efficiency of the gas turbine power plant was 33.77%, and exergy efficiency was 32.25%. The irreversibility occurred since there are large temperature differences between the combustion chamber and the working fluid.

M I Riady, D Santoso, and M D Bustan (2018) [98] presented the thermodynamic and thermoeconomic performance evaluation in 326 MW combined cycle power plant at Tanjung Priok using combined pinch and exergy analysis. The exergy analysis results on the gas turbine cycle indicated that the combustion chamber had the highest irreversibility and the exergy destruction on the combustion chamber is 191.55 MW. Combined cycle net power is directly dependent on the heat recovery potential in the HRSG. Pinch and exergy analysis was used to improve the combined cycle's overall exergetic efficiency by decreasing the temperature difference in the HRSG. Besides, the HP and LP drum pressure also had a significant impact on the power plant's performance, and it was evident by the increase of the whole plant exergetic efficiency as a result of optimization on the drum pressure. Overall, the exergetic efficiency and the network produced by the CCPP were increased by 1.23 percent and 2.67 percent, respectively. This result also shows that the investment for the HRSG stays constant for a given gas turbine system while the net power output for the combined cycle power plant increases with increasing the heat exchanger effectiveness. Thermoeconomic analysis generally involves more uncertainties than a thermodynamic analysis.

2.5 Conclusion

Energy auditing, exergy analysis, and exergoeconomic analysis of a plant are done previously in many countries in the world. As a developing country, Bangladesh, only energy auditing is done to evaluate the performance of a power plant previously. It is also used to improve efficiency. However, only energy auditing can improve a plant's performance properly without using the exergy and exergoeconomic tools. Natural gas is the primary fuel to produce energy by both using the turbine machine and reciprocating engine. The turbine engine is only 21% of the whole power generation. The reciprocating engine is 39% of total power generation, and it is possible to improve the plant efficiency by doing exergy and exergoeconomic analysis and energy auditing.

Exergy measures quantity and quality. Whereas energy a measure of quantity only. Exergy destruction is a measure of the work potential lost due to irreversibilities in the system. Various exergy, or exergetic, analysis methods provide engineers a way to determine the location, magnitude, and sources of irreversibilities. Another thing is that, for the analysis, design, and improvement of energy systems, exergy analysis is a method that uses the conservation of mass and conservation of energy principles together with the second law of thermodynamics. Exergy analysis is based on both the first and second laws of thermodynamics. Exergy analysis can indicate the location of energy degradation in a process. An exergy method is a useful tool for furthering the goal of more efficient energy resource use, for it enables the locations, types, and magnitudes of wastes and losses to be determined. The primary purpose of exergy analysis is to identify the causes, types, and locations and calculate the magnitude of thermal losses.

With this information, plant layout can improve, capital and operational costs can decrease, and more efficient operating schedules can establish. So, it can be easy to say that if exergy analysis is happening in our country and energy analysis of a power plant, we can improve our plant efficiency and reduce the overall costing of the power plant.

Chapter 3: BACKGROUND

3.1 Combined Cycle Power Plant

A combined-cycle power plant uses both gas turbine and steam turbine to produce up to 50 percent more electricity from the same fuel than simple-cycle plant. Gas turbine's waste heat is routed to the nearby steam turbine, which generates extra power.

Combined Cycle's idea has grown out of the need to improve the simple Brayton –Cycle efficiency by utilizing the waste heat in the turbine exhaust gases. The large quantity of energy leaving with the turbine exhaust generates steam for a steam turbine power plant. The gas turbine is a relatively high-temperature machine (2000 to 3000°F, 1100 to 1650°C), whereas the steam turbine is a relatively low-temperature machine (1000 to 1200°F, 540 to 650°C). This joint operation of the gas turbine at the "hot end" and the steam turbine at the "clod end" is called a combined cycle power plant.

Besides high efficiency and high power outputs, combined cycles characterize flexibility, quick part-load starting, suitability for both base-load and cyclic operation, and high efficiency over a wide range of loads.

It was not, however, until 1950 that the first plant was installed. This was followed by a rapid rise in the number of installations, especially in the 1970s. Estimated 100 plants, with a total of 1,50,000 MW output, had been installed by the end of the 1970s throughout the world.

In the years since, power generation has played a vital role in everyday life. With rising energy costs, increased plant efficiency has become increasingly necessary and desirable. Even with the most efficient components, simple cycle power plants are still wasting about half of their energy.

For many years, engineers have researched many ways to reduce wasted energy and harness the most work potential possible. Of these methods, cogeneration and combined cycle power plants have emerged as two of the top contenders.

Combined cycle power plants utilize waste heat from a prime mover to produce steam to power a steam turbine to generate electricity. In the case of Combined cycle power plants, the prime mover was a combustion gas turbine.

A typical simple-cycle gas turbine will convert 30% to 40% of the fuel input into shaft output. All but 1% to 2% of the remainder is in the form of exhaust heat. The combined cycle is generally defined as one or more gas turbines with heat-recovery steam generators in the exhaust, producing steam for a steam turbine generator, heat-to-process, or a combination.

The figure shows a combined cycle in its simplest form. High utilization of the fuel input to the gas turbine can be achieved with some of the more complex heat-recovery cycles, involving multiple-pressure boilers, extraction or topping steam turbines, and avoidance of steam flow to a condenser to preserve the latent heat content. Attaining more than 80% utilization of the fuel input by combining electrical power generation and process heat is not unusual.

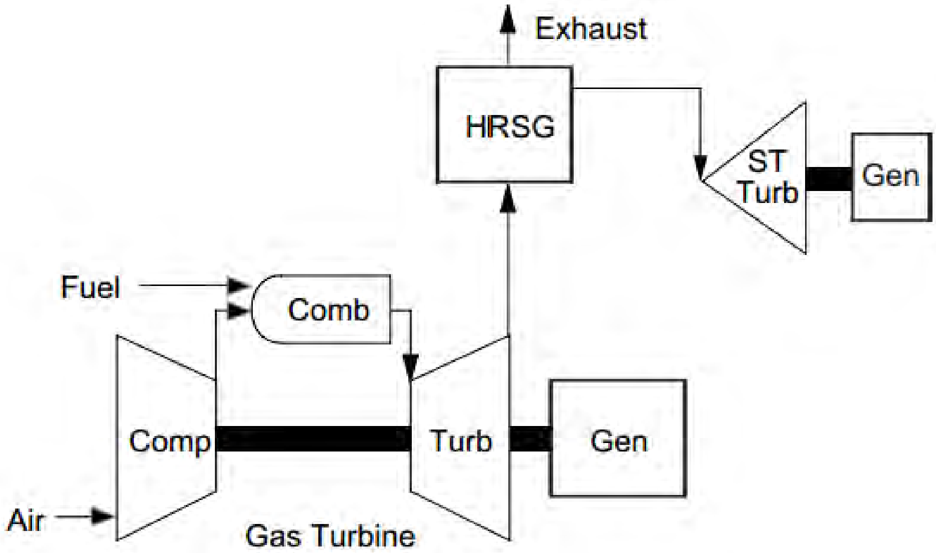

Figure 3.1: Flow diagram of a combined cycle power plant.

Combined cycles producing only electrical power are in the 50% to 60% thermal efficiency range using the more advanced gas turbines.

A combined-cycle power plant is more efficient than a conventional power plant because it uses a higher proportion of the fuel's energy when it burns. Increasing the overall efficiency of power plants, recover and utilize the residual heat energy in hot exhaust gases. In combined cycle mode, power plants can achieve electrical efficiencies up to 60 percent. The term "combined cycle" refers to the combining of multiple thermodynamic cycles to generate power. Combined cycle operation employs a heat recovery steam generator (HRSG) that captures heat from high-temperature exhaust gases to produce steam, which is then supplied to a steam turbine to generate additional electric power. The process for creating steam to produce work using a steam turbine is based on the Rankine cycle.

The most common type of combined cycle power plant utilizes gas turbines and is called a combined cycle gas turbine (CCGT) plant. Because gas turbines have low efficiency in simple cycle operation, the steam turbine output accounts for about half of the CCGT plant output. There are many different configurations for CCGT power plants, but each GT has its own associated HRSG, and multiple HRSGs supply steam to one or more steam turbines. For example, at a plant in a 2x1 configuration, two GT/HRSG trains supply one steam turbine; likewise, there can be 1x1, 3x1, or 4x1 arrangements. The steam turbine is sized to the number and capacity of supplying GTs/HRSGs.

3.2 Working Principle of Combined Cycle Power Plant

The HRSG is a heat exchanger, or rather a series of heat exchangers. It is also called a boiler, as it creates steam for the steam turbine bypassing the hot exhaust gas flow from a gas turbine or combustion engine through banks of heat exchanger tubes. The HRSG can rely on natural circulation or utilize forced circulation using pumps. As the hot exhaust gases flow past the heat exchanger tubes in which hot water circulates, heat is absorbed, causing steam in the tubes. The tubes are arranged in sections or modules, each serving a different function in the production of dry superheated steam. These modules are referred to as economizers, evaporators, superheaters/reheaters, and preheaters.

The economizer is a heat exchanger that preheats the water to approach the saturation temperature (boiling point), supplied to a thick-walled steam drum. The drum is located adjacent to finned evaporator tubes that circulate heated water. As the hot exhaust gases flow past the evaporator tubes, heat is absorbed, causing steam in the tubes. The steam-water mixture in the tubes enters the steam drum, where steam is separated from the hot water using moisture separators and cyclones—the separated water recirculated to the evaporator tubes. Steam drums also serve storage and water treatment functions. An alternative design to steam drums is a once-through HRSG, which replaces the steam drum with thin-walled components that are better suited to handle changes in exhaust gas temperatures and steam pressures during frequent starts and stops. In some designs, duct burners add heat to the exhaust gas stream and boost steam production; they can produce steam even if there is insufficient exhaust gas flow.

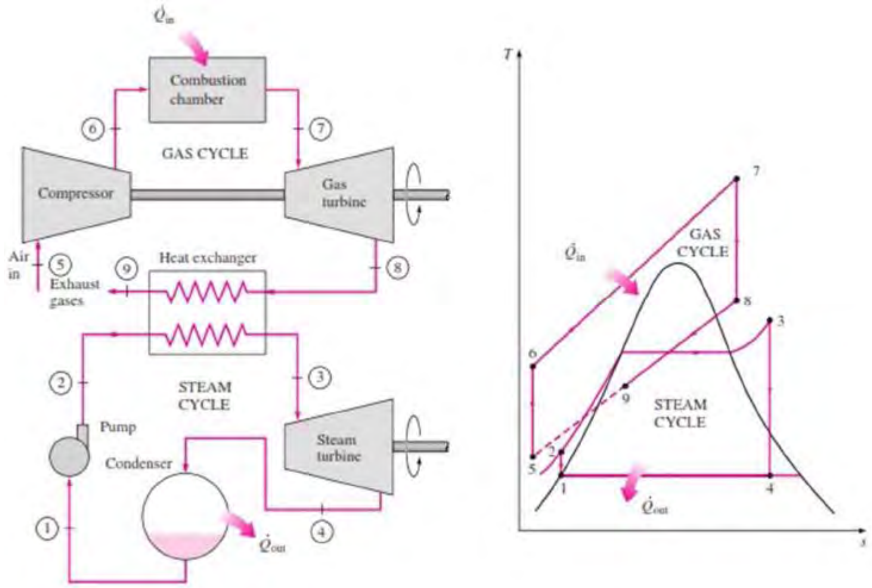

Figure 3.2: Schematic & T-S Diagram of Combined Cycle Power Plant (from Moran Sopherie) [99].

Saturated steam from the steam drums or once-through system is sent to the superheater to produce dry steam required for the steam turbine. Preheaters are located at the coolest end of the HRSG gas path and absorb energy to preheat heat exchanger liquids, such as water/glycol mixtures, thus extracting the most economically viable amount of heat from exhaust gases.

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

3.3 Description of the Fenchuganj Combined Cycle Power Plant

In this thesis, energy, exergy, and exergoeconomic analysis of the Fenchuganj Combined Cycle Power Plant has been carried out. This plant is located in Fenchuganj, Sylhet Division, Bangladesh. The plant work was started on 23.02.2006. The total area of the plant is 38.05

acres. Another essential and significant thing is that the plant is installed near the Kushiara river. As we know that, in the Combined cycle plant, water supply is one of the vital components of this type of plant.

The facility of the Fenchuganj Combined Cycle Power Plant means the Gas-fired combined cycle power station comprising of the gas turbine generator and steam turbine generator, capable of 45 MW of capacity (net, without supplemental firing, at Reference Site Conditions and Power Factor Adjustments) to be owned and constructed by the FCCPP within Fenchuganj Combined Cycle Power Plant (FCCPP) complex area, Fenchuganj, Sylhet, Bangladesh, whether completed or at any stage of its construction, including without limitation or regard to the level of development, all energy-producing equipment, and its auxiliary equipment and all transmission facilities, water intake and discharge facilities (if any), water treatment facilities, solid waste disposal facilities, fuel (Gas) receiving facilities and equipment on the FCCPP's side of the Point of Delivery, the Metering System, together with the residential facilities made available to individual employees of the FCCPP, the Contractor(s) and any subcontractors.

The Plant is a combined cycle power generating facility with a net capacity of 45 MW at Reference Site Conditions. It has 1 x 1 x 1 (1 GT x 1 HRSG x 1 ST) configuration together with all associated auxiliary equipment and systems including gas booster compressors (GBC), pumps, water treatment plant, cooling tower and circulating water system, unit step-up transformer(s), and high voltage (HV) underground cable for power evacuation.
Other plant buildings and structures outside of the main power block include

- water and waste-water treatment facility,
- solid waste facilities,
- river water intake structures (or alternative makeup water ground wells),
- cooling towers,
- control room,
- storeroom,
- maintenance facility,
- Electrical Interconnection Facilities,

- chemical and oil storage facility, and
- other associated buildings and structures that may be required for the balance of the facility.

The total project cost of this plant is approximately 717.23 Crore. The facility shall be designed for a life of 25 years. This plant is installed, developed, and operated by the Bangladesh Power Development Board since 2006. At present, a total of 25 employees work on this plant. The fuel is supplied by Jalalabad Gas Transmission and Distribution Company Ltd. Sylhet. This plant is one of the oldest combined cycle power plants in the public sector.

Figure 3.3: Administration building of the Fenchuganj Combined Cycle Power Plant.

3.3.1 Gas Turbine

A heavy-duty gas turbine unit is a mechanical power engine installed to drive a generator to supply an electrical network. Typically, a simple-cycle gas turbine will convert 30% to 40% of the fuel input into shaft output. All but 1% to 2% of the remainder is in the form of exhaust heat. The combined cycle is generally gas turbines with heat-recovery steam generators in the exhaust, producing steam for a steam turbine generator, heat-to-process, or a combination. The gas turbine is the prime mover of a Combined Cycle Power Plant.

High utilization of the fuel input to the gas turbine can be achieved with some of the more complex heat-recovery cycles, involving multiple-pressure boilers, extraction or topping steam turbines, and avoidance of steam flow to a condenser to preserve the latent heat content. Attaining more than 80% utilization of the fuel input by combining electrical power generation and process heat is not unusual. Combined cycles producing only electrical power are in the 50% to 60% thermal efficiency range using the more advanced gas turbines.

In Fenchuganj Combined Cycle Power plant, one gas turbine of a heavy-duty directly coupled to the generator equipped with 50 Hz hydrogen or air-cooled generators. The Specification of the Gas Turbine are:

- Name of the Manufacturer: GE, USA.
- Model: Frame 06
- Generation Capacity: 35 MW
- Generation Voltage: 11 kV
- Cooling System: Closed cycle cooling system.

Figure 3.4: Gas Turbine Unit

The gas turbine power engine includes an axial airflow compressor, a multi-chamber combustion system, and a three-stage turbine. The main components of the gas turbine are listed here below.

The axial airflow compressor is a 17 stages compressor with:

- Adjustable inlet guide vanes (IGV) to control the airflow during starting and loading sequences.
- Bleed valves to bypass part of the airflow for starting and shut down to escape from surging.

The combustion system comprises:

- Fuel nozzles fitted on the combustion chamber's cover.
- Ten combustion chambers where the fuel burns permanently from firing speed to full load.
- Ten crossfire tubes connecting the combustion chamber.
- Ten transition pieces downstream the combustion chamber connected to the first turbine stage nozzle.
- Two spark plugs for the fuel ignition.

The three stages turbine includes the first, second, and third stage nozzle and the second and third wheels.

The turbine and the axial flow compressor belong to the same shaft connected to:

- The auxiliary gearbox and the starting mean at the front end.
- The generator at the rear end through a load gearbox.

Figure 3.5: Gas Turbine Unit side view.

The gas turbine was:

- installed with a standard acoustic, ventilated enclosure incorporating fire detection and protection facilities;
- provided with all associated ancillary and auxiliary equipment and systems for the safe, efficient, and reliable operation of these units in combined cycle mode. A simple cycle bypass stack provided, the height of which determined to ensure adequate exhaust gas dispersions following the required standards.

While the gas turbine is running, filtered ambient air is drawn through the inlet plenum assembly, then compressed in the 17th-stage axial flow compressor. Compressed air from the compressor flows into the annular space surrounding the ten combustion chambers, from which it flows into the spaces between the outer combustion casings and the combustion liners and enters the combustion zone through metering holes in each of the combustion liners.

The fuel nozzles introduce the fuel into each of the ten combustion chambers, where it mixes with the combustion air and burns.

The hot gases from the combustion chambers expand into the ten separate transition pieces attached to the downstream end of the combustion chamber liners and flow from there to the machine's three-stage turbine section. Each stage consists of a row of fixed nozzles followed by a row of turbine buckets. In each nozzle row, the jet's kinetic energy is increased, with an associated pressure drop, and in each following row of moving buckets, a portion of the jet's kinetic energy is absorbed as useful work on the turbine rotor.

After passing through the 3rd-stage buckets, the exhaust gases are directed into the exhaust casing and diffuser, which contains a series of turning vanes to turn the gases from an axial direction to a radial direction, thereby minimizing exhaust hood losses. Then, the gases pass into the exhaust plenum and are introduced to the atmosphere through the exhaust stack. Resultant shaft rotation turns the generator rotor to generate electrical power and drives the auxiliaries through the accessory gearbox.

3.3.2 Heat Recovery Steam Generator

A heat recovery steam generator or HRSG is an energy recovery heat exchanger that recovers heat from a hot gas stream. The HRSG uses the gas turbine exhaust to produce high temperature, high-pressure steam for use in the steam turbine.

The gas turbine exhaust gas flows through the HRSG exhaust gas inlet ducting and the HRSG before being routed to the atmosphere via the HRSG exhaust gas stack. The HRSG heating surfaces are divided into the following sections in the direction of the exhaust flow;

- HP superheater
- HP evaporator

- HP economizer 2
- LP evaporator
- HP economizer 1

The heating surface primarily consists of horizontal finned tubes joined together to form heating surface packages. After it leaves the last heating surface, the cooled exhaust gas flows through the HRSG exhaust gas stack. Diverter damper to be opened with 10%, 50% & 100% sequentially exhaust flow.

There are three main types of HRSGs: unfired, supplementary fired, and fully fired. Unfired models use only the exhaust energy to produce steam. Supplementary and fully fired models use additional fuel to increase the production of steam.

Here, in this plant unfired model is used. Unfired model HRSG only uses the exhaust energy of the Gas Turbine to produce steam. The steam output from the HRSG is combined in a header system and directed to a steam turbine. Exhaust gas diverter dampers can divert flow to either the bypass stack for simple cycle operation or the HRSG for combined cycle operation.

The Specification of the HRSG are:

Name of the Manufacturer:
Doosan heavy Industry and Construction Co. Ltd.
Type: Unfired (only use the exhaust energy of
Gas Turbine to produce steam).
Shape: Vertical.
Steam Generation Capacity: 78 t/hr.

The HRSG has two pressure levels (HP) and low pressure (LP), forced circulation, and vertical gas flow design. The High Pressure (HP) system steam generation system generates HP steam of specific quality, which means correct pressure and temperature, from the thermal energy contained in the GT exhaust gas. The steam is produced in the HRSG and fed to the HP primary steam system. Besides, it fulfills the following object:

- Delivers feed water to the HP drum during start-up, shutdown, and power operation of the combined-cycle unit.

- Shuts of feed water supply during feedwater control malfunction in order to prevent overfeeding of the HRSG.
- Supplies HP steam produced by the HRSG to the HP main steam system during regular operation.
- Supplies HP feed water to the HRSG HP desuperheating system.
- Maintains and safeguards the HP superheated steam temperature within the allowable main steam system limit during part-load operation at high ambient temperatures.

Figure 3.6: Heat recovery steam generator.

The HP Steam Generation system is located downstream of the exhaust gas inlet of the HRSG. The heating surface is fabricated mainly from finned tubes. The HP system is subdivided into the following sections, listed in the order in which exhaust gas flows through them;

- HP Superheater
- HP Evaporator
- HP Economizer

The HP economizer recovers the remaining heat contained in the exhaust gas at the HP evaporator outlet. The HP evaporator generates steam through a forced circulation loop from and to the HP drum. The HP superheater heats the saturated steam from HP drum temperature up to the required conditions.

The HP superheater and the HP economizer are cross counter flow heat exchangers, and the HP evaporator flow is cross to the exhaust gas flow.

The feedwater is supplied by HP feedwater pumps from the feed water tank to the HP economizers, where it is heated up to economizer outlet temperature and then delivered to the HP Drum, an HP feedwater control station is located downstream of the HP economizer. The location downstream of the economizer is chosen to avoid steaming in economizer.

Water is supplied by the HP evaporator circulation pump from the HP drum through downcomers to the HP evaporator's inlet header. Water partly evaporates in the HP evaporator, and the water/steam mixture is fed in tube risers from the outlet header back to the HP drum driven by the forced circulation. The HP drum has the following functions:

- ensure adequate mixing of feed water and boiler water.
- constitute a water reserve required for the circulation system
- allow water expansion during start-up
- ensure a thorough water and steam separation

The HP drum is installed outside of the casing thus is not heated by exhaust gas. The separation of water and steam is achieved using a water/steam separation system, which restricts the carryover of water to the superheater within limits. Saturated steam flows from the HP drum through connecting piping to the HP superheater. The HP main steam piping system receives HP steam from the HRSG and transfers it to the steam turbine.

The Low Pressure (LP) steam generation system generates LP steam of specific quality, which means correct pressure and temperature, from the thermal energy contained in the GT exhaust gas. The steam is produced in the HRSG and fed to the LP main steam system. Besides, it fulfills the following object:

- Delivers feed water to the LP drum during start-up, shutdown, and power operation of the combined-cycle unit.
- Shuts of feed water supply during feedwater control malfunction in order to prevent overfeeding of the HRSG.
- Supplies LP saturated steam produced by the HRSG to the LP main steam system during regular operation.

The LP Steam Generation system is located downstream of the exhaust gas inlet of the HRSG behind HP economizer 2. The heating surface is fabricated mainly from finned tubes. The LP system consists of an evaporator system only without an economizer and superheater suitable for producing saturated steam. The LP evaporator generates steam through a forced circulation loop from and to the LP drum.

The LP evaporator flow is cross to the exhaust gas flow. The feedwater is supplied by feedwater pumps from the feed water tank, then delivered to the LP drum; water is supplied by an LP evaporator circulation pump from the LP drum through downcomers to the inlet header of the LP evaporator.

The LP drum has the following functions:

- ensure adequate mixing of feed water and boiler water
- constitute a water reserve required for the circulation system
- allow water expansion during start-up
- ensure a thorough water and steam separation

The LP drum is installed outside of the casing thus is not heated by exhaust gas. Saturated steam flows from the LP drum through connecting piping to the LP steam system.

The HRSG exhaust gas system comprises the exhaust gas path's gas-tight component with HRSG exhaust gas inlet ducting, HRSG casing, and HRSG exhaust gas stack.

HRSG shall exhaust through a separate or a standard multi-flue exhaust stack, the height of which and the exhaust gas's exit temperature is determined to ensure adequate dispersion of the exhaust gases following the requirements of the Laws of Bangladesh and the Environmental Guidelines, as applicable.

3.3.3 Steam turbine (ST)

A steam turbine is a device that extracts thermal energy from pressurized steam and uses it to do mechanical work on a rotating output shaft. Here, the L17.5-6.7 17.5MW model is used for CCPP, which collaborated with a gas turbine, heat recovery steam boiler, generator, and auxiliaries to form a power generation equipment set. L17.5-6.7 17.5MW steam turbine is a sub-high pressure, single-cylinder, impulse, condensing steam turbine used to improve the whole CCPP efficiency and reduce power generation cost.

The Specification of steam turbine are:

Name of the manufacturer	:	NANJING TURBINE & ELECTRIC MACHINERY(GROUP) CO., LTD.
Model	:	L17.5-6.70
Generation Capacity	:	17.5 MW
Generation Voltage	:	11 kV
Pressure	:	6.9 MPa
Temperature	:	465^{O}C
Designed Speed	:	3000 rpm

Figure 3.7: Steam Turbine Hall.

The designed speed of ST is 3000rpm; it cannot drive the machine of different or variable speed. The ST unit has the performance of 100% main stream flow rate by-pass to the condenser. The ST unit is of high auto, and the governing system adopts electronic-hydraulic control.

The steam turbine was a proven design, directly coupled to a 50 Hz generator. Design criteria were applied for a long creep life, cyclic duty, and fast start-up. Last stage, blade materials, and exhaust steam wetness were selected to minimize blade erosion.

3.3.4 Cooling Tower

A cooling tower is a heat rejection device that rejects waste heat to the atmosphere through the cooling of a water stream to a lower temperature. Cooling towers have used the evaporation of water to remove process heat and cool the working fluid to near the wet-bulb air temperature or, in the case of closed circuit dry cooling towers, rely solely on air to cool the working fluid to near the dry-bulb air temperature.

Cooling towers are a type of heat exchanger that allows water and air to contact each other to lower the hot water temperature. During this process, small volumes of water evaporate, lowering the water temperature that’s being circulated throughout the cooling tower. The cooling tower cools down water that gets over heated by power plant equipment and processes.

The water is pumped through pipes directly into the cooling tower. Cooling tower nozzles are used to spray the water onto the –fill media,” which slows the water flow down and exposes the maximum amount of water surface area possible for the best air-water contact. The water is exposed to air as it flows throughout the cooling tower. The air is being pulled by a motor-driven electric –cooling tower fan.”

When the air and water come together, a small water volume evaporates, creating an action of cooling. The colder water gets pumped back to the process/equipment that absorbs heat or the condenser. It repeats the loop over and over again to cool down the heated equipment or condensers continuously. A crossflow type cooling tower is used for cooling this plant.

In crossflow cooling tower systems, the water vertically flows through the fill media while the air horizontally flows across the falling water. That is why it calls it "crossflow" because of the air and water cross paths or flows. Because of the crossing of flows, the air does not need to pass through the distribution system. It permits hot water flow via gravity and distribution basins on the top of the tower right above the fill media. The basins are a standard of crossflow cooling towers and are applied to all units.

The Specification of cooling tower:

Name of the manufacturer	:	Tianjin Latino Environmental Technology Co., Ltd.
Type of Cooling Tower	:	Cross Flow
System Volume	:	Basin & Pipe 2900 m^3
Recirculation rate	:	10500 m^3/hr
Heat Exchange Type	:	Tube & Shell
Metalurgy	:	Cu – Alloy & MS
p^H	:	7.5-8.5*

Cooling Tower Fan

Number	1B263
Type	LF854A
Flow	213x10 m^3/hr
Total Pressure	115 Pa
Blade Angle	4^0
Speed	149 r/min
Motor Power	110 Kw
Date of Production	2006.9

Cooling Water Controlling Parameter

p^H	7.5-8.5*
RSI	4-7
Conductivity	Maximum 6 Times of makeup water
TDS	Maximum 6 Times of makeup water
Ca. Hardness	Up to 400 ppm $CaCO_3$
~~M~~” Alkalinity	As Per C.O.C
Silica	<150 Ppm SiO_2
Iron	< 2 Ppm Fe
T. Phosphate	3.0 – 6.0 Ppm
Free Chlorine	0.3 – 0.5 ppm As Cl_2
Suspended Solids	20 Max. Ppm

Figure 3.8: Cooling Tower.

3.3.5 Closed Cycle Circulating Water Pump

Closed Cycle Circulating water pump is used to circulate the hot water from the hot well and passes it to the Cooling tower and passes through the cool water to the condenser wall for the cooling process of the plant. The cooling method is one of the vital processes of a combined cycle power plant. Without proper cooling, higher efficiency cannot be achieved.

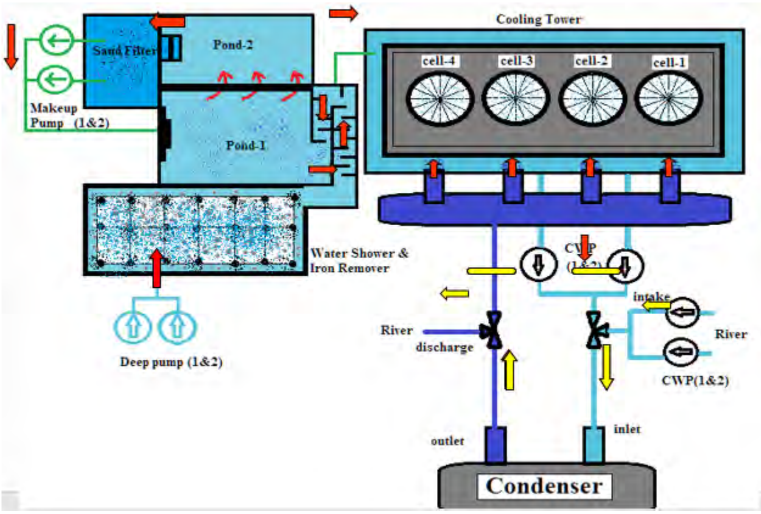

Figure 3.9: Flow diagram of Cooling System.

The Specification of Closed Cycle Circulating Water Pump:

Name of the manufacturer	:	Chaina Shanghai Kaiquan Pump Co. Ltd.
TYPE	:	KOSN 300-N9/392 (T)
Capacity	:	710 m3/hr

Figure 3.10: Closed cycle circulating water pump house.

3.3.6 Water Treatment Tanks

For power generation, water is used in high-pressure boilers, turbines, and cooling towers. Water treatment for power generation is a vital process that requires dependable technology. High purity water ensures proper steam generation system operation and reduces blowdown frequency and use of boiler chemicals. High purity water is also able to better protect against erosion and damage to equipment.

Cooling water quality also affects power plant performance. Water sources need to evaluate for each constituent, or constituent pair should be analyzed individually to determine the maximum allowable concentration. The concentration limit is typically defined by the solubility thresholds of one or more constituents. The first three criteria apply to power plants.

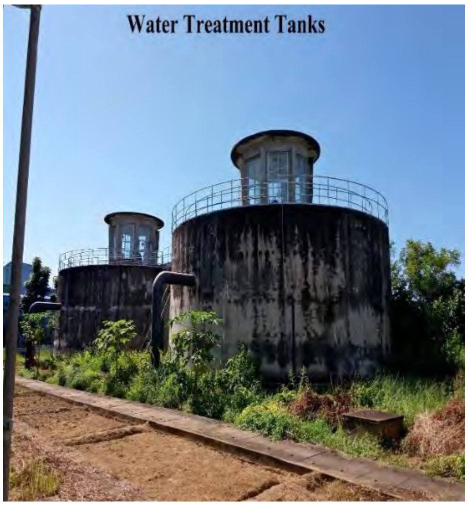

Figure 3.11: Water treatment Tanks.

ZLD systems aim for complete recovery of cooling water, where no wastewater should leave the site, and all waste should be converted to dry form for ultimate disposal. ZLD systems offer the potential for sustainable cooling and a viable waste treatment option for areas where evaporation ponds are infeasible or prohibited. Though ZLD is ideal, not all power plants can achieve these systems, but waste minimization technologies are desirable.

These systems are currently limited by high costs but can be alleviated by developing higher recovery technologies, such as advanced RO membranes. Increased recovery with increased pre-treatments can reduce water volume and, thereby, lower evaporative costs. However, higher solid waste disposal and chemical costs will offset any cost benefits of reduced wastewater volume. With power plant water treatment systems, recovery benefits are often difficult to balance with costs.

3.3.7 Service Air Unit (Screw type air Compressor)

The screw air compressor has been widely used due to its safe operation and convenience in the industrial field. Compressed-air generation is energy-intensive, and for most industrial operations, the energy cost fraction of compressed air is significant compared with overall energy costs. Compressed air is different from the primary energy, and it is converted from the primary energy or secondary energy by an air compressor.

Nevertheless, delivering compressed air to a manufacturing facility is an expensive operation. Delivery requires costly equipment that consumes significant amounts of electricity and requires frequent maintenance.

Rotary screw compressors are also positive displacement compressors. They consist of two rotors inside a casing that compress the air internally. Since there are no valves, maintenance is less intensive. The rotors are oil-cooled, and this oil also seals the internal clearances. The oil is, in its, turn cooled by air or by water. The efficiency of this type of compressor is approximately 71%.

Oil-free screw compressors utilize specially designed air ends to compress air without oil in the compression chamber yielding proper oil-free air. Their efficiency is around 73%. Compressed air is used widely throughout the industry and is often considered the ―fourth utility" at many facilities.

Compressed air systems consist of a supply-side, including compressors and air treatment, and a demand side, which includes distribution and storage systems and end-use equipment. An adequately managed supply side will result in clean, dry, stable air delivered at the appropriate pressure in a dependable, cost-effective manner. For service air unit supply screw type air compressor is used to supply air in the plant.

3.3.8 Condenser

The condenser performance is another parameter that holds much importance in the overall power plant performance, which depends on condenser arrangement, condenser operating pressure optimization, cleanliness factor, number of passes for the condenser, and make up material properties. There are many factors apart from the ones mentioned, but these serve to influence the overall operation considerably.

The Specification of condenser:

Condenser Parameters	Value
Type	N-3000-2
Model	2-phase, surface type
Cooling area	3000 m^2
Cooling & circulating water temperature	38 °C
Cooling & circulating water pressure	0.196 Mpa
Flow speed of Cooling & circulating water	1.85 m/s
Condensing pipe nameplate	HSn70-1
Condensing pipe specification	Φ25 x 1 x 7174 mm
Quantity of condensing pipe	5440 pieces (2720 Pieces in Per chamber)
Process of cooling & circulating water	2 Chambers

3.3.9 Gas Supply Unit

Natural gas is not sufficient in our country, and gradually the volume of natural gas is reduced day by day. Jalalabad gas transmission and distribution company supply only 6 bar pressure gas to the plant. But, for the gas turbine operation, it required 20 to 22 bar gas pressure. A gas booster is used to boost the gas pressure from 6 bar to 22 bar.

Figure 3.12: Gas booster Unit.

3.3.10 Substation & National Grid

The main electrical equipment (isolators and circuit breakers) of 132kV substation yard the specification is:

Main parameter of Circuit Breaker:

Type of circuit breaker	LW6-145(TH) 1250A,40kA,100kA
Type of mechanism	Oil pressure-charged spring
Medium pressure	SF6, 0.7Mpa at 20℃
Phase number	3
Normal operating cycle	O-0.3s-CO-3min-CO (IEC)

Main Parameter of Isolators:

Description	Parameter
Output moment	≥600 N.m
Output angle	90±4°(adjustable)
Time of one opening	9±2 seconds
Time of one closing	9±2 seconds
Motor's power	450W
Control circuit voltage	DC 110V
Auxiliary circuit voltage	AC 220V

Parameter of potential transformer:

1PT	Ratio (11/√3 ∕∕ 0.11√3), Single phase, Accuracy 0.5.
2PT	Ratio (11/√3 ∕∕ 0.11√3 ∕∕0.11/3), V, Accuracy 0.5.

Main parameter of Current Transformer:

Type	LB-145THW2 132kV,500-600/5A
Ratio	500-600/5
Accuracy	0.5

Parameter of main transformer:

Type	SFZ9-55000/132TH
Rated voltage	132±8x1.25%/11 kV
Vector group	YND1
Rated power	55000 KVA
Number of phase	3
Frequency	50 HZ
Cooling method:	ONAN/ONAF 75/100%
Rated current	240.6 / 2887 A
Winding	55 ℃
Top oil	45℃
Surface of tank	70℃


Figure 3.13: 132 kV substation yard.

3.4 Flow Diagram of the Fenchuganj Combined Cycle Power Plant

At first, the air enters into the gas turbine's air compressor at ambient temperature (25^{O}C) and ambient pressure (1.013 bar). Here, the air is compressed approximately at 9 bar and increases the temperature of the air 344^{O}C. Then, it goes to the Combustion chamber. Fuel is also injected in the combustion chamber, which comes from the gas supply unit by JGTDCL, and the inlet pressure is 22.8 bar & temperature is 42^{O}C. In the combustion chamber, air & fuel mixing is burnt & temperature rises to 1091^{O}C & pressure is 8.9bar. After combustion, the flue gas passes through the Turbine section. Here, the flue is expanded by turbine blade and produced electricity 28-30 MW.

After expansion the exhaust gas of the GT passes to the bypass damper section & enters the HRSG inlet section. Diverter damper to be opened with 10%, 50% & 100% sequentially exhaust flow. The exhaust gas temperature is 584^{O}C, and the mass flow of flue gas is 482 kg/hr.

In HRSG two types of flow are done, one is liquid & another is flue gas line. The shape of the HRSG is Vertical type. Here, exhaust flue gas passes to the HRSG in a vertical direction & the tubes are fined in a horizontal position. The exhaust flue gas from GT at 565^{O}C first passed through the HP superheater and heated the last produced steam, which temp is around 484.3^{O}C & this superheated steam is the final product of HRSG & this steam will go to the main stream line of the steam turbine.

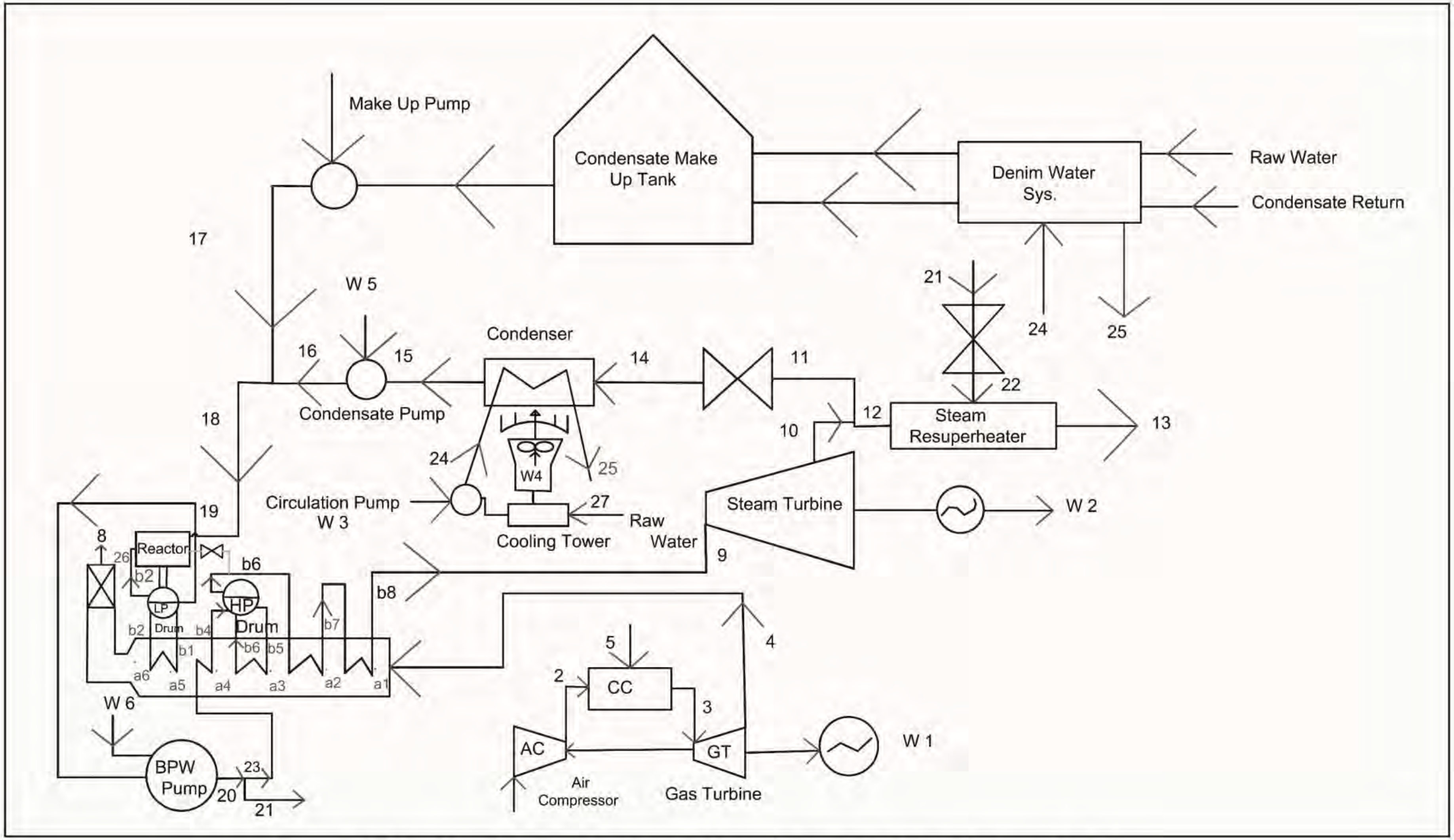

Figure 3.14: Flow diagram of the Fenchuganj Combined Cycle Power Plant.

After passing through the heat to the HP S/H, it passes through HP EVAP, HP2ECO, LP EVAP, & HP1ECO. Here, the LP drum is used to balance HP drum Level & Dearator level. In the HP drum, steam was produced at a pressure of approximately 5.5 MPa, where the LP drum produced a 0.61 MPa pressure supplied to the deaerator & feed water pump to heat the feed water & raise the temp of the feed water at 101.7^{O}C & 10.2 MPa approximately. A boiler feed pump is used to supply feed water to the HP drum & Gland steam tank. Here, Dearator is used to reduce the dissolved oxygen from water & increase the quality of water.

In the waterline, raw water comes from the nearest river called kushiara river & goes to the water treatment section for proper treatment. Then it comes to the Demi tank. The treated water then passes to the condensate makeup tank, where a make-up water pump is used to supply the water into the condensate wall. The cooling tower is used to cool the hot water from the HRSG & exhaust steam. Here, 4 Cooling tower fan is used.

The produced steam from the HP drum passes through the HP superheater & increase the steam temp & then it goes to the main stream line of the Steam turbine. The steam turbine's inlet pressure is 450^{O}C and 5.2 MPa & the exit steam temp of steam turbine is passed through in two ways. One is going to the steam re-superheater or the Gland steam tank & the other is to the Condenser; then, after cooling by a closed-cycle loop by using the Cooling tower and circulating pump, the condensate water is feed to Dearator & LP drum by Condensate Pump again.

In condenser water is reduced their heat by closed cycle loop by using the cooling tower. Raw water from river is the first to come to the water treatment plant. After treating the raw water, the treated water passes to the system by circulating the water pump to condenser section to cool the hot water.

Chapter 4: METHODOLOGY

This section presents the basic methodology for the energy, exergy, and exergoeconomic analysis present for the major components of the Fenchuganj combined cycle power plant located at Fenchuganj, Sylhet, Bangladesh.

4.1 Energy Analysis

Energy analysis based on the First Law of Thermodynamics is the most commonly used power plant performance diagnostic method. Efficiencies and work interactions were calculated for all major components, i.e., compressors, expanders, combustors, HRSG, and steam turbines.

The steps for performing the study will be

- studying the power plant operational characteristics and historical data,
- onsite data collection,
- analyzing the data,
- benchmarking, and
- thermoeconomic analysis of the scope of improvement.

At first, the assessment will be done qualitatively by setting up an interview with the power plant management, engineers, and operators. In the qualitative assessment, easy-to-answer questionnaires will be prepared with due consideration of power plant domain knowledge. This assessment identifies both the best and more inadequate practices found in the power plant and recommend solutions that would help adopt some of the plant's best practices. The qualitative assessment also identifies the weak subareas in the plant where more detailed quantitative analysis is needed to arrive at detailed cost-benefit calculations.

For the quantitative assessment, the overall plant will be sub-divided into sub-systems, i.e., Gas Turbine, HRSG, Steam Turbine, Condenser system, Compressed air system, Deaerator system, Gas Booster system, Nitrogen Plant system, Water treatment system, River water Supply system, Pumps, and fan systems, Transformers, Etc. Relevant data will be collected, and the performance of these systems and the overall plant will be evaluated.

Conventionally, energy assessment for industrial processes is performed by domain experts/engineers, based on their experience, by performing steps like recording, analysis, comparing/benchmarking, targeting, and finally reporting and control. This onsite energy assessment exercise is quite tedious, costly, and prone to human errors and often lacks thorough analysis. This paper proposes an automated energy assessment tool for user-friendly power plants, involves thorough energy analysis through qualitative and quantitative assessment, and can use by less experienced domain experts.

The qualitative assessment is done through easy-to-answer questionnaires prepared with due consideration of power plant domain knowledge. In the qualitative assessment, an interview is a setup with power plant management, engineers, and operators, and answers are obtained. Based on these answers, the tool automatically identifies the plant's improvement areas and possible solutions needed to improve the plant's performance.

The quantitative energy assessment simplified to the following key steps, which are also the functions of this energy assessment tool:

(i) Data preparation
(ii) Data Pre-processor
(iii) Energy Assessment
(iv) KPI Analysis
(v) Benchmarking and gap analysis
(vi) Cost Savings and reporting.

4.1.1 Qualitative Assessment

In the qualitative assessment, the overall plant is sub-divided into sub-systems:

1. HRSG,
2. Steam turbine
3. Condenser system,
4. Compressed air system,
5. Dearator System,
6. Gas Booster System,

7. Nitrogen Plant System,
8. Water Treatment System,
9. Chlorination Plant System,
10. River Water Supply System,
11. Pumps and fan systems,
12. Transformers etc.

The assessment will be done in three aspects of these sub-systems, namely:

1. Monitoring and Benchmarking practices
2. Maintenance and Operation practices and
3. Control and Optimization practices.

The tool asks questions related to each of these subsystems, and then based on the answers, the tool automatically profiles the plant practices against the best in class practices found in the industry. This assessment identifies both the best and more deficient practices found in the power plant and recommends solutions that would help adopt some of the industry's best practices. The qualitative assessment also identifies the weak subareas in the plant where more detailed quantitative analysis is needed to arrive at detailed cost-benefit calculations.

4.1.2 Quantitative Assessment

The details of the various steps involved in quantitative analysis highlighted below:

Data Preparation

This step involves identifying the right tags, collecting and storing historical data from plant historian. The historical data, covering the full operating range of critical variables, are stored in an excel spreadsheet, importing into the tool.

Data Preprocessor

Typically, the historical data is noisy, abundant with missing data, and outliers. So, this function aims to filter out noisy and missing data; perform outlier-detection and replacement. The tool also allows the user to crop data into subsets based on its range filter.

Energy Assessment

A power plant would typically have boilers, air preheaters, coal grinding mills, fans, pumps, and heat exchangers associated with the ‘Boiler Island,’ and steam turbines, gas turbines, heat recovery steam generators (HRSG), condensers, water heaters, pumps, etc. in ‘Turbine Island’ and power transformers, generators in the ‘Power Island.’ The tool allows configuration and calculation of energy losses and various KPIs such as plant hourly profit, heat rate, equipment efficiency, boiler blow-down rates, Etc.

KPI Analysis

The main objective of key performance indicators (KPIs) evaluation and monitoring consists of detecting low performance in power plant operation, investigating issues, and setting up maintenance plans to minimize the operational costs. Another objective is to point out the commissioning and inspection of power plants after significant repairs so that the results recorded during at least six months will be compared with the expected results from the climatic conditions, design and exposure point of view, Etc. The objective entails identifying errors related to layout, incorrect installation, equipment failure, damage, premature aging, Etc. The outputs of KPI will be shown by using trend plots, charts, histograms, pie charts, Etc., for ease of analysis.

Benchmarking and Gap Analysis

Benchmarking is the process of comparing the current project, methods, or processes with the best practices and using this information to drive improvement. The objective of benchmarking is to set the product's appropriate reliability and quality metrics based on similar products in the industry. The gap analysis naturally flows from the Benchmarking Analysis. A gap analysis aims to measure the gap between where they currently are and where they want to be (or where you have set your reliability goals, often based on Benchmarking results). The power plant's current operating performance is compared against a benchmark established based on historical/design data and standards. Various proposals, such as steam blowdown heat recovery, flue gas heat recovery, reduction in variation of critical variables such as excess oxygen in flue gas, superheated steam temperatures, increase in inlet temperature of the steam turbine, augmenting pumps and fans with variable frequency drives (VFDs), Etc., are then evaluated from cost-benefit viewpoints.

Benchmarking against a designed condition can give us valuable information because this helps define the reliability and quality metrics. The gap analysis becomes the basis for identifying implementation actions and priorities. Knowing the size of the gap for each particular reliability and quality metric is valuable because it will tell us how much resources we will dedicate to meeting these metrics.

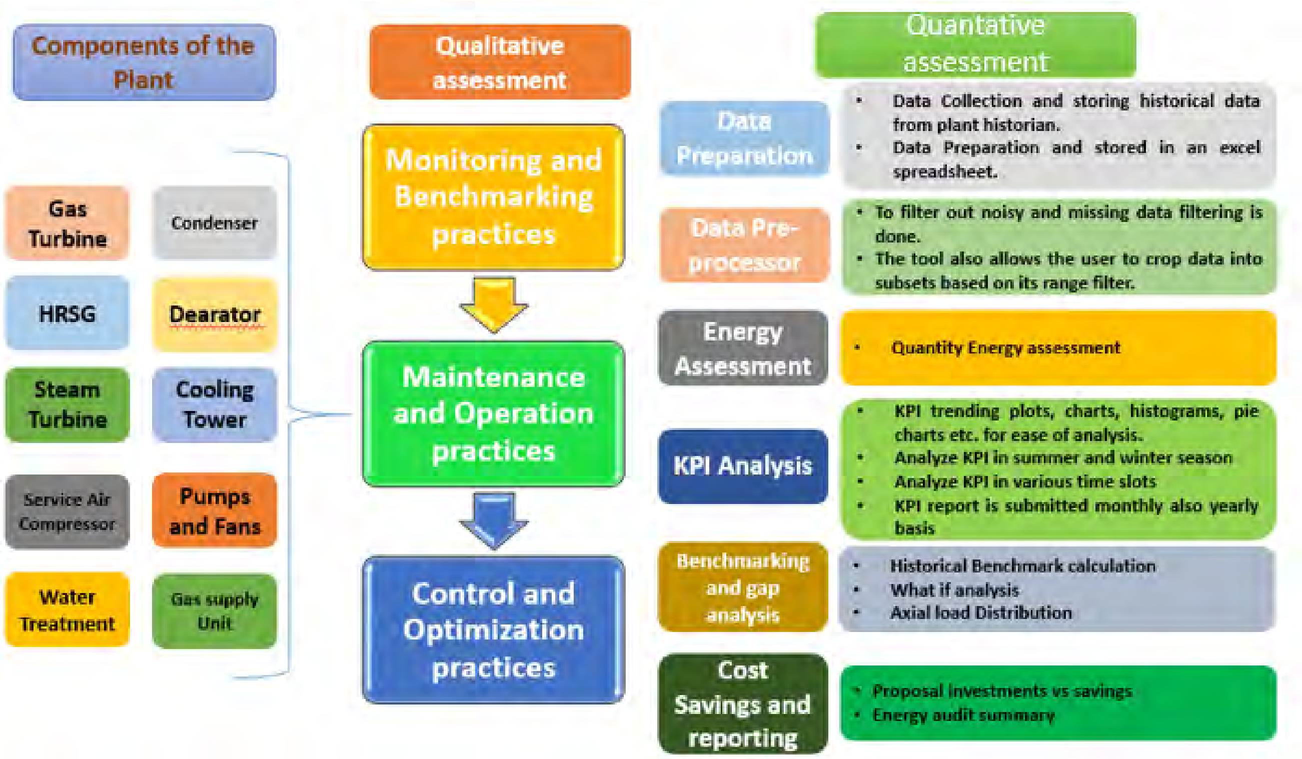

Figure 4.1: Flow Diagram of Energy Audit Methodology.

Cost Savings and Reporting:

Strategic cost-cutting helps lower the cost structure, but it is not about getting cheaper. Instead, strategic cost-cutting helps ensure an organization is ready for growth. It focuses on the aspects of the business that are controllable while freeing up resources to fund the transformation and future growth.

4.2 Exergy Analysis

Exergy analyses of a power plant are based on the Second Law of Thermodynamics. According to Kotas, the Second Law of Thermodynamics is required to establish the difference in quality between thermal and mechanical energy and indicate the directions of spontaneous processes.

A general exergy-balance equation applicable to any thermal system component will be formulated using the first and second thermodynamics laws. The total exergy of a system E will be divided into four components: physical exergy, kinetic exergy, potential exergy, and chemical exergy:

$$E = E^{PH}+E^{CH}+E^{PT}+E^{KN} \quad 4.1$$

Above equation can be expressed on unit-of-mass basis:

$$e = e^{PH}+e^{CH}+e^{PT}+e^{KN} \quad 4.2$$

Kinetic exergy and potential exergy are as follows:

$$e^{KN} = \frac{1}{2}\, V^2 \quad 4.3$$

$$e^{PT} = g.z \quad 4.4$$

In a steady process, when the system is at rest relative to the environment, there is no change in the height and speed. Therefore, kinetic and potential exergies are negligible. In this case, physical exergy, also known as thermodynamic exergy, would be defined as the maximum theoretical useful work. The maximum theoretical useful work is the amount of work obtained from the interaction between a system and its environment until it reaches the dead state.

The chemical exergy is associated with the departure of a system's chemical composition from its chemical equilibrium. The chemical exergy is an important part of exergy in the combustion process. 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 resulting from irreversibilities.

For Exergy analysis, the plant is separated into eight subsystems. Exergy inflows and outflows to these subsystems, exergy destruction in these subsystems, and exergy losses to the environment are calculated. Schematic of these subsystems and exergy balances for them may be found in the following equations of air compressor, combustion chamber, gas turbine, steam turbine, HRSG, Condenser, condensate pump and related junction, Boiler Feed Water Pump.

The exergy-balance equations and the exergy destroyed during the process occur in each component of the power plant and calculate the exergetic efficiency and overall exergetic efficiency of the system accordingly.

4.3 Exergoeconomic Analysis

Exergoeconomics is a combination of exergy analysis and economic principles, which helps us understand how costs flow in a system and optimize system performance. The term exergoeconomics is to characterize a combination of exergy analysis with economic analysis when, in this combination, the exergy-costing principle will be used. In this way, a distinction can be made between exergoeconomic methods and applications on one side, and other numerous applications on the other side, in which results from a thermodynamic analysis (sometimes including an exergy analysis) and an economic analysis are presented (under the term thermoeconomic analysis) but without applying the exergy-costing principle.

Existing methods of exergoeconomic analysis and optimization of energy systems operate with single average or marginal cost values per exergy unit for each material stream in the system being considered. These costs do not contain detailed information on (a) how much exergy and (b) at what cost each exergy unit was supplied to the stream in the upstream processes. The cost of supplying exergy, however, might vary significantly from one process step to the other. Knowledge of the exergy addition and the corresponding cost at each previous step can be used to improve the costing process. This paper presents a new approach to exergy costing in exergoeconomics. The monetary flow rate associated with the thermal, mechanical, and chemical exergy of a material stream at a given state is calculated by considering the complete previous history of supplying and removing the corresponding exergy form units to and from the stream being considered. When exergy is supplied to a stream, the cost of adding each exergy unit to the stream is calculated using the cost of the product exergy unit for the processor device in which the exergy addition occurs. When the stream being considered supplies exergy to another exergy carrier, the last-in-first-out (LIFO) principle of accounting is used for the spent exergy units to calculate the cost of exergy supply to the carrier. The new approach eliminates the need for auxiliary assumptions in the exergoeconomic analysis of energy systems and improves the costing process's fairness by taking a closer look at both the cost-formation and the monetary-value-use processes. This closer look mainly includes the simultaneous consideration of the exergy and the corresponding monetary values added to or removed from a material stream in each process step. In general, the analysis becomes more complicated when the new approach is used instead of the previous exergoeconomic methods. The benefits of using the new approach, however, significantly outweigh the increased efforts. The new approach, combined with some other recent developments, makes exergoeconomics an objective methodology for analyzing and optimizing energy systems.

The objective of a thermoeconomic analysis shall be

1. to calculate separately the cost of each product generated by a system having more than one product,
2. to understand the cost formation process and the flow of costs in the system,
3. to optimize specific variables in a single component, or
4. to optimize the overall system.

For the evaluation of exergoeconomic analysis, here SPECO method will be used.

4.3.1 SPECO Method

The Specific exergy costing method (SPECO) consists of the following three steps. The first step is the identification of exergy streams. All material and energy streams cross the boundaries of the components being considered, which should be identified first. This is accomplished by inspection of the process flow diagram. The exergy streams associated with the entering and exiting material and energy streams are known from the exergy analysis. At this point, a decision must be made concerning whether the analysis of the components should be conducted using total exergy or separate forms of the total exergy of a material stream. Considering separate exergy forms usually improves the accuracy of the results. However, this improvement is often marginal and unnecessary for extracting the main conclusions from the thermoeconomic evaluation.

The second step is the definition of fuel and product. In evaluating the performance of a component, it is, in general, meaningful and appropriate to operate with exergy differences associated with each material stream between inlet and outlet. Exergy differences (exergy additions to or removals from a stream) should be applied to all exergy streams associated with a change of physical exergy and some exergy streams associated with converting chemical exergy. In many cases involving conversion of chemical exergy (e.g., conversion of the chemical exergy of solid fuel into chemical and thermal exergy through a gasification process), the purpose of owning and operating the component dictates that the chemical exergy at the outlet is considered on the product side and the chemical exergy at the inlet on the fuel side.

The third step is writing the cost balances and auxiliary equations. Exergy costing usually involves cost balances formulated for each system component separately.

The steady-state form of control volume cost balance is:

$$\sum_{j=1}^{n} \dot{C}_{j,k,in} + \dot{Z}_k = \sum_{j=1}^{m} \dot{C}_{j,k,out} \qquad 4.5$$

The above equation states that the exiting exergy streams' total cost equals the total expenditure to obtain them: the cost of the entering exergy streams plus the capital and other costs. The total cost of a stream will be defined as:

$$\dot{C}_j = c_j \, E_j \qquad 4.6$$

The term cj in the above equation is the Levelized cost per unit of exergy. In analyzing a component, we may assume that the costs per exergy unit are known for all entering streams. These costs are known from the components they exit or if a stream enters the overall system consisting of all components under consideration, from this stream's purchase cost. Consequently, the unknown variables that need to be calculated with the aid of the cost balance for the kth component are the costs per unit exergy of the exiting streams. This is shown in the following Figure:

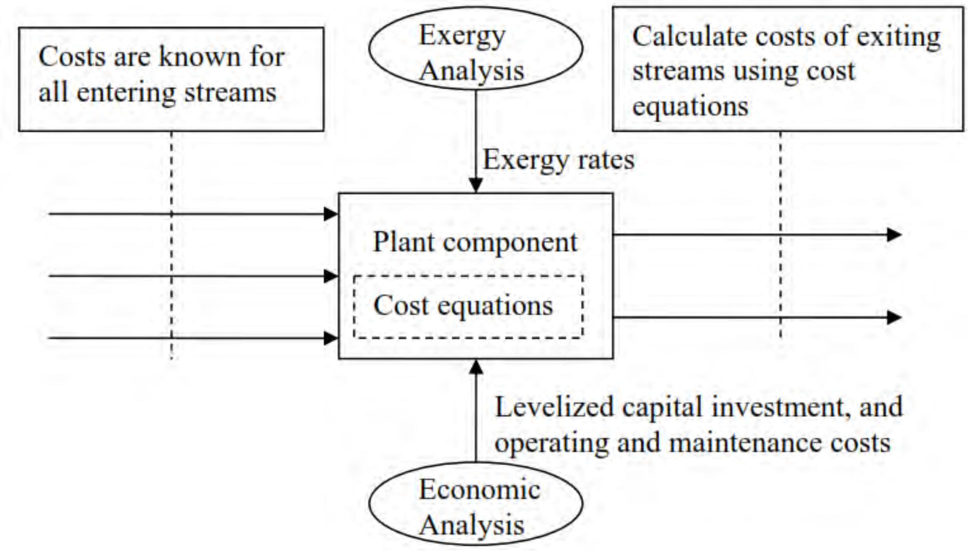

Figure 4.2: Schematic diagram of the SPECO Method Description.

In general, if Ne exergy streams are exiting the component being considered, we have Ne unknowns and only one equation, the cost balance. Therefore, we need to formulate Ne-1 auxiliary equations. This is accomplished with the aid of the F and P rules.

F- rules

The total cost associated with removing exergy must be equal to how the removed exergy supplied to the same stream in upstream components.

P- rules

Each exergy unit is supplied to any stream associated with the product at the same average cost.

Chapter 5: Energy Analysis

Bangladesh is a developing country. For the development of a country, the first essential requirement that is needed is energy. Energy is the prime mover for developing a country both socially and economically, but nowadays energy resources overview of the world is not in pleasurable condition, and for Bangladesh, the condition is worse. Though the quantity of natural resources decreases day by day, wastage of natural resources in a different field is not controlled due to lack of proper management. Energy Auditing will be a possible strategy in Energy management cases to control such waste in industries, power plants, houses, different types of shops, and Hospitals.

An energy audit is a technique for identifying energy losses, quantifying them, estimating conservation potential, evolving technological options for conservation, and evaluating techno-economics for the measures suggested. Assist industries in reducing their energy consumption, promote energy-efficient technologies among industry sectors, disseminate information on energy efficiency through training programs and workshops, promote the transfer of energy-efficient and environmentally sound technologies to the industrial sectors in the context of climate change.

A perfect energy audit will always help to understand more about the ways energy and fuel are used in their industry and identify areas where waste can occur and where the scope for improvement is needed. The energy audit would positively orient the energy cost reduction, preventive maintenance, and quality control programs, which are vital for production and utility activities. Such an audit program will help keep the focus on variations in the energy costs, availability, and reliability of supply of energy, help decide on the appropriate energy mix, identify energy conservation technologies, and retrofit for energy conservation equipment. The energy audit's primary objective is to determine ways to reduce energy consumption per unit of product output or to lower operating costs. The energy audit provides a benchmark, or reference point, for managing and accessing energy use across the organization and provides the basis for ensuring more effective energy use.

The fundamental goal of energy management is to produce goods and provide services with the least cost and least environmental effect. The term Energy Management means adjusting and optimizing energy, using systems and procedures to reduce energy requirements per unit of output while holding constant or reducing total costs of producing the output from these systems. The objective of Energy Management is to achieve and maintain optimum energy procurement and utilization throughout the organization and:

- To minimize energy costs and waste without affecting production and quality.
- To minimize environmental effects.
- Improving energy efficiency and reducing energy use, thereby reducing costs.

Power systems diagnosis is conducted, using thermal analysis tools, to determine the locations, magnitudes, and types of waste and losses within these systems. Identifying the sizes and locations of the deficiencies will assist in making modifications to improve the system. The primary analysis and assessment tools used to optimize thermal energy systems are energy analyses and have long been analyzed and evaluated using energy tools.

The basic methodology for the energy analysis of the Fenchuganj combined cycle power plant's major components is described in this section.

5.1 Energy Analysis

Energy analysis was carried out using the First Law of Thermodynamics is the most commonly used method for power plant performance diagnostics. The steps for performing the study will be

- studying the power plant operational characteristics and historical data,
- onsite data collection,
- analyzing the data,
- benchmarking, and
- thermo-economic analysis of the scope of improvement.

It is possible to calculate the cycle thermal efficiency, which is the work output ratio to the heat input. Also, the energy loss in each component and the entire plant will compute by using energy balance. For any control volume at steady state with negligible potential and kinetic energy changes, energy balance reduces to [Avval et al., 2011]:

$$Q\text{-}W = \sum m_e h_e + \sum m_i h_i \qquad 5.1$$

Energy analysis of an active CCPP (Combined Cycle Power Plant) is performed with the help of the actual operating data taken from the plant's computer control unit. For the plant's thermodynamical performance assessment, energy analysis related to thermodynamics' first law is performed.

Performance analysis methods are explained in the following clauses. The equations, thermophysical properties of the working fluids, and the thermodynamical analysis of the power plant components are given in the following parts. Given equations are used for parametric analysis of the system, and results are compared with actual operating results.

The energy balance equations and performance indicator equations for various components, and all the parameters are presented in below table as per the flow diagram power plant shown in Figure 3.14 are as follows:

Table 5.1: Energy Performance Parameter Equations.

SL No	Particulars	Equations	
1	**Air Compressor**	Pressure ratio, r = $\frac{P_2}{P_1} = \frac{P_4}{P_3}$ where, P_2 and P_1 are the compressor inlet and outlet air pressures, respectively.	5.2
		Compressor work done: $W_{AC} = m_a\ C_{pa}\ (T_2 - T_1)$	5.3
2	**Combustion Chamber**	Heat supplied in the system: $Q_{in,CC} = m_f$ x LHV	5.4
		Combustion efficiency: $m_g C_{pg}\ (T_3 - T_2) = \eta_{cc}\ Q_{in,cc}$	5.5
		$m_g = m_f + m_a$	5.6
3	**Gas Turbine**	Turbine Work: $W_{GT} = m_g\ C_{pg}\ (T_3 - T_4)$	5.7
		Friction Loss = $m_{oil}\ C_{p,oil}\ (T_i - T_o)$ Where; T_i = Inlet oil temperature T_o = Outlet oil temperature	5.8
		Net Work: $W_{net} = W_{GT} - W_{AC}$	5.9
4	**Steam Turbine**	Work done on Steam turbine: $W_{ST} = m_s\ (h_8 - h_9)$	5.10

SL No	Particulars	Equations	
5	**Heat Recovery Steam generator (HRSG)**	Mass flow rate of the steam generated at the HP steam drum: $m_{superheated\ steam} = m_g\ [(1-HL)(h_{a1}-h_{a4})/(h_{b8}-h_{b4})]$	5.11
		$Q_{HP} = m_{HP}\ x\ h_{HP}$	5.12
		$Q_{LP} = m_{LP}\ x\ h_{LP}$	5.13
		Where; Q_{HP} = HP steam heat output (kW) m_{HP} = HP steam heat output (kg/s) h_{HP} = HP steam enthalpy (kJ/kg) Q_{LP} = LP steam heat output (kW) m_{LP} = LP steam heat output (kg/s) h_{LP} = LP steam enthalpy (kJ/kg)	5.14
		HRSG efficiency = $\frac{Heat\ Output}{Heat\ Input}$ x 100%	5.15
6	**Condenser**	The heat rejection in condensing system: $m_s\ h_{14} = Q_{rej,condenser} + m_s\ h_{15}$ $Q_{rej,condenser} = m_s\ (h_{14} - h_{15})$	5.16
7	**Pump**	$W_{cond,pump} = m_w\ (h_{15} - h_{16})$	5.17
		$W_{feedwater\ pump} = m_w\ (h_{20} - h_{19})$	5.18
		$W_{pump} = W_{feedwater\ pump} + W_{cond,pump} + W_{Makeup\ water,pump} + W_{CWpump}$	5.19
		Net Work output of steam cycle: $W_{net,ST} = W_{ST} - W_{pump}$	5.20
8	**Service Air Unit (Screw type air Compressor)**	Volume rate of flow: $q_{FAD} = q_N x \frac{T_{FAD}}{T_N} x \frac{P_N}{P_{FAD}}$ Or, $q_N = (q_{FAD}$ x T_Nx $P_{FAD}) / (T_{FAD}\ x\ P_N)$	5.21
		Where, q_{FAD} = Free air delivery (m^3/min) q_N = Normal volume rate of flow (m^3/min) T_{FAD} = Standard inlet temp °C T_N = Normal reference temp °C P_{FAD} = Standard inlet pressure bar P_N = Normal reference pressure bar	
		Mass Flow, $\dot{m}$(kg/min) = q_N (m^3/min) x ρ (kg/m^3)	5.22
		Isothermal Power, (kw) = $\frac{P_1 Q_f \log r}{36.7}$	5.23
		Where, P_1=Absolute inlet pressure (kg/cm^2) r=compression ratio Q_f= Free air delivered (m^3/hr)	

SL No	Particulars	Equations	
		Compressor Idle Time, $t_I = \frac{V_R(P_{max}- P_{min})}{q}$	5.24
		Compressor Running Time, $t_R = \frac{V_R(P_{max}- P_{min})}{\dot{V}-L_B}$	5.25
		Motor Cycle Speed, A= $\frac{60}{t_I+t_R}$	5.26
		% leakage = $\frac{T}{T+t}$ x 100	5.27
		System leakage quantity, q (m^3/min) = $\frac{T}{T+t}$ x Q	5.28
		Leakage quantity per day = q x 24 x 60	5.29
		Energy lost due to leakage/day = 0.11 x Leakage quantity per day	5.30
		Annual electricity cost:	
		$\frac{Motor\ Power\ x\ Annual\ Hr\ of\ Operation\ x\ electricity\ cost\frac{TK}{kWh}x\%time\ running\ fully\ loadedx\%Full\ load\ Power}{motor\ efficiency}$	5.31
		Ventilation requirement in the compressor room: P = $\dot{m}C_P\Delta T$	5.32
		Where, P = Roughly 94% of the supplied shaft power to the compressor + the difference between the supplied total power to the compressor package & the supplied shaft power to the compressor + the stated heat flow from the refrigerant dryer.	
		Pressure drop in the piping: ΔP = 450 $\frac{q_c^{1.85}xl}{d^5x\ P}$	5.33
		Air receiver volume: V = $\frac{TxCxP_a}{P_1-P_2}$	5.34
		Where, V= receiver volume, m3 T=Time allowed for pressure drop to occur C= Air demand Pa= Absolute atm. Pressure, bar P_1= Initial receiver pressure, bar P_2= Final receiver pressure, bar	
		Cost savings = [(Energy Savings in kw/yr)/(Kw/unit of fuel)x(Tk/unit of fuel)]/Primary heater efficiency	5.35

SL No	Particulars	Equations	
9	**Condenser**	Condenser Effectiveness: $\varepsilon = \frac{\Delta T}{T_{sat} - T_{in}}$	5.36
		Log- Mean Temperature Differences: LMTD = $\frac{\Delta T}{ln\frac{T_{sat-} T_{in}}{T_{sat-} T_{out}}}$	5.37
		Condenser Efficiency: $\eta_{Cond} = \frac{\text{Actual Cooling Water Temperature rise}}{\text{Maxsimum possible Temperature rise}}$ x 100%	5.38
10	**Cooling Tower**	Cooling Tower Approach: $CT_{Approach} = T_{w1}$-WBT	5.39
		Cooling Tower Range: $CT_{Range} = T_{w1} - T_{w2}$	5.40
		Mass Flow Rate of Water, M_w = 11005 m^3/hr x 1000 = 11005 $x10^3$ Kg/hr	5.41
		Heat Loss By Water, HL = M_w x C_{pw} x($T_{w1} - T_{w2}$) Kj/hr	5.42
		Volume of Air Required, V = (HL x V_{S1})/ [($H_{a2} - H_{a1}$) – ($W_2 - W_1$) x C_{pw} x T_{a2}	5.43
		Mass of Air required = Volume of Air Required / V_{S1}	5.44
		Heat gain by Air = $\frac{V \text{ x } (H_{a2} - H_{a1}) - (W_2 - W_1) \text{ x } C_{pw} \text{ x } T_{a2}}{V_{S1}}$	5.45
		Effectiveness,$\varepsilon = \frac{T_{w1} - T_{w2}}{T_{w1} - T_{a1}}$	5.46
		Efficiency,$\eta = \frac{T_{w1} - T_{w2}}{T_{w1} - WBT} = \frac{CT_{Range}}{CT_{Approach}}$	5.47
		Various Loses:	
		Drift Losses, DL = (0.2 x M_w) / 100	5.48
		Wind age Losses, WL = 0.005 x M_w	5.49
		Evaporation Losses, EL = 0.00085 x M_w x $T_{w1} - T_{w2}$	5.50
		Cycles of Concentration, COC = $\frac{\text{Dissolved Solids in Circulating Water}}{\text{Dissolved Solids in Make Up Water}}$	5.51
		Blow Down Losses, BL = $\frac{\text{Evaporation Loss}}{(\text{C.O.C} - 1)}$	5.52
		Total Loss Of Cooling Tower = DL+WL+EL+BL	5.53

Table 5.2: Performance Indicator Parameter of the Combined Cycle Power Plant.

Performance Indicator Parameter		
For Simple Cycle	Thermal efficiency of the plant: $\eta_{GT} = \frac{W_{net,GT}}{Q_{in,CC}}$	5.54
	Work ratio = $\frac{W_{net}}{W_{gt,actual}}$	5.55
	Specific Fuel Consumption = $\frac{m_f x\ 3600}{W_{net} x\ 1000}$	5.56
	Heat rate = $\frac{3600}{\eta_{th}}$	5.57
For Combined cycle	Total net power o/p in combined cycle = $W_{net,GT} + W_{net,ST}$	5.58
	$\eta_{Combined\ Cycle} = \frac{W_{net,GT}+W_{net,ST}}{Q_{in,CC}}$	5.59
	Work ratio = $\frac{W_{net,GT}+W_{net,ST}}{W_{GT}+W_{ST}}$	5.60
	Heat rate in combined cycle = $\frac{3600}{\eta_{combined\ cycle}}$	5.61

5.2 Result and Discussion

For Energy analysis, at first, the Fenchuganj Combined cycle power plant's operating data were collected. Data were taken between 2018 to 2019, both in the summer and winter seasons. Every day 24 hours of data were taken for this analysis. The majority percentage of the data was collected directly through the distributive control system (DCS) & the rest of the data was collected manually using a fire gun, anemometer, multimeter, tachometer, Etc. For calculation purposes, here, consider the temperature parameter is the main factor. The temperature parameter is the reference parameter, and the overall calculation was done accordingly.

The power plant's operating data were collected in the winter season & summer season (2018-2019)-the collected data from the power plant attached in annexure-1 for details.

For the energy analysis of the Fenchuganj Combined cycle power plant, the equations are shown in table 5.1 & 5.2, and the operating parameter collected from the plant, use to analyze the energy performance.

5.2.1 Effect of Power Output of Gas Turbine with change in Inlet Air Temperature

A survey through the result showed that the power output of the GT within the range of 26.98 MW to 29.19 MW. Figure 5.3 shows that:

1. The power output of the GT decreased as the ambient Temperature Increases.
2. The plant's power output did not reach the plant-designed specification, i.e., 35 MW, because it is based on ISO specification of 15˚C.
3. The GT's power output is more decreased in between the temperature range is 30^{O}C to 35^{O}C.
4. An inlet air cooler can increase the gas turbine's power output by decreasing the inlet air temperature. To install an air cooling process in the plant to initially reduce the inlet air temperature, it needs to invest approximately $240,000.00 initially. After installing, the power output will be increased by 2 MW more than the previous system.
5. Increase in work ratio & mass flow rate of the working fluid increase the Power Output. Another way to increase the gas turbine work output is to install an additional air compressor to supply more air to the combustion chamber. It will help to burn the fuel more. To install an air compressor, spend approximately $380,000.00, and the power output will be increased more than 3 to 4 MW more.

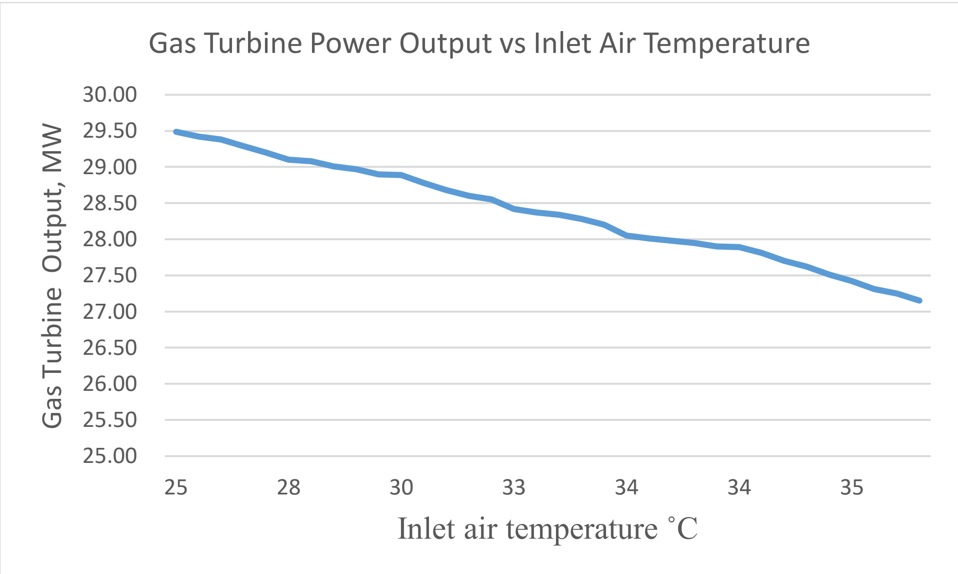

Figure 5.1: Effect of Gas Turbine Power Output with the change in Inlet Air Temperature.

5.2.2 Effect of thermal efficiency of Gas turbine with change in Inlet Air Temperature

It was discovered that the thermal efficiency of the Gas Turbine is within the range of 28.93% to 32.80%.

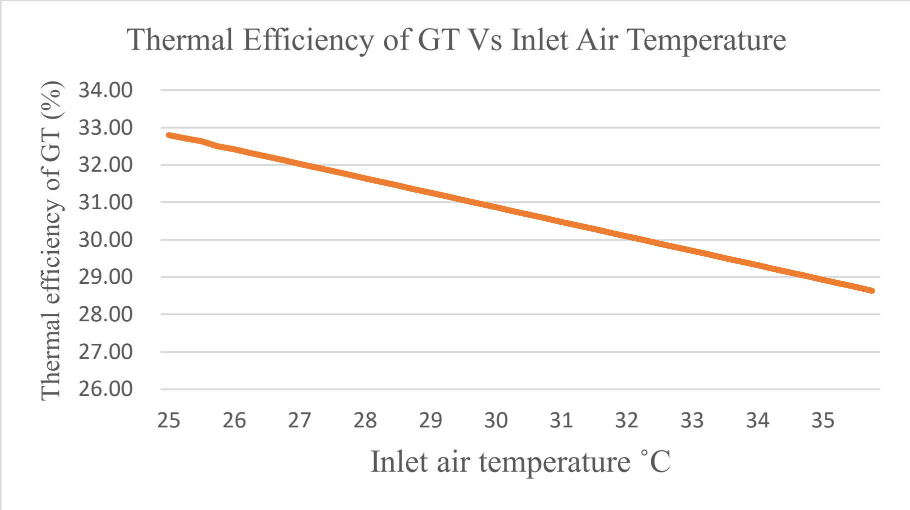

Figure 5.2: Effect of Gas Turbine thermal efficiency with the change in Inlet Air Temperature.

Figure 5.4 shows that:

1. Thermal efficiency of the GT decreased as the inlet air temperature (ambient Temperature) Increases.
2. The GT's power output is more decreased in between the temperature range is 31^{O}C to 35^{O}C. From 26^{O}C to 30^{O}C, the thermal efficiency of the plant is moreover the same.
3. Using an inlet air cooling system or additional air compressor will increase the gas turbine's thermal efficiency.

5.2.3 Effect of thermal efficiency of Gas turbine with Specific fuel consumption

From figure 5.5, the thermal efficiency of the GT is within the range of 29.96% to 32.32% when SFC is 0.22 to 0.24. So, it is clear that, When the plant is operating in full load, and the ambient temperature is also near about 25^{O}C, then the SFC in that machine is also low. SFC increases with increasing the ambient air temperature & decreases with the load.

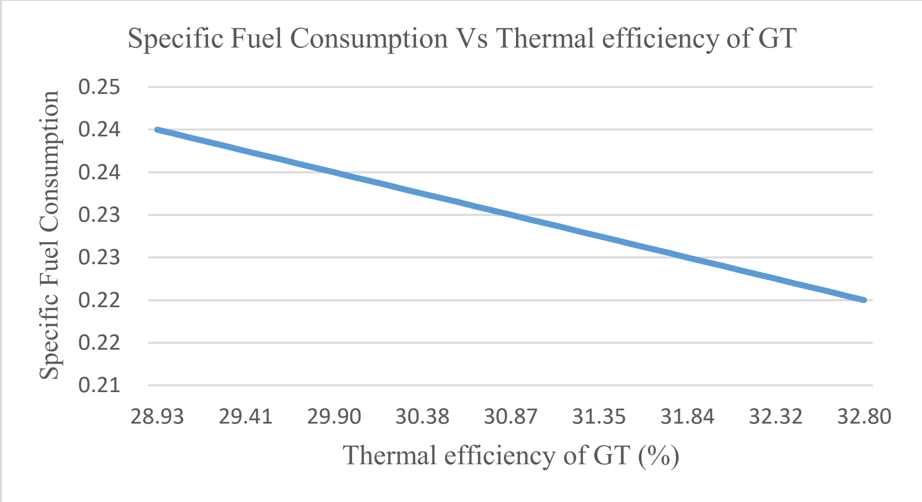

Figure 5.3: Effect of Gas Turbine thermal efficiency with SFC.

5.2.4 Effect of Heat Rate (HR) in simple cycle with Inlet Air Temperature

From figure 5.6, the Heat rate of the Plant in Simple Cycle is within the range of 11,138 kJ/kWhr to 12,017.06 kJ/kW-hr. The Heat Rate of the plant in simple cycle operation is increased with the increasing of Inlet air (ambient) Temperature. When the plant's Heat Rate increased, then the plant's SFC also increased, increasing the Operation cost of the plant and decreasing the machine efficiency.

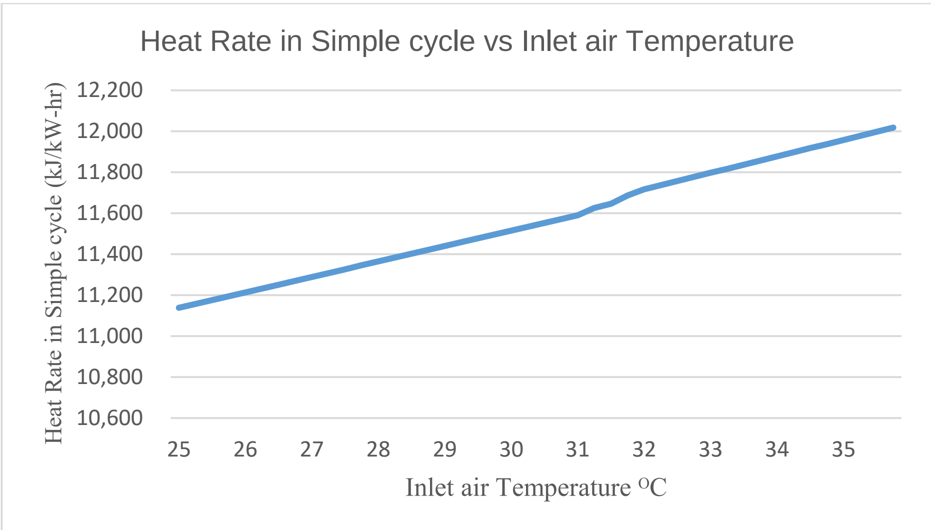

Figure 5.4: Effect of Heat rate of the plant in simple Cycle operation with the change in Inlet Air Temperature.

5.2.5 Effect of Overall efficiency in simple cycle with Inlet Air Temperature

It was discovered that the gas turbine's overall efficiency is within the range of 29.81% to 31.84%.

From figure 5.8, it is being seen that:

1. Overall efficiency of the plant in simple cycle decreased as the Inlet air (ambient Temperature) Increases.
2. The plant's overall efficiency did not reach the plant specification of 35 MW because it is based on an ISO specification of 15˚C.
3. Using an inlet air cooling system or additional air compressor will increase the gas turbine's overall efficiency.

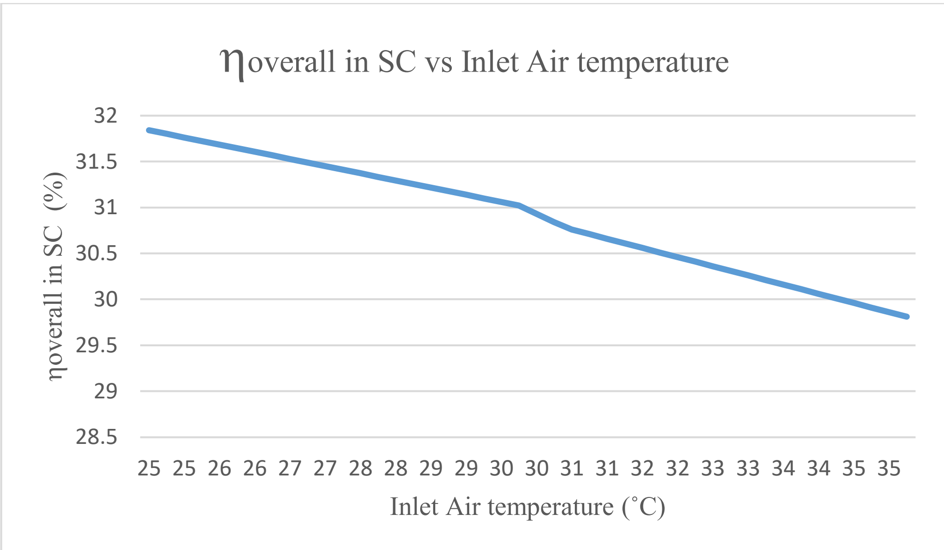

Figure 5.5: Effect of Heat rate of plant in simple Cycle operation with change in Inlet Air Temperature.

5.2.6 Effect of steam turbine net-work output with varying load

Figure 5.9 shows that the maximum network done by steam Turbine is 14.56 MW at 100% load, and the minimum net-work done by steam Turbine is 6.7 MW at 50% load. If needed to decrease the load more, we need to shut down the steam cycle from the plant. The minimum steam load in that plant is 50%, the net-work output of the steam cycle increase with increasing the load in that plant.

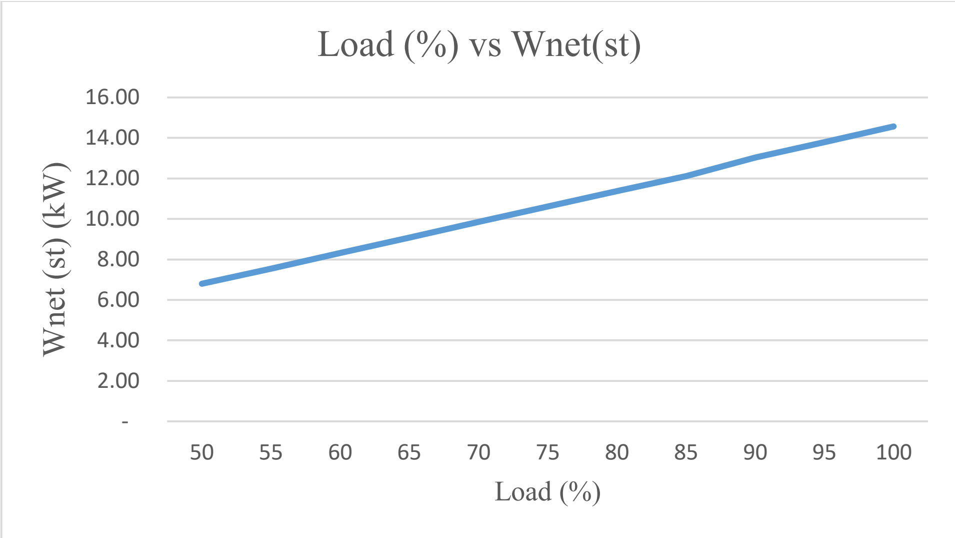

Figure 5.6: Effect of steam turbine net-work output with varying load.

5.2.7 Effect of steam flow rate with steam turbine work output

In the following figure 5.10, the steam turbine's net-work decreased with the decrease in steam's mass flow rate. The mass flow rate of steam can be governed by operating the main guide valve & decrease the percentage of inlet guide vane of GT exhaust. Here, the minimum steam flow rate is 7.65 kg/s, and the maximum steam flow rate is 15.5 kg/s.

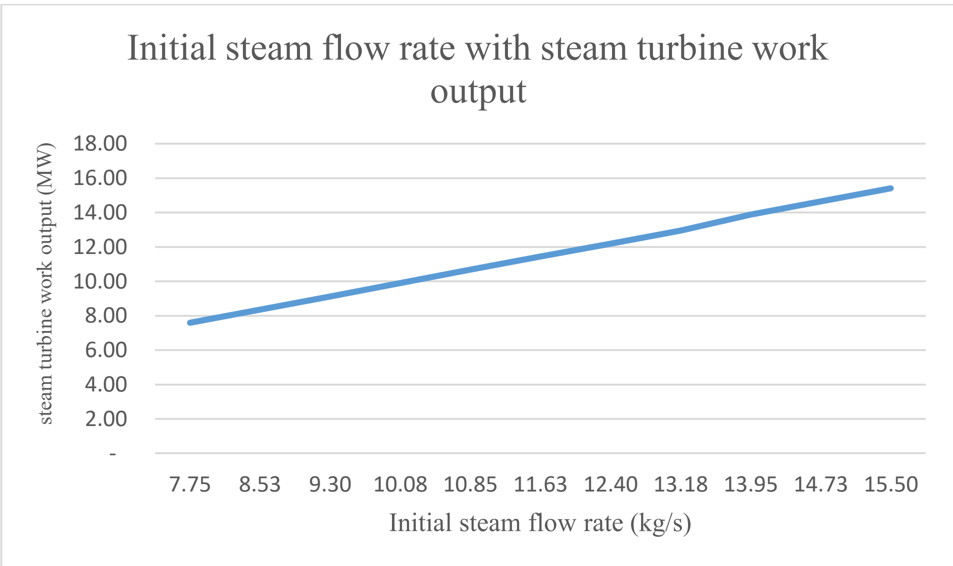

Figure 5.7: Effect of steam turbine net-work output with the mass flow rate of steam.

5.2.8 Effect of Steam consumption/rating with steam flow rate

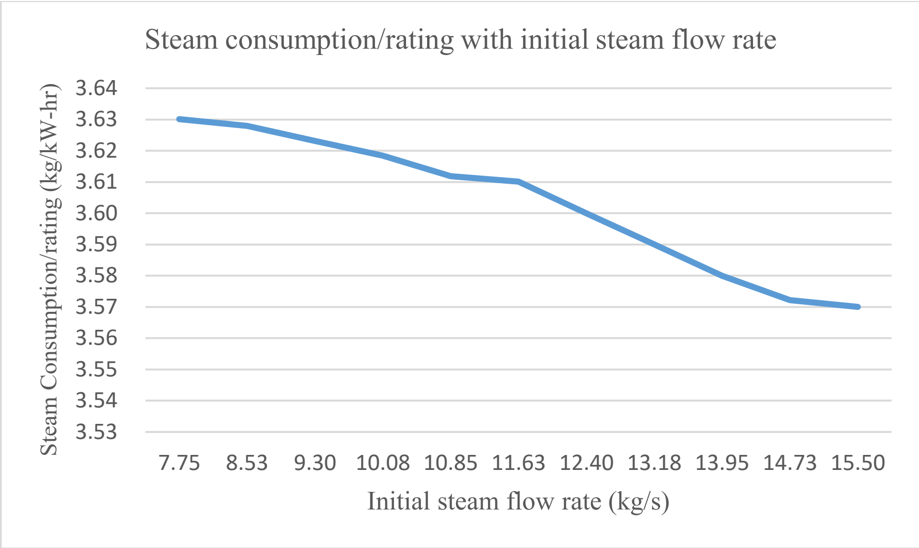

Figure 5.8: Effect of Steam consumption/rating with the steam flow rate.

In the above figure 5.11, the steam consumption/rating of the steam cycle increases with decreasing the steam's mass flow rate. The maximum steam consumption rating is 3.63 kg/kW.h, when the mass flow rate is 7.65 kg/s & the minimum steam consumption rating is 3.57 kg/kW-h when the mass flow rate is 15.5 kg/s.

5.2.9 Effect of steam consumption/rating with varying load

From the following figure 5.12, steam consumption/rating is increasing with decreasing the load. The maximum steam consumption/rating at 50% load and minimum steam consumption/rating at 100% load, i.e., 3.57 kg/kW-hr.

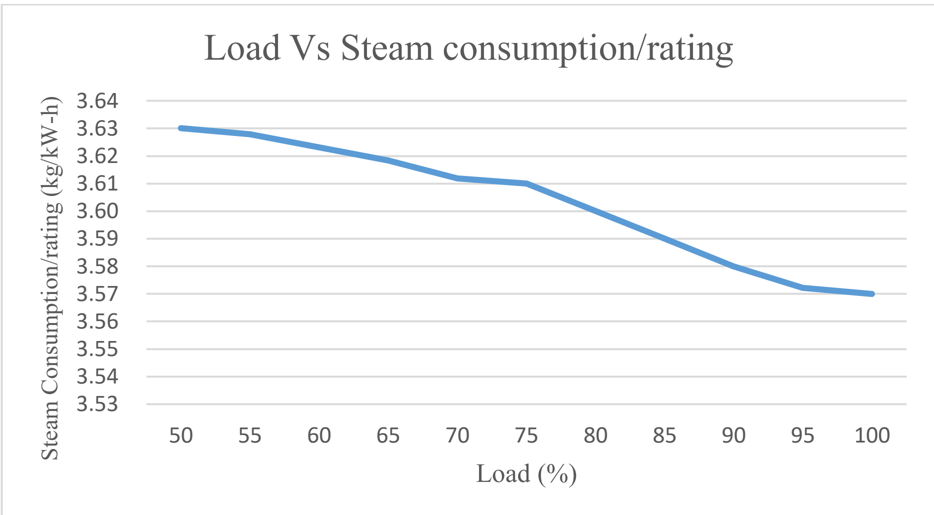

Figure 5.9: Effect of steam consumption/rating with varying load.

5.2.10 Effect of combined cycle efficiency with varying load

From figure 5.13, the combined cycle efficiency decreased with a decreased load. The combined cycle efficiency maximum at 48.45% at 100% load & minimum is 38.74 % at 50% load.

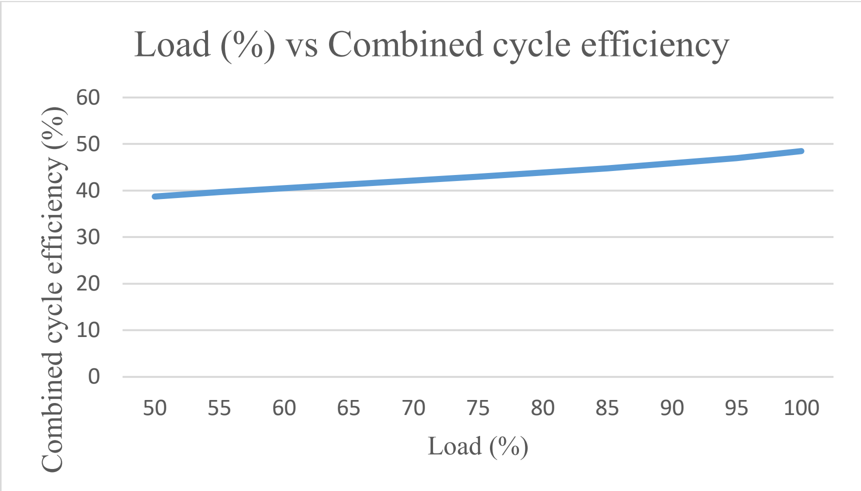

Figure 5.10: Effect of combined cycle efficiency with varying load.

Also, the combined cycle efficiency decreases with the decreased mass flow rate of steam. The mass flow rate of steam can be governed by operating the main guide valve & decrease the percentage of inlet guide vane of GT exhaust.

5.2.11 Effect of Heat rate with Total work output

From the above result and following figure 5.15, the combined cycle's HR decreased in increasing load and total power output in the combined cycle. The minimum Heat Rate is 7429.94 kJ/kWhr at maximum output (43.76MW), and the maximum Heat Rate is 9293.56 kJ/kWhr at the minimum output (34.82 MW).

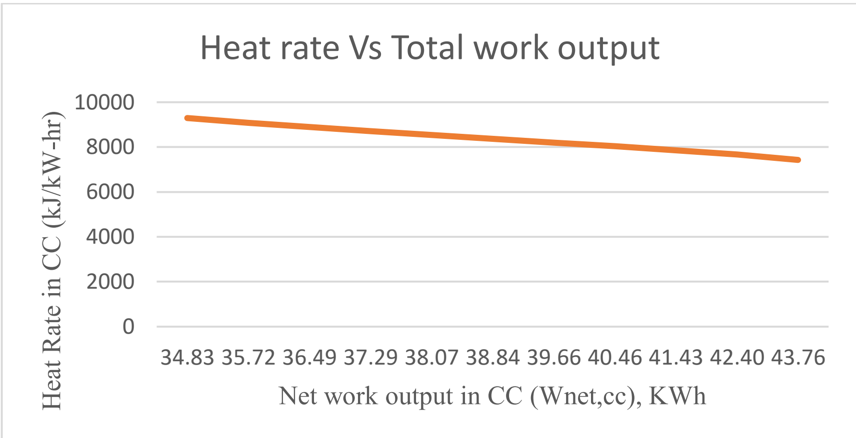

Figure 5.11: Effect of Heat rate with Total work output.

5.2.12 Heat balance of Heat Recovery Steam Generator (HRSG)

The heat balance diagram of HRSG, the feed-water flow measurements, water/steam temperature measurements, water/steam gauge, and differential pressure measurements, exhaust gas temperature measurements, ambient air measurements, exhaust gas pressure measurements, fuel constituent measurements, drum level measurements, exhaust gas oxygen measurements, Etc. is needed.

Feed water mass flow will be measured using a long-radius flow nozzle or orifice under ISO 5167 and ASME-MFC-3M for "zero additional uncertainty." For the calculation of performance test results, water mass flows will be used. The steam mass flow will be used only as secondary data.

Superheated steam temperature measurement points will be less than 2 meters from the test boundary and more than 6m downstream of the stations. For water/steam gauge and differential pressure measurements the devices will be located to minimize vibration to the instruments. The devices will be not being located in areas of extreme temperature.

For exhaust gas temperature measurements, it is necessary to measure each point in the temperature measurement grids. Exhaust gas temperature at stack will be measured using station measurement that is collected from DCS. The absolute pressure transmitter will measure ambient pressure. A psychrometer will measure ambient temperature and relative humidity.

For exhaust gas pressure measurements one measurement port on HRSG inlet duct will be installed to measure exhaust gas pressure drop. The analysis of fuel gas will be measured by on-line chromatograph.

In the event of drum level changes between the start and end of a performance, then the steam mass flow will be calculated from the measured feed water flow and the amount of water added to or removed from the drum, as measured by the drum level change. The drum level of the HP drum & LP drum is taken from the DCS system. Exhaust gas oxygen will be obtained from CEMS at the stack.

Using the above process for taking HRSG operating data at different conditions and states are attached at annexure 1, Where it describes the HRSG operating parameter at a specific load. Those data are chosen here to calculate the performance indicator parameter of HRSG at a different state. Those data are also used to locate the defective state or abnormal condition of HRSG.

With the help of the design condition of the HRSG shown in Annexure 1, it is easy to find out the difference between the design parameter and the HRSG parameter of HRSG. It is also helpful to develop the defective portion of HRSG by identifying the defective point on HRSG.

Using the data shown in annexure 1, the performance indicator parameter of HRSG, i.e., Heat absorption, steam flow rate of HRSG, is calculated. The gas turbine's heat input is 79,612.22 kJ/s, and the heat output to the exhaust of HRSG to the environment is 13.7 kJ/s. There has some heat loss due to some problems in HRSG. The major problem is the leakage of heat pipes into the HRSG. In that case, the water consumption rate of HRSG is high, so the drum level maintenance is costly. Because the treated water is used in the HRSG is costly. Produced steam is also broken out to the environment after HP superheater output because of pipe leakage. The loss of produced steam also harms running the steam turbine at design condition, which is caused to the reduction of load.

The heat absorption and steam flow rate at different conditions and different points are shown in table 5.3.

Table 5.3: HRSG performance indicator parameter.

Content	Parameter	Value	Unit
HPSH	HRSG heat input by GT exhaust gas, Q_f	79,612.22	kJ/s
	Heat absorption, Q_b	11,041.11	kJ/s
	Heat transfer rate, K	204.60	Kj/m^2.h.C
	Steam flow rate, m_{steam}	19.72	kg/s
	Steam Temperature outlet temp, T_o	468.00	°C
	Steam Temperature inlet temp, T_i	286.80	°C
	Steam Outlet Pressure, P_{out}	69	bar
HPEVAP	Heat absorption, Q_b	29,869.11	kJ/s
	Heat transfer rate, K	174.05	Kj/m^2.h.C
	Steam flow rate, m_{steam}	5.48	kg/s
	Steam Temperature outlet temp, T_o	286.80	°C
	Steam Temperature inlet temp, T_i	284.60	°C
	Steam Outlet Pressure, P_{out}	71	bar
	Gas Temp T_g	299.40	°C
HPITECON	Heat absorption, Q_b	10798.33	kJ/s
	Heat transfer rate, K	162.18	Kj/m^2.h.C
	Steam flow rate, m_{steam}	5.48	kg/s
	Steam Temperature outlet temp, T_o	284.6	°C
	Steam Temperature inlet temp, T_i	167.2	°C
	Steam Outlet Pressure, P_{out}	74.5	bar
	Gas Temp T_g	227	°C
LPEVAP	Heat absorption, Q_b	6386.11	kJ/s
	Heat transfer rate, K	163.44	Kj/m^2.h.C
	Steam flow rate, m_{steam}	2.81	kg/s
	Steam Outlet Pressure, P_{out}	4	bar
	Steam Outlet Pressure, P_{in}	9.5	
	Gas Temp T_g	183.1	°C
HPLTECON	Heat absorption, Q_b	5341.11	kJ/s
	Heat transfer rate, K	157.17	Kj/m^2.h.C
	Steam flow rate, m_{steam}	5.48	kg/s
	Steam Outlet Pressure, P_{out}	76.5	bar
	Steam Outlet Pressure, P_{in}	80	
	Gas Temp T_g	146.7	°C
	Steam Temperature outlet temp, T_o	167.2	°C
	Steam Temperature inlet temp, T_i	104	°C

Heat Balance Diagram for Thermal Power Station is the schematic representation of the whole steam cycle from Boiler to High Pressure (HP) Turbines Intermediate Pressure (IP) Turbines and Low Pressure (LP) Turbines to the condenser to pumps to re-heaters and again to the boiler. By using table 5.3, the heat balance diagram of HRSG is drawn.

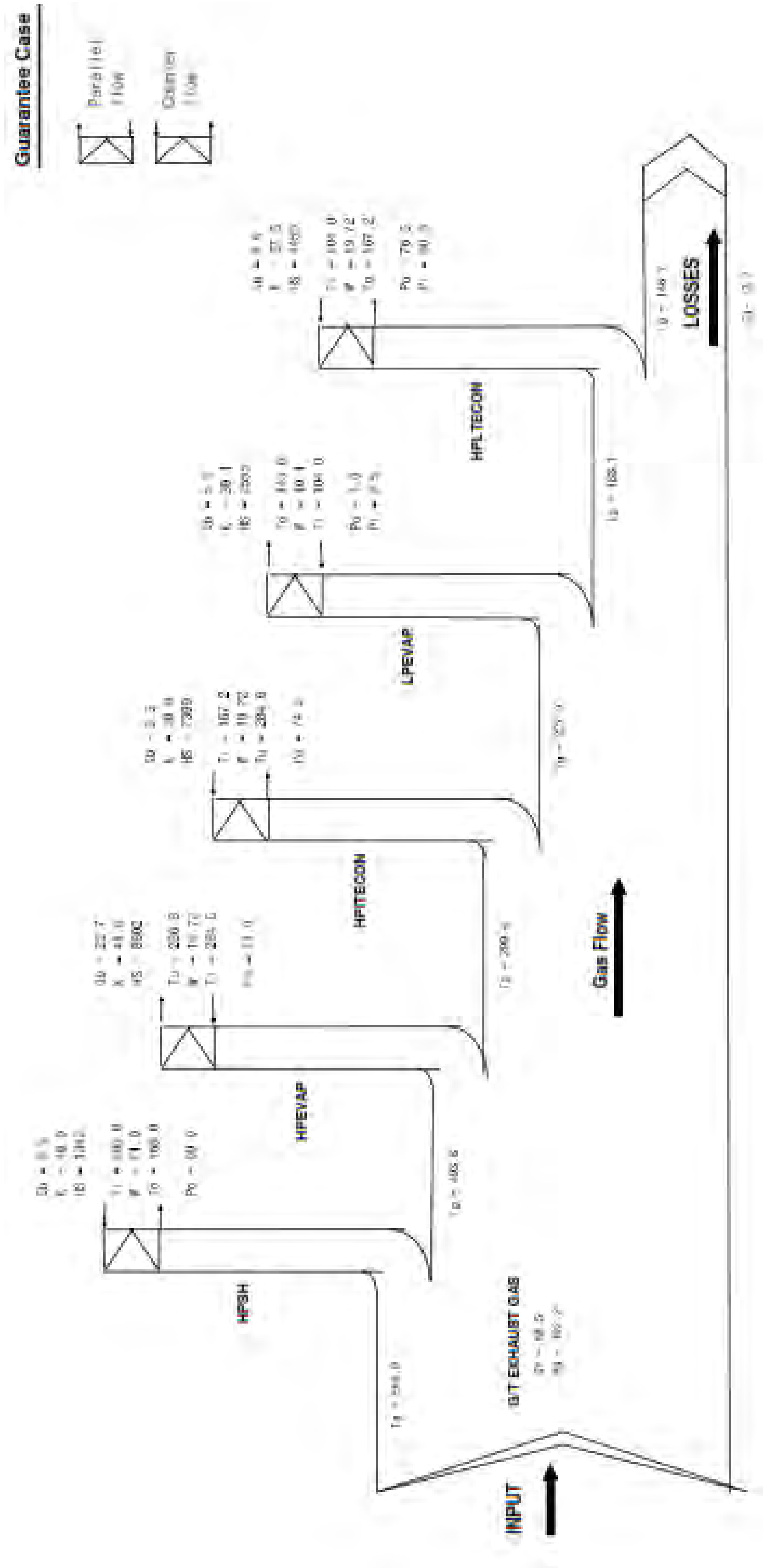

Figure 5.12: Heat balance diagram of HRSG

5.2.13 Service Air Unit (Screw type air Compressor)

The calculated performance parameter of service air unit of the plant at different condition is given below:

Particulars	For Compressor 1	For Compressor 2
% leakage	21%	23%
System leakage quantity, m^3/min	2.53	2.82
Leakage quantity per day, m^3/day	3650	4057
Energy lost due to leakage/day, kwh/day	401	446
Average Energy lost due to leakage/day, kwh/day	424	

Annually energy use of CAS:

Conditions	For Compressor 1	For Compressor 2
Loading (Kwh)	212463	211,302
Unloading (Kwh)	70821	70434
Sub-Total	283,284	281,736
	Total (Compressor 1+ Compressor 2)	**565,020**

Annual electricity cost:

Conditions	For Compressor 1	For Compressor 2
Loading (Tk)	359345.45	268036.3
Unloading (Tk)	15,904	11863.19
Sub-Total	375249.95	279899.49
	Total (Compressor 1+ Compressor 2)	**655,149.44**

Volume of air produced annually:

Particulars	Volume of air produced, m^3	Total volume of air produced annually, m^3	Cost Per, m^3 (Tk/ m^3)	Cost per 10 m^3 (Tk/ m^3)	Cost for store 3 sotage tank (10m3/tank) (Tk/ m^3)	Cost for 1000 m^3 (Tk/ m^3)
For Compressor 1	1902823	1986312	0.33	3	9.9	330
For Compressor 2	83489					

Energy Balance:

Generated heat to produce compressed air		Kwh/day	2256
Losses	Leakage	Kwh/day	423.86
	Waste Heat	Kwh/day	361.01
	Actuation	Kwh/day	92
	Conveying & Others	Kwh/day	1369.13

Service air unit (compressed air) heat flow diagram is given below:

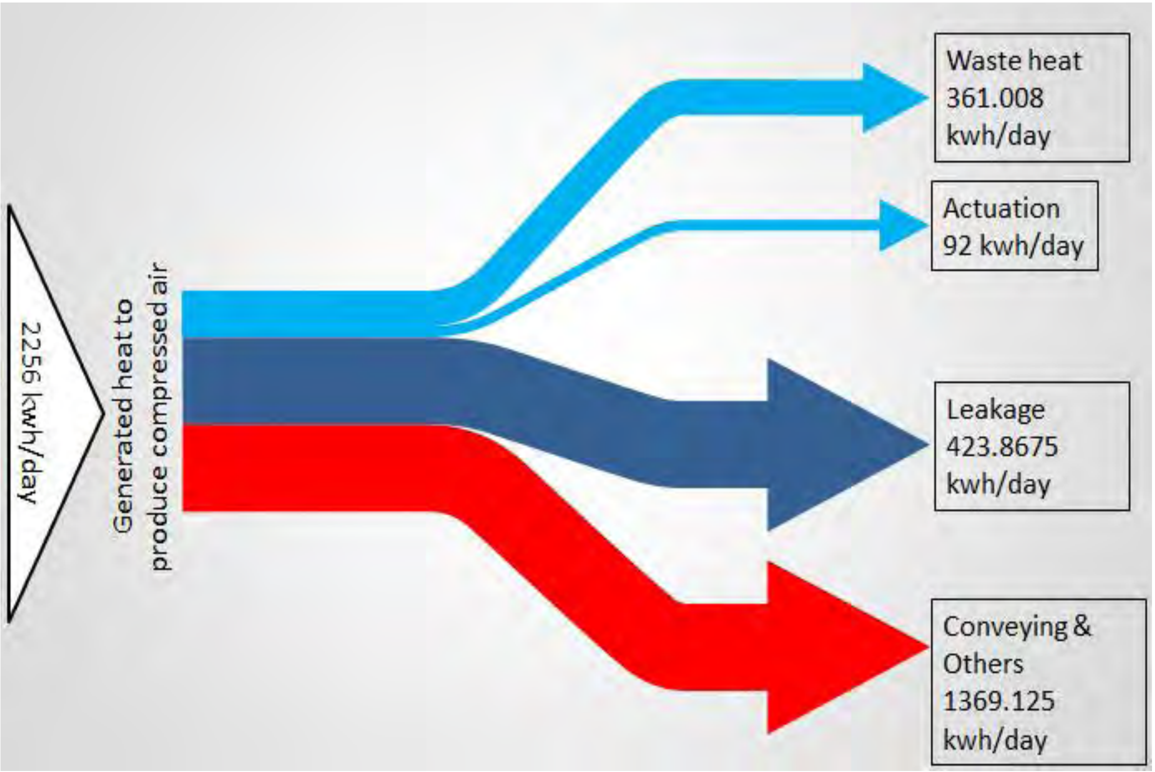

Figure 5.13: Compressed air heat flow diagram.

From the figure 5.26 shows that, the generated heat to produce compressed air is 2256 kWh/day. The percentage of waste heat to the environment is 361.008 kWh/day, for leakage 423.8675 kWh/day, actuation is 92 kWh/day and conveying & others is 1369.125 kWh/day.

5.2.14 Condenser

The key indicators for the condenser performance include the condenser effectiveness (ε), log-mean temperature difference (LMTD) is given in table 5.12.

Table 5.4: Key indicators of Condenser performance

T_{in} °C	T_{out} °C	T_{sat} °C	ΔT °C	Condenser Effictiveness, ε	LMTD	Condenser efficiency (%)
29	41	43	12	0.86	6	86
28	42	43.5	14	0.90	6	90
28.5	40	42	11.5	0.85	6	85
30	41	43	11	0.85	6	85
30.5	39.8	42.5	9.3	0.77	6	78
31	42	43.2	11	0.90	5	90
31.5	41.5	42.6	10	0.90	4	90
31.4	42.58	44	11.18	0.88	5	89
30.6	41.6	42.6	11	0.92	4	92
30.4	39	44	8.6	0.63	9	63
31	39.5	43	8.5	0.71	7	71

5.2.15 Cooling Tower

The operating parameters in the summer and winter seasons are taken from the fenchuganj combined cycle power plant in the summer and winter seasons. By analyzing both the season data, the cost & other performance indicator parameters will also find out.

To evaluate the performance indicator of cooling tower system, its need to calculate the Cooling Tower Approach, Cooling Tower Range, Mass Flow Rate of Water, Heat Loss by Water, Volume of Air Required, Mass of Air required, Heat gain by Air, Effectiveness (ε), Efficiency (η).

The circulating water (CW) Common HDR Temperature (Tw2ºC), Heat Exchanger water temp (Tw1ºC), ambient air temperature (TaºC), Closed Cycle Circulating Water (CCCW) Outlet Pressure, CCCW Exchanger Outlet pressure, Wet bulb temperature is taken from DCS system & manual method. The Circulating water pump's operating parameter, cooling tower fan, and Closed Cycle Circulating water pump in the summer season are given in Annexure 1.

Drift Losses (DL), Windage Losses (WL), Evaporation Losses (EL), and Blow Down Losses (BL) of the cooling tower system is also needed for evaluation of the cooling tower performance.

The various kinds of losses of a cooling tower in the summer season and winter season are given in table 5.5 and table 5.6.

Table 5.5: Various kinds of losses of Cooling tower in summer season

Time Hrs	Tw1 ^0C	Tw2 ^0C	TDS Of CW	TDS Of MU	C.O.C	Mw Kj/hr ($x10^6$)	DL, Kg/hr	WL, Kg/hr	EL, Kg/hr	BL, Kg/hr	Total Loss, Kg/hr
0	38.5	30.1	118	61.3	1.92	10.32	20658.02	51645.04	73749.12	79732.29	225784.5
1	38.2	29.8	128	61	2.09	10.25	20518.43	51296.09	73250.81	66691.04	211756.4
2	38.6	29.5	120	62	1.93	10.06	20122.95	50307.39	77825.53	83192.8	231448.7
3	37.9	29.1	122	62	1.96	10.13	20262.54	50656.34	75781.88	78307.95	225008.7
4	38.1	29.2	114	64	1.78	10.14	20285.8	50714.5	76731.04	98215.73	245947.1
5	38.1	28.5	170	63	2.69	10.03	20076.43	50191.07	81911.82	48228.46	200407.8
6	38.1	28.3	175	62	2.82	99.91	19983.37	49958.43	83230.75	45666.43	198839
7	36.5	28.3	135	64	2.10	99.45	19890.32	49725.8	69064.16	62255.02	200935.3
8	37.8	28.1	145	64	2.26	10.09	20192.75	50481.86	83244.59	65773.5	219692.7
9	38.4	29.2	155	63	2.46	10.13	20262.54	50656.34	79226.51	54252.94	204398.3
10	37.9	29.5	160	63	2.53	10.21	20425.38	51063.45	72918.61	47359.51	191766.9
11	37.8	29.5	148	64	2.31	10.16	20332.33	50830.82	71722.28	54645.55	197531
12	37.3	29.3	150	65	2.30	10.25	20518.43	51296.09	69762.68	53347.93	194925.1
13	37.1	28.1	114	64	1.78	10.36	20727.81	51819.52	79283.86	101483.3	253314.5
14	37.5	27.5	170	63	2.69	10.29	20588.23	51470.56	87499.96	51518.67	211077.4
15	37.8	27.5	175	62	2.82	10.22	20448.64	51121.61	89513.94	49113.84	210198
16	37.8	29.8	135	64	2.10	10.06	20122.95	50307.39	68418.05	61672.6	200521
17	37.3	29.5	145	64	2.26	10.10	20216.01	50540.02	67016.07	52950.97	190723.1
18	38.1	29.1	155	63	2.46	10.29	20588.23	51470.56	78749.96	53926.6	204735.4
19	38.1	29.2	160	63	2.53	10.30	20611.49	51528.72	77962.96	50635.73	200738.9
20	36.9	28.5	175	62	2.82	10.17	20355.59	50888.97	72669.46	39871.74	183785.8
21	36.8	28.3	135	64	2.10	10.29	20588.23	51470.56	74374.96	67042.22	213476
22	37.5	28.3	145	64	2.26	10.24	20495.17	51237.93	79874.81	63110.96	214718.9
23	36.2	28.1	155	63	2.46	10.14	20285.8	50714.5	69833.86	47821.02	188655.2
24	37.8	29.2	120	62	1.93	10.09	20192.75	50481.86	73804.48	78894.45	223373.5
				Total in one day =			**508750.2**	**1271875**	**1907422**	**1555711**	**5243759**

Table 5.6: Various kinds of losses of Cooling tower in winter season

Time, Hrs	Tw1,°C	Tw2,°C	TDS Of CW	TDS Of MU	C.O.C	Mw, Kj/hr (x10^{6})	DL, Kg/hr	WL, Kg/hr	EL, Kg/hr	BL, Kg/hr	Total Loss, Kg/hr
0	37.1	28	300	61.3	4.89	10.02	20053.16	50132.91	77555.61	19916.88	167658.6
1	37.5	28	128	61	2.09	98.75	19750.74	49376.84	79743.6	72602.39	221473.6
2	37.8	27	120	62	1.93	98.28	19657.68	49144.21	90228.77	96451.44	255482.1
3	37.9	27	122	62	1.96	96.54	19308.73	48271.83	89447.7	92429.29	249457.5
4	38.1	26.8	114	64	1.78	96.77	19355.26	48388.14	92953.63	118980.6	279677.7
5	38.1	26	170	63	2.69	96.65	19331.99	48329.99	99414.78	58533.94	225610.7
6	38.1	26.5	175	62	2.82	96.54	19308.73	48271.83	95192.04	52229.26	215001.9
7	36.5	25.8	135	64	2.10	96.77	19355.26	48388.14	88018.04	79340.2	235101.6
8	36.9	25.2	145	64	2.26	96.65	19331.99	48329.99	96128.34	75953.26	239743.6
9	38.4	26.5	155	63	2.46	95.96	19192.41	47981.03	97065.63	66468.86	230707.9
10	36.2	26.8	160	63	2.53	97.70	19541.37	48853.42	78067.76	50703.8	197166.3
11	37.8	27.3	175	62	2.82	96.54	19308.73	48271.83	86165.21	47276.49	201022.3
12	37.3	27.8	135	64	2.10	98.75	19750.74	49376.84	79743.6	71881.56	220752.7
13	37.1	28.5	145	64	2.26	98.28	19657.68	49144.21	71848.83	56769.45	197420.2
14	37.5	29	155	63	2.46	93.63	18727.14	46817.86	67651.8	46326.78	179523.6
15	37.8	29.1	120	62	1.93	93.05	18610.82	46527.06	68813.53	73559.29	207510.7
16	37.8	29.2	122	62	1.96	91.90	18378.19	45945.47	67172.28	69411.36	200907.3
17	37.3	28.5	114	64	1.78	90.72	18145.55	45363.89	67864.37	86866.4	218240.2
18	38.1	28.2	170	63	2.69	96.77	19355.26	48388.14	81437.25	47949.03	197129.7
19	38.1	27.6	175	62	2.82	96.65	19331.99	48329.99	86269.03	47333.45	201264.5
20	36.5	27.1	135	64	2.10	95.96	19192.41	47981.03	76673.69	69114.31	212961.5
21	36.9	26.5	145	64	2.26	96.07	19215.68	48039.19	84933.29	67107.79	219295.9
22	38.4	26.3	155	63	2.46	94.79	18959.78	47399.44	97500.66	66766.75	230626.6
23	36.2	26.5	160	63	2.53	94.21	18843.46	47108.65	77682.16	50453.36	194087.6
24	37.8	26.4	148	64	2.31	94.91	18983.04	47457.6	91972.84	70074.54	228488
				Total in one day=			**480647.8**	**1201620**	**2089544**	**1654501**	**5426312**

Remove corrosion and scale inhibitor in power plant equipment; chemical dosing is essential. To keep the water parameter with designed parameter dosing is also very important. Chemical dosing treated the raw water taken from the river to designed parameter water. The treatment water keeps the plant equipment more efficient and less corrosive. For chemical dosing, the following products are used in this plant. The following table 5.22 shows the total cost of chemical dosing of the plant in a year.

Table 5.7: Cost of chemical dosing for water treatment.

Product & Scale or Used as in CWT	Buckman Product	Dose amount	Unit	Kg/year(365 days)	Price Tk/Kg	Total Tk in one year
Corrosion & Scale Inhibitor	BULAB-9181	10	Kg/day	3650	600	2190000
Dispersant	BULAB-8012	25	Kg/week	1300	620	806000
Non Oxidizing Biocide	BULAB-6057	150	Kg/month	1800	560	1008000
HYPO		300	Kg/week	15600	20	312000
Total:				22350	1800	4316000

For the cooling tower system's operation, some electrical component is also used to keep the plant's cooling system's operation. Circulating water pumps, cooling tower fans, closed cycle circulating water pumps are used to operate the cooling tower system. The rate of electricity that is used for pumps & fans running is 1.2 Tk/Kw. The cost of energy usage of the cooling tower system in table 5.23 below:

Table 5.8: Cost of energy usage of cooling tower system.

	Summer Season		Winter Season	
Pumps & Fan	**Energy Usage for 9 months in Kwh**	**Electricity Cost for 9 months inTk**	**Energy Usage for 3 months in Kwh**	**Electricity Cost for 3 months inTk**
CW Pumps	5283174	6339810	1756213	2107456
Cooling Tower Fans	673233	807880	223860	268632
CCCW Pumps	674898	809878	235845	283014
Total:	**6631306**	**7957567**	**2215918**	**2659102**

So, The annually Energy Usage of cooling tower system = 8847224 kWh

Annually Electricity Cost of cooling tower system = 10616669 Tk

= USD 126,388

Annually Chemical using Cost = 4316000 Tk = USD 51,381

Total Operating Cost of cooling tower system = 14932669 Tk

= USD 177,770

5.3 Comparing with other research works

The Fenchuganj combined cycle power plant's Energetic performance data is used to compare with other power plant data by using other research work. Here several studies on energy and exergy analyses of thermal power plants have been carried out by several researchers (Ameri, Ahmadi, & Hamidi, 2009; Aras & Balli, 2008; Can, Celik, & Dagtekin, 2009; Gorji-Bandpy & Ebrahimian, 2006; Gorji-Bandpy & Goodarzian, 2011; Igbong & Fakorede, 2014; Marzouk & Hanafib, 2013; Mousafarash & Ahmadi, 2014; Mousafarash & Ameri, 2013; Kaviri, Mohd Jafar, Tholudin, & Barzegar Avval, 2011; Sahoo, 2008; Singh & Kaushik, 2014). Once another excellent research works done by S.O. Oyedepo, R.O. Fagbenle, S.S. Adefila, and Md. Mahbub Alam, 2015 on Performance evaluation of selected gas turbine power plants in Nigeria using energy and exergy methods. This study presents a thermodynamic analysis of eleven selected gas turbine power plants' design and performance using the first and second laws of thermodynamics concepts. Here, Energy analysis showed that the combustion chamber and the turbine have the highest energy loss in the plants. Increasing gas turbine inlet temperature (GTIT), the exergy destruction of this component can be reduced. The Energy Performance data of the Fenchuganj Combined Cycle Power Plant & selected gas turbine power plants in Nigeria in the simple cycle are given in Table 5.9:

Table 5.9: Average Operating Data for Selected Gas Turbine Power Plants and Energy Performance Analysis

Plant Name	FCCP (35 MW)	DEL1 (25 MW)	DEL2 (25 MW)	AES1 (33 MW)	AES2 (33 MW)	AES3 (33 MW)	AF1 (75 MW)	AF2 (75 MW)	Algeria (87 MW)	DEL3 (100 MW)	DEL4 (100 MW)	AF3 (138 MW)	AF4 (138 MW)
Installed Rated Power	35	25	25	33.5	33.5	33.5	75	75	87	100	100	138	138
Pressure ratio	8.9	10.91	10.84	9	9.14	9.48	9.38	9.67	10.175	10.68	10.7	9.48	9.48
Mass flow rate of fuel (kg/s)	1.8	3.08	3.1	2.58	2.54	2.81	5.5	6.4	7.275	8.15	8.13	8.1	8.4
Inlet mass flow rate of air (kg/s)	108	140	140	122	122	122	359	359	367	375	375	470	470
Energy input (MW)	90	365	367	298	312	315	558	595	641	691	690	714	704
Power output (MW)	29	19	20	30	29	32	50	58	82	92	93	132	135
Thermal Efficiency in SC(%)	32	24	22	22	20	21	36	40	40	41	41	31	30

Where,

FCCP means the Fenchuganj Combined Cycle Power plant and its installed capacity of 35 MW, located in Fenchuganj, Sylhet, Bangladesh.
AES 1 means AES barges station 1 and its installed capacity of 33 MW, located in Nigeria.
AES 2 means AES barges station 1 and its installed capacity of 33 MW, located in Nigeria.
AF 1 means Afam gas turbine station 1 and its installed capacity of 75 MW, located in Nigeria.
AF 2 means Afam gas turbine station 2 and its installed capacity of 75 MW, located in Nigeria.
AF 3 means Afam gas turbine station 3 and its installed capacity of 138 MW, located in Nigeria.
AF 4 means Afam gas turbine station 4 and its installed capacity of 138 MW, located in Nigeria.
DEL 1 means Delta gas turbine station 1 and its installed capacity of 25 MW, located in Nigeria.
DEL 2 means Delta gas turbine station 2 and its installed capacity of 25 MW, located in Nigeria.
DEL 3 means Delta gas turbine station 3 and its installed capacity of 100 MW, located in Nigeria.
DEL 4 means Delta gas turbine station 4 and its installed capacity of 100 MW, located in Nigeria.

From Table 5.9, The thermal efficiencies of the selected gas turbine plants vary between 20% and 41%. AES 2 unit has the least thermal efficiency (20%), and DEL3, DEL4 unit has the highest efficiency (41%). The results show that over 50% of the selected plants have thermal efficiency below 30% in a simple cycle. This is attributed to the high-energy degradation experienced in different components of the plants. The results also show that about 1.44 to 3.83 MW of thermal energy is continuously discharged from the gas turbine engine to the environment at various stations for each MW of electrical energy produced. The thermal discharge index (TDI) should be as low as possible to improve the plant's efficiency and keep the pollution level low.

The energy losses in air compressors increase at high ambient temperatures. The air compressor work increases as inlet air temperature increases, leading to a decrease in the gas turbine's net-work. Air compressor work can be minimized when the air inlet temperature and mass flow rate are reduced. This shows that the compressor inlet air temperature can manage compressor work.

Another parameter besides the compressor inlet air temperature is the compression ratio-the effect of pressure ratio on air compressor energy loss. An increase in pressure ratio brings about a decrease in energy loss in the air compressor. Moreover, compressor work can be reduced by decreasing the compression ratio.

The heat energy loss in the combustion chamber decrease with an increase in air mass flow rate. This implies that a high mass flow rate of air can minimize the combustion chamber's energy losses as this would introduce more air for combustion. The unburnt air in the combustion chamber serves as a coolant. Therefore, the energy losses decrease as the temperature of the hot gases' mass flow rate is decreased. This is due to the high quantity of air mass flow, which lowers the hot gases' temperature.

For the Combined Cycle Comparison, Garri 2 Combined cycle power plant and Guddu 747 MW Combined cycle power plant located in Pakistan is also considered to compare with Fenchuganj Combined Cycle Power plant. The present deaerated capacity of the Fenchugangj combined cycle is 45 MW; the Overall Energy efficiency in the combined cycle is 48.45%.

The 90 MW Garri 2 Combined cycle power plant, located in Pakistan, and it is a present capacity of 85 MW. The Overall Energy efficiency in the combined cycle is 56.16%, wherein Brayton cycle efficiency is 60.21%, and Rankine cycle efficiency is 52.14%
Another one is, Guddu 747 MW power plant located in Pakistan. The net power output of the combined cycle of Guddu 747 MW power plant is 737.8 MW; overall Energy efficiency in the combined cycle is 59.12%, wherein Brayton cycle efficiency is 62.01%, and Rankine cycle efficiency is 56.38%.

Deciding for plant performance improvement based on energetic performance results may not be sufficient. For complex systems like gas turbine combined cycle plant with multiple components, this may be misleading as quantifying actual losses in the different system control volumes might not be accurately achieved. Only energetic analysis for decision making is lopsided since it does not reveal a detailed plant performance presentation. Therefore, the results obtained from energetic performance analysis should be considered with exergetic analysis, allowing an improved understanding by quantifying the effect of irreversibility occurring in the plant and the locations.

Chapter 6: Exergy Analysis

Thermal energy systems have long been analyzed and evaluated using energy tools; however, in the last two decades, exergy analysis has been increasingly used to evaluate and optimize the thermal system designs for more efficient usage of energy resources. Many researchers currently perform exergy and energy analysis simultaneously to increase the value of the research and provide a complete illustration of energy system characteristics.

Based on the first law of thermodynamics, energy analysis cannot provide sufficient information about the degradation of energy during a process or the quality of the various forms of energy. These limitations can overcome by exergetic analysis.

Exergy analysis allows the identification of irreversibility sources or the inefficiencies in a thermal power system-exergy within a thermal process is consumed due to irreversibilities.

As energy analysis cannot determine the irreversibilities within a thermal system, this successfully demonstrates the importance of exergy analysis, which allows for, and increases the possibility of enhancing thermal systems by using different modifications to reduce the irreversibilities resulting from entropy generation.

This section presents the basic methodology of exergy analysis for the major components of the Fenchuganj combined cycle power plant located at Fenchuganj, Sylhet, Bangladesh.

6.1 Exergy Analysis

Exergy analyses are based on the Second Law of Thermodynamics. According to Kotas, the Second Law of Thermodynamics is required to establish the difference in quality between thermal and mechanical energy and indicate the directions of spontaneous processes.

A general exergy-balance equation applicable to any thermal system component may be formulated using the first and second thermodynamics laws [Oh et al., 1996; Ebadi and Gorji-Bandpy, 2005].

The total exergy of a system E divided into four components: physical exergy, kinetic exergy, potential exergy and chemical exergy:

$$E = E^{PH}+E^{CH}+E^{PT}+E^{KN} \quad 6.1$$

Above equation expressed on the unit-of-mass basis:

$$e = e^{PH}+e^{CH}+e^{PT}+e^{KN} \quad 6.2$$

Kinetic exergy and potential exergy are as follows:

$$e^{KN} = \frac{1}{2}\, V^2 \quad 6.3$$

$$e^{PT} = g.z \quad 6.4$$

In a steady process, when the system is at rest relative to the environment, there is no change in the height and speed. Therefore, kinetic and potential exergies are negligible. In this case, physical exergy, also known as thermodynamic exergy, would be defined as the maximum theoretical useful work. The maximum theoretical useful work is the amount of work obtained from the interaction between a system and its environment until it reaches the dead state.

The physical exergy for ideal gas stream is defined by the following:

$$e^T = C_P\, [(T- T_o) - T_o\, \ln \frac{T}{T_o})] \quad 6.5$$

$$e^P = RT_o \ln \frac{P}{P_o} \quad 6.6$$

$$e^{PH} = e^T + e^P \quad 6.7$$

and for the simple compressible pure substances the physical exergy is given by:

$$e^{PH} = (h-h_o) - T_o\, (S - S_o) \quad 6.8$$

Where;

P_o and T_o= The pressure and temperature, respectively, at standard state;

R = The gas constant

C_P = The specific heat at constant pressure

The chemical exergy is associated with the departure of a system's chemical composition from its chemical equilibrium. The chemical exergy is an essential part of exergy in the combustion process.

For a hydrocarbon fuel, C_aH_a standard chemical exergy tables may be used. The most accurate relation for chemical exergy of that fuel can be defined as below:

$$\bar{e}_F^{CH} = [\bar{h}_f + (a+\frac{b}{4}).\,\bar{h}_{o2} - a.\,\bar{h}_{co2} - \frac{b}{2}\,.\,\bar{h}_{H2o(g)}]\,(T_o,P_o) - T_o\,[\bar{S}_F + (a+\frac{b}{4}).\,\bar{S}_{o2} - a.\,\bar{S}_{co2} - \frac{b}{2}\,.\,\bar{S}_{H2o(g)}\,]\,(T_o,P_o) + R\,.\,T_o.\,\ln\,[\frac{(X_{o,O2})^{a+b/4}}{(X_{o,CO2})^a(X_{o,H2O})^{b/2}}] \quad 6.9$$

An approximate formulation for chemical exergy of gaseous hydrocarbon fuels is given as:

$$\frac{e_F^{CH}}{LHV} = 1.033 + 0.0169\,(b/a) - 0.0698/a \quad 6.10$$

For methane CH_4, a = 1 and b = 4.

The exergy rate is defined as by following:

$$E_x = \dot{m}.e_x \quad 6.11$$

Where, ṁ is the mass flow rate of the working fluid.

In steady state, exergy balance for control volume is given as [Bejan et al., 1996; Kotas, 1995]:

$$E_x = \sum_j (1 - \frac{T_o}{T_j})\,Q_j + W_{CV} + \sum_i m_i e_i - \sum_e m_e e_e \quad 6.12$$

The subscripts i, e, j and 0 refer to conditions at inlet and exits of control volume boundaries and reference state. Above equation can be written as:

$$E_i^{tot} - E_e^{tot} - E_D = 0 \quad 6.13$$

The above equation implies that 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 resulting from irreversibilities.

6.2 Exergy Components and Exergy Balances

The plant is separated into eight subsystems. Exergy inflows and outflows to these subsystems, exergy destruction in these subsystems and exergy losses to the
environment is calculated. Schematic of these subsystems and exergy balances for them may be found in following equations of air compressor, combustion chamber, gas turbine, steam turbine, HRSG, Condenser, condensate pump and related junction, Boiler Feed Water Pump. Every state of the power plant and the following equations are written below with respect to the figure 5.1. Where; T mentioned as Temperature, P means pressure, S means entropy, E means the exergy rate, Ed means exergy destruction rate and ε means exergetic efficiency.

The exergy-balance equations and the exergy destroyed during the process taking place in each component of the power plant are expressed in following tables:

Table 6.1: Exergy equations of major component of the plant.

SL No	Component	Equations	
1	Air Compressor	$E_{WAC} = (\dot{E}_1^T - \dot{E}_2^T) + (\dot{E}_1^P - \dot{E}_2^P) + T_o\,(\dot{S}_2 - \dot{S}_1)$	6.14
		$E_{dest} = T_o\,(\dot{S}_2 - \dot{S}_1)$	6.15
2	Combustion Chamber	$E_{CHE} + (\dot{E}_2^T + \dot{E}_5^T - \dot{E}_3^T) + (\dot{E}_2^P + \dot{E}_5^P - \dot{E}_3^P) + T_o\,(\dot{S}_3 + \dot{S}_5 - \dot{S}_2 + \frac{Q_{cc}}{T_o})$	6.16
		$E_{dest} = (\dot{S}_3 - \dot{S}_2 + \dot{S}_5 + \frac{Q_{cc}}{T_o})$	6.17
3	Gas Turbine	$E_{WGT} = (\dot{E}_3^T - \dot{E}_4^T) + (\dot{E}_3^P - \dot{E}_4^P) + T_o\,(\dot{S}_3 - \dot{S}_4)$	6.18
		$E_{dest} = T_o\,(\dot{S}_3 - \dot{S}_4)$	6.19
4	HRSG: (including dearator)	$E_{dest} = (E_4 - E_8) + (E_{23} - E_9) + (E_{18} - E_{19})$	6.20
		$\varepsilon = \frac{(E9 - E23) + (E19 - E18)}{(E4 - E8)}$	6.21

SL No	Component	Equations	
5	Steam Turbine	$E_{dest} = (E_9 – E_{10}) – E_{w4}$	6.22
		$\varepsilon = \frac{E_4}{(E_9 - E_{10})}$	6.23
6	Condenser, condensate pump and related junction	$E_{dest} = E_{14} – E_{16} + E_{w3} + E_{w4} + E_{w5} + E_{25} – E_{24} + E_I - E_J$	6.24
		$\varepsilon = 1 - \frac{E_{dest}}{(E_{19} - E_{15})}$	6.25
7	Boiler Feed Water Pump (BFW)	In = Out + Destruction $E_{19} = E_{20} + E_{w6} + E_{dest}$ $E_{dest} = E_{19} - E_{20} + E_{w6}$	6.26
		$\varepsilon = 1 – (\frac{E_{dest}}{E_{w4}})$	6.27

6.3 Exergetic efficiency

In defining the exergetic efficiency, it is necessary to identify both a product and a fuel for the thermodynamic system being analyzed. The product represents the desired result produced by the system. Accordingly, the definition of the product must be consistent with the purpose of purchasing and using the system. The fuel represents the resources expended to generate the product, and is not necessarily restricted to being an actual fuel such as natural gas, oil or coal. Both the product and fuels are expressed in terms of exergy.

The exergy efficiency of the plant and its component is

$$\varepsilon = \frac{usefule\ exergy}{exergy\ supplied} = 1 - \frac{Eaxergy\ destroyed}{Exergy\ supplied\ by\ the\ unit} \qquad 6.28$$

Overall Plant efficiency:

$$`\varepsilon_{overall} = 1 - \frac{E_{d,plant}}{E^{CHE}} \qquad 6.29$$

Overall exergetic efficiency of the system can be defined in terms of exergy destruction ratio.

$$\varepsilon_{overall} = 1 - \sum E_d \qquad 6.30$$

Exergetic destruction efficiency:

$$\varepsilon_d = \frac{E_{D,component}}{E^{CHE}} \qquad 6.31$$

6.4 Result and Discussion

For exergy analysis of the Fenchuganj Combined cycle power plant, data were taken at a different time to measure the performance in different conditions attached in annexure 1 in detail.

For analyzing the plant's exergy, the equations are shown in table 6.1, and the operating parameter collected from the plant is used. The following operating parameters are shown at every stage of the power plant schematic diagram shown in Figure 3.14.

Following table 6.1 and 6.2 shows the Exergy rate of the Gas cycle and steam cycle, which is used to evaluate the exergetic performance indicator calculation:

Table 6.2: Exergy rate at different point of Gas cycle.

State	Substance	P	T	mass flow rate	Cp	R	ex	E
		(bar)	°C	(kg/s)	kJ/kg.k	kJ/kg.k	kJ/kg	kW
Ref	Air	1.013	25		1.005	0.287		
1	Air	1.013	28	108.22	1.005	0.287	0.0000	-
2	Air	9	350	108.22	1.005	0.287	292.58	31,663.05
3	Comb. Gas	8.8	1105	109.999	1.148	0.29	902.80	99,307.50
4	Comb. Gas	1.039	578	109.999	1.148	0.29	278.05	30,586.10
5	Natural Gas	22.8	42	1.779	2.34	0.5	52,331.65	93,098.01
8	Comb. Gas	1.039	131	109.998	1.148	0.29	19.67	2,164.57

Table 6.3: Exergy rate at different point of steam cycle.

State	Substance	P	T	h	s	m	ex	E
		(bar)	°C	(kj/kg)	(kJ/kg.k)	(kg/s)	kJ/kg	kW
9	Water	69	468	3,329.00	6.96	15.30	1,260.61	19,287.31
10	Water	0.145	53	2,597.07	8.02	0.27	212.08	57.45
11	Water	0.145	53	2,597.07	8.02	0.27	212.08	57.45
12	Water	0.145	53	2,597.07	8.02	0.27	212.08	57.45
13	Water	13	192	2,787.60	6.50	4.17	856.50	3,571.59
14	Water	0.145	53	2,597.07	8.02	0.07	212.08	15.04
15	Water	0.145	53	2,597.07	8.02	0.07	212.08	15.04
16	Water	6.5	54	684.21	1.96	0.07	104.03	7.38
17	Water	6.5	32	134.83	0.47	4.17	0.28	1.17
18	Water	6	37	153.26	0.53	11.26	0.86	9.65

State	Substance	P	T	h	s	m	ex	E
		(bar)	°C	(kj/kg)	(kJ/kg.k)	(kg/s)	kJ/kg	kW
19	Water	1.7	115	483.01	1.47	11.26	48.08	541.33
20	Water	46	104	439.38	1.35	9.53	42.13	401.49
21	Water	16	104	437.06	1.35	0.35	38.56	13.49
22	Water	16	98	411.93	1.28	0.35	34.65	12.13
23	Water	46	104	439.38	1.35	9.53	42.13	401.49
24	Water	3.9	31	130.27	0.45	6.02	0.55	3.29
25	Water	3.9	38	159.52	0.55	6.02	1.46	8.78
26	Water	1.7	101	423.33	1.32	1.68	34.81	58.49

The exergetic rate of the gas cycle and steam cycle are used to calculate the exergy destruction, exergetic efficiency, exergetic destruction efficiency of air compressor, combustion chamber, gas turbine, steam turbine, HRSG, Condenser, condensate pump and related junction, Boiler Feed Water Pump which is shown in table 6.3.

Table 6.4: Exergetic performance parameter of combined cycle power plant.

Component	Ed (kW)	ε%	εd%	% of Ed
Air Compressor	3,359.66	90.41	3.61	6.85
Combustion Chamber	25,453.56	79.60	27.34	51.93
Gas Turbine	2,172.45	96.84	2.33	4.43
HRSG	9,004.04	68.32	11.31	21.48
Steam turbine	4,009.42	79.15	5.04	9.56
Condenser	492.13	6.49	0.62	1.17
BFWP	63.90	2.99	0.08	0.15
Exhaust Loss	2164.57	0.00	2.33	4.42

The maximum exergy destruction occurs in the combustion chamber of, and, i.e., 25,453.56 kW and the lowest exergy destruction happens in BFWP and, i.e., is 63.90 kW shown in table 6.4. The maximum exergetic efficiency occurs in gas turbine and, i.e., is 96.84%.

The maximum exergy destruction rate and percentage of exergy destruction in the combustion chamber can be reduced by installing a high-quality combustion chamber or changing the spare parts (spark plug, fusible plug, connecting pipe, Etc.) of the combustion chamber to increase the efficiency of the combustion chamber.

From the above table 6.4, the following figures show here to represent the exergetic performance of the different components in different conditions graphically.

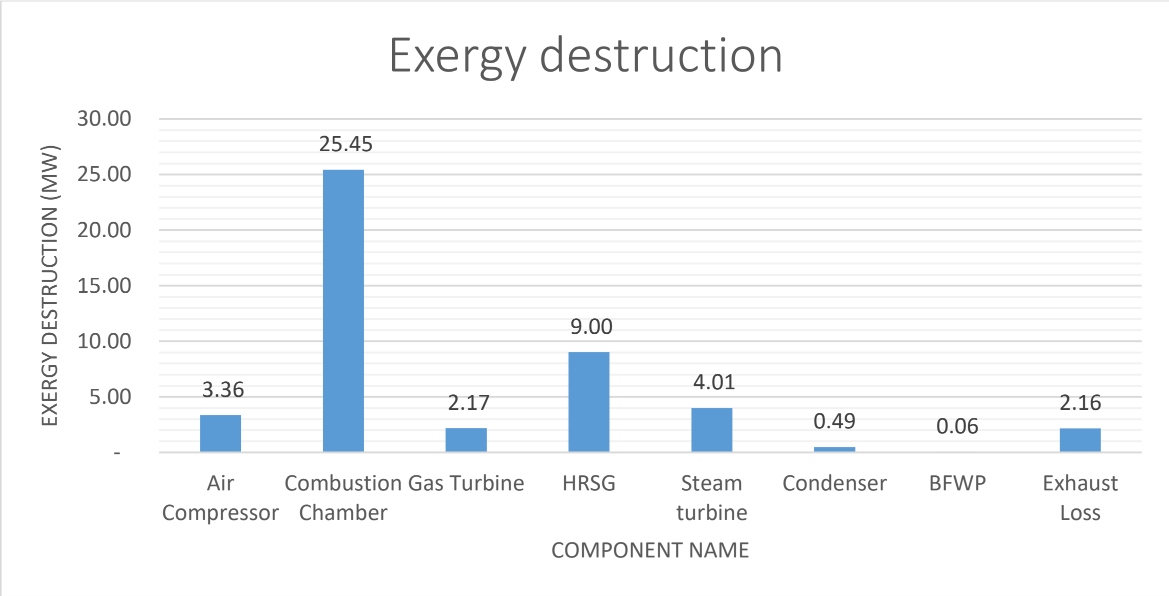

Figure 6.1: Exergy destruction in various components.

From the figure 6.1 the maximum exergy destruction occurs in combustion chamber. and i.e. 25.45 MW and the lowest exergy destruction happens in BFWP and i.e. is 0.063 MW.

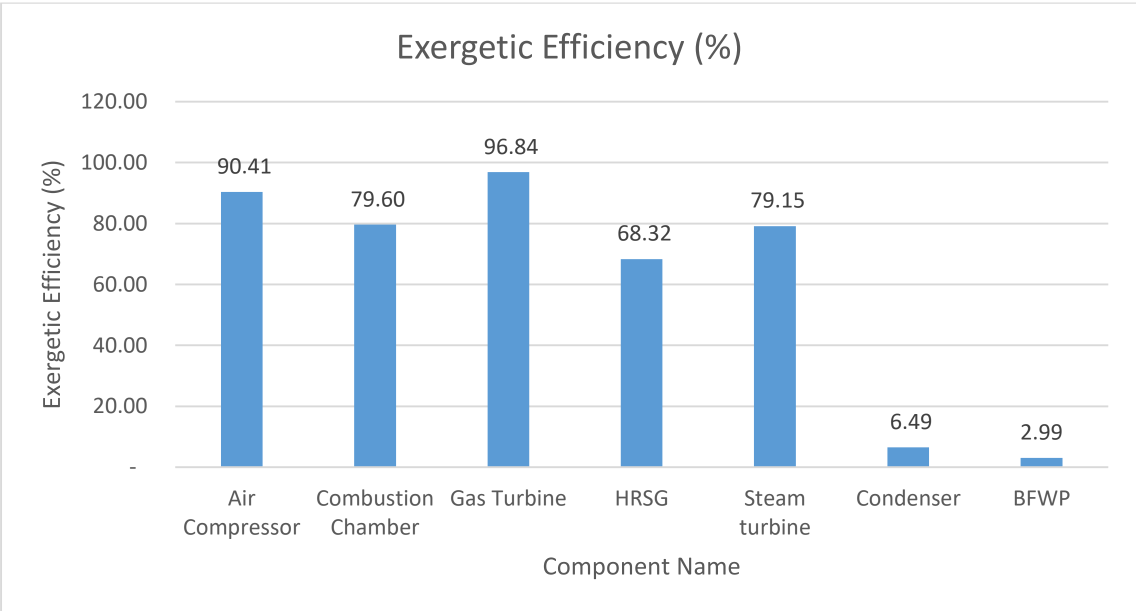

Figure 6.2: Exergetic efficiency of various components.

From the figure 6.2 the maximum exergetic efficiency occur in gas turbine, i.e. 96.84%.

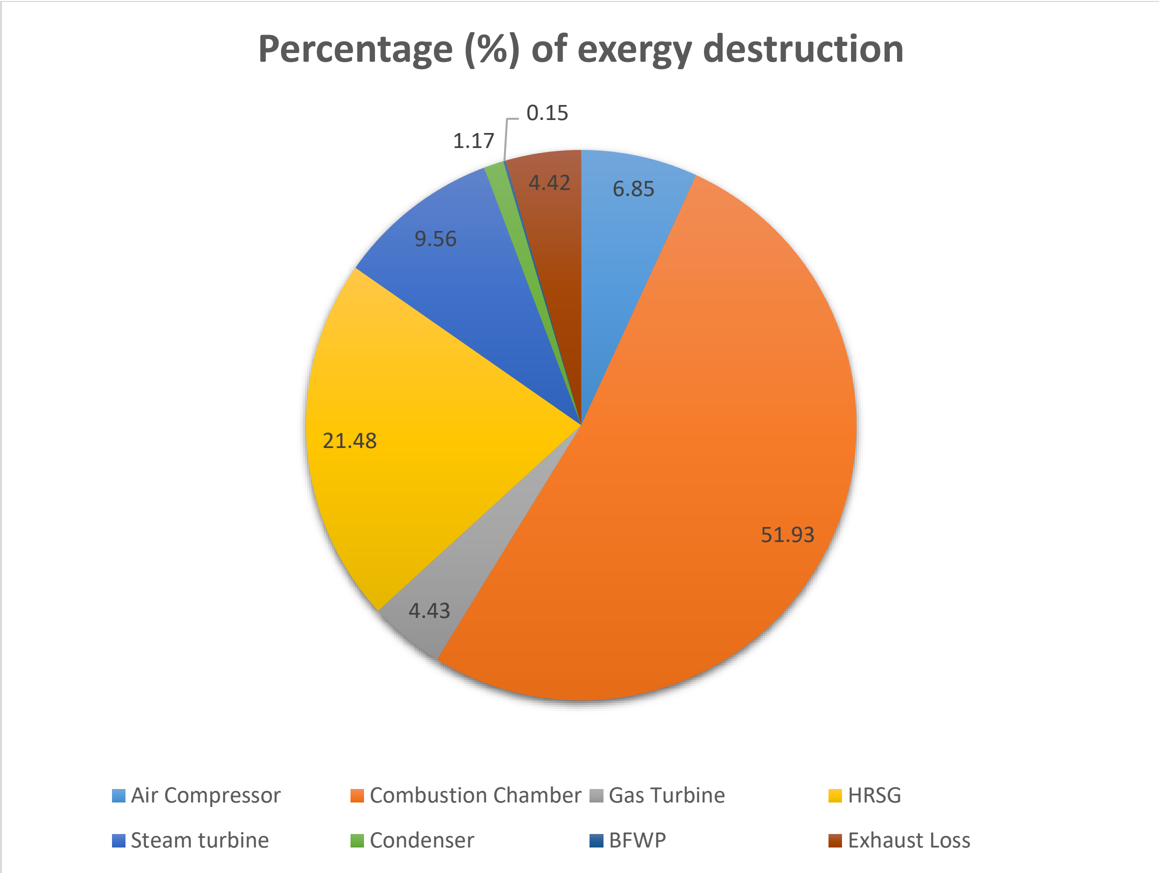

Figure 6.3: Percentage of exergetic destruction rate of power plant.

From the figure 6.4 the percentage of exergy destruction of the power plant shown graphically. In which the maximum exergy destruction is 51.93% in combustion chamber all over the system of the plant.

Total Plant Overall exergetic efficiency = 0.4734 = 47.35%.

6.5 Comparing with other research works

For comparing the Fenchuganj combined cycle power plant data with other power plants by using other research work, here several studies on energy, exergy, and exergy costing analyses of thermal power plants have been carried out by several researchers (Ameri, Ahmadi, & Hamidi, 2009; Aras & Balli, 2008; Can, Celik, & Dagtekin, 2009; Gorji-Bandpy & Ebrahimian, 2006; Gorji-Bandpy & Goodarzian, 2011; Igbong & Fakorede, 2014; Marzouk & Hanafib, 2013; Mousafarash & Ahmadi, 2014; Mousafarash & Ameri, 2013; Kaviri, Mohd Jafar, Tholudin, & Barzegar Avval, 2011; Sahoo, 2008; Singh & Kaushik, 2014).

Once another excellent research works done by S.O. Oyedepo, R.O. Fagbenle, S.S. Adefila, and Md. Mahbub Alam, 2015 on Performance evaluation of selected gas turbine power plants in Nigeria using energy and exergy methods. This study presents a thermodynamic analysis of eleven selected gas turbine power plants' design and performance using the first and second laws of thermodynamics concepts. Here, Energy analysis showed that the combustion chamber and the turbine have the highest proportion of energy loss in the plants. Increasing gas turbine inlet temperature (GTIT), the exergy destruction of this component can be reduced.

An exergy balance for the components of the gas turbine plants and of the overall plants is at this point performed and the net exergy flow rates crossing the boundary of each component of the plants, together with the exergy destruction, exergy defect and exergy efficiency in each component are calculated and are presented in Table 6.5 to Table 6.7 for simple cycle operation. The two most important performance criteria are exergy efficiency and exergy destruction rate.

The exergy Performance data of the Fenchuganj Combined Cycle Power Plant & selected gas turbine power plants in Nigeria and Algeria in the simple cycle are given in Table 6.5:

Table 6.5 Exergy Destruction Rate in different power plants in simple cycle operation.

Plant Name	FCCP (35 MW)	DEL1 (25 MW)	DEL2 (25 MW)	AES1 (33 MW)	AES2 (33 MW)	AES3 (33 MW)	AF1 (75 MW)	AF2 (75 MW)	Algeria (87 MW)	DEL3 (100 MW)	DEL4 (100 MW)	AF3 (138 MW)	AF4 (138 MW)
Component	**Ed (MW)**	**Ed (MW)**	**Ed (MW)**	**Ed (MW)**	**Ed (MW)**	**Ed (MW)**	**Ed (MW)**	**Ed (MW)**	**Ed (MW)**	**Ed (MW)**	**Ed (MW)**	**Ed (MW)**	**Ed (MW)**
AC	3.36	3.14	3.63	4.69	4.98	5.64	8.62	8.09	6.94	13.36	12.48	13.14	14.8
CC	25.45	62.52	61.76	56.55	56.58	55.35	139.42	159.84	39.73	171.84	173.33	176.78	180.83
GT	2.17	0.39	0.14	0.29	0.52	0.23	5.99	1.47	6.84	0.7	1.8	9.39	14.5
Total	30.98	66.05	65.53	61.53	62.08	61.22	154.03	169.4	53.51	185.9	187.61	199.31	210.13

Where,

FCCP means the Fenchuganj Combined Cycle Power plant and its installed capacity of 35 MW, located in Fenchuganj, Sylhet, Bangladesh.

AES 1 means AES barges station 1 and its installed capacity of 33 MW, located in Nigeria.

AES 2 means AES barges station 1 and its installed capacity of 33 MW, located in Nigeria.

AF 1 means Afam gas turbine station 1 and its installed capacity of 75 MW, located in Nigeria.

AF 2 means Afam gas turbine station 2 and its installed capacity of 75 MW, located in Nigeria.

AF 3 means Afam gas turbine station 3 and its installed capacity of 138 MW, located in Nigeria.

AF 4 means Afam gas turbine station 4 and its installed capacity of 138 MW, located in Nigeria.

DEL 1 means Delta gas turbine station 1 and its installed capacity of 25 MW, located in Nigeria.
DEL 2 means Delta gas turbine station 2 and its installed capacity of 25 MW, located in Nigeria.
DEL 3 means Delta gas turbine station 3 and its installed capacity of 100 MW, located in Nigeria.
DEL 4 means Delta gas turbine station 4 and its installed capacity of 100 MW, located in Nigeria.
Algeria means Algeria 87 MW gas turbine power station and its installed capacity 87 MW, Located in Algeria.

Table 6.5 shows the different plant exergy destruction rate in simple cycle operation only. The total exergy destruction rate of the selected gas turbine plants varies between 30.98 MW to 210.13 MW. FCCP unit has the least total exergy destruction rate (30.98 MW), and AF4 unit has the highest total exergy destruction rate (210.13 MW). But, the component-wise exergy destruction is different in different plants. Table 6.5 also showed that the lowest exergy destruction rate in the air compressor happens in the DEL1 unit and the highest exergy destruction rate in the AF4 unit. The least exergy destruction rate in the combustion chamber happens in the FCCP unit and the highest exergy destruction rate in the AF4 unit. The least exergy destruction rate in gas turbine happens in DEL2 unit and highest exergy destruction rate in AF4 unit. Table 6.2 represents that, above three components of simple cycle operation, the maximum exergy destruction in a simple cycle occurs in the combustion chamber.

Table 6.6 Exergetic efficiency in different power plants in simple cycle operation.

Plant Name	FCCP (35 MW)	DEL1 (25 MW)	DEL2 (25 MW)	AES1 (33 MW)	AES2 (33 MW)	AES3 (33 MW)	AF1 (75 MW)	AF2 (75 MW)	Algeria (87 MW)	DEL3 (100 MW)	DEL4 (100 MW)	AF3 (138 MW)	AF4 (138 MW)
Component	**ε%**	**ε%**	**ε%**	**ε%**	**ε%**	**ε%**	**ε%**	**ε%**	**ε%**	**ε%**	**ε%**	**ε%**	**ε%**
AC	90.41	95.21	90.95	85.99	85.17	83.17	90.85	91.57	84.19	87.48	88.31	87.54	85.97
CC	79.6	77.25	77.69	74.36	75.95	76.71	57.49	56	75.91	61.05	60.63	61.5	59.73
GT	96.84	99.57	99.85	99.62	99.32	99.67	96.86	99.32	92.58	99.71	99.25	95.75	93.09

Table 6.6 shows the different plant exergetic efficiency in simple cycle operation only. In component-wise exergetic efficiency is different in different plants. Table 6.6 also showed that the lowest exergetic efficiency in the air compressor happens in the AES3 unit and the highest exergetic efficiency in the DEL1 unit. The least exergetic efficiency in the combustion chamber happens in the AF2 unit and the highest exergetic efficiency in the FCCP unit. The least exergetic efficiency in gas turbine happens in Algeria unit and highest exergy destruction rate in DEL2 unit. Table 6.6 also represents the maximum exergetic efficiency found in a gas turbine in a simple cycle and the minimum exergetic efficiency

found in the combustion chamber. A higher temperature difference occurs in the combustion chamber, which causes maximum thermal efficiency losses in the combustion chamber.

Table 6.7 Percentage of exergy destruction in different power plants in simple cycle operation.

Plant Name	FCCP (35 MW)	DEL1 (25 MW)	DEL2 (25 MW)	AES1 (33 MW)	AES2 (33 MW)	AES3 (33 MW)	AF1 (75 MW)	AF2 (75 MW)	Algeria (87 MW)	DEL3 (100 MW)	DEL4 (100 MW)	AF3 (138 MW)	AF4 (138 MW)
Component	**% of Ed**	**% of Ed**	**% of Ed**	**% of Ed**	**% of Ed**	**% of Ed**	**% of Ed**	**% of Ed**	**% of Ed**	**% of Ed**	**% of Ed**	**% of Ed**	**% of Ed**
AC	10.84	4.75	5.54	7.62	8.03	9.21	5.59	4.78	12.96	7.19	6.65	6.6	7.04
CC	82.15	94.67	94.25	91.9	91.13	90.39	90.51	94.36	75.24	92.43	92.39	88.7	86.05
GT	7.01	0.58	0.21	0.48	0.84	0.41	3.89	0.87	12.8	0.38	0.96	4.71	6.9

Table 6.7 represents the Percentage of exergy destruction is maximum in simple cycle operation is found in the combustion chamber. A higher temperature difference occurs in the combustion chamber, which causes maximum thermal efficiency losses in the combustion chamber. The heat energy loss in the combustion chamber decreases with an increase in air mass flow rate. This implies that a high mass flow rate of air can minimize the combustion chamber's energy losses as this would introduce more air for combustion. The unburnt air in the combustion chamber serves as a coolant. For this purpose, an additional air compressor booster unit can be introduced for additional mass flow of air into the combustion chamber. Therefore, the energy losses decrease as the temperature of the hot gases' mass flow rate decreases.

Table 6.8 Exergy Destruction Rate in different power plants in Combined cycle operation.

Component	FCCP (35 MW)	Ataer PP (50 MW)	Guddu (747 MW)	Garri 2 (90 MW)
	Ed (MW)	**Ed (MW)**	**Ed (MW)**	**Ed (MW)**
Air Compressor	3.4	3.93	60.5	9.29
Combustion Chamber	25.45	69.9	683.12	154.27
Gas Turbine	2.2	13.1	68.87	33.34
HRSG	9.01	7.9	89.75	15.1
Steam turbine	4.01	6.01	18.65	11.58
Condenser	0.51	1.3	13	5.67
BFWP	0.10	0.7	Non	Non
Stacks	2.20	2.1	Non	15.27

Where,

FCCP means the Fenchuganj Combined Cycle Power plant and its installed capacity of 35 MW, located in Fenchuganj, Sylhet, Bangladesh.

Ataer PP means Ataer Power station and its installed capacity of 50 MW, located in Nigeria.

Guddu means Guddu 747 MW combined cycle power station and its installed capacity of 747 MW, located in Pakistan.

Garri 2 means Garri 90 MW combined cycle power station 2 and its installed capacity of 90 MW, located in Pakistan.

Table 6.8 shows the different plant exergy destruction rate in combined cycle operation only. The component-wise exergy destruction is different in different plants. Table 6.8 also showed that the highest exergy destruction occurs in combustion chamber in Brayton cycle, then second highest exergy destruction occurs in HRSG in Rankine cycle.

Table 6.9 Percentage of exergy destruction in different power plants in Combined cycle operation.

Component	FCCP (35 MW)	Ataer PP (50 MW)	Guddu (747MW)	Garri 2 (90MW)
	% of Ed	**% of Ed**	**% of Ed**	**% of Ed**
Air Compressor	7	4	6	4
Combustion Chamber	52	70	73	63
Gas Turbine	4	13	7	14
HRSG	22	10	10	6
Steam turbine	10	6	2	6
Condenser	1	1	1	5
BFWP	0.15	0.0	Non	Non
Stacks	4	2	Non	2

Table 6.9 shows that the maximum Percentage of exergy destruction occurs in the combustion chamber, and its values vary from 52% to 73%. The maximum exergy destruction in the combustion chamber found in the Guddu unit, and the second-highest exergy destruction found in the FCCP unit is 22%.

Table 6.10 shows the maximum exergetic efficiency found in the gas turbine of the FCCP unit and the second-highest exergy destruction found in the Air compressor unit of the Ataer PP unit. The exergetic efficiency values varied from 96.84% to 81.7% and 1.32–2.01, respectively, for the considered plants.

Table 6.10 Exergetic Efficiency in different power plants in Combined cycle operation.

Component	FCCP (35 MW)	Ataer PP (50 MW)	Guddu (747MW)	Garri 2 (90MW)
	ε%	ε%	ε%	ε%
Air Compressor	90.41	94.9	92.1	95
Combustion Chamber	79.6	64	73	76
Gas Turbine	96.84	81.7	89.48	90
HRSG	68.32	71	82.6	89.8
Steam turbine	79.15	81.1	91.19	79.9
Condenser	6.49	85	93.04	Non
BFWP	2.99	0	Non	Non
Stacks	0	Non	Non	Non

For both simple and combined cycle operational modes, the combustion chamber is the primary source of thermodynamic inefficiency in the plants considered due to the irreversibility of combustion and the large temperature difference between the air entering the combustion chamber and the flame temperature. These immense losses mean that a large amount of energy present in the fuel can be wasted with the remarkable capacity to generate useful work. The variations in the plants' performance may be ascribed to inadequate maintenance procedures, faulty components, and discrepancies in operating data.

By comparing the above table's information, in combined cycle operation, the overall exergetic efficiency in combined cycle varies from 47% to 65%, where overall energetic analysis showed previously on 48 to 62%. Using only energetic analysis for decision making is lopsided as it does not reveal the detailed presentation of plant performance. Therefore, the result obtained from energy considerations should be considered along with those from exergy analysis. This allows an improved understanding by quantifying the effect of irreversibility occurring in the plant and its locations.

The component with the highest exergy improvement potential is the combustion chamber. This high improvement potential in the combustion chamber is due to the irreversibility of combustion and the large temperature difference between the air entering the combustion chamber and the flame temperature. These immense losses mean that a large amount of energy present in the fuel to generate useful work is wasted. Exergy improvement potential can be afforded in the combustion chamber by preheating the reactants and reducing the heat loss and the excess air entering the combustion chamber. The lower improvement potential in the air compressor, compared with the combustion chamber, is due to relatively heat loss from the air compressor through friction compared to the large temperature difference between the air entering the combustion chamber and the flame temperature. These results have made it possible to determine the critical points of the gas turbine system stating hierarchy on its components so that the measure is applied in the places where they will be most effective.

Also, the turbine component's exergy efficiency and the plant's overall exergetic efficiency decreased with increased ambient temperature, whereas the compressor and turbine's exergy efficiencies increased with increased ambient temperature. The overall exergetic efficiency decreased from 18.53 to 17.26% for the ambient temperature range of 294 – 310K. It was found that a 5 K rise in ambient temperature resulted in a 1.03% decrease in the overall exergetic efficiency of the plant. The low overall exergetic efficiency is due to extensive exergy destruction in the combustion chamber

The exergetic efficiency (or second law efficiency) of the plant was also found to depend significantly on a change in turbine inlet temperature. The above table shows that the plant's second-law efficiency increases steadily as the turbine inlet temperature increases. The turbine material temperature limits the increase in exergetic efficiency with an increase in turbine inlet temperature. This can be seen from the plant efficiency defect curve. As the turbine inlet temperature increases, the plant efficiency defect decreases to a minimum value at a particular TIT, after which it increases with TIT.

Chapter 7: Exergoeconomic Analysis

Thermoeconomics combines exergy analysis and economic principles to provide the system designer or operator with information not available through conventional energy analysis and economic evaluations but crucial to the design and operation of a cost-effective system. Thermoeconomics can be considered as exergy-aided cost minimization.

The objective of a thermoeconomic analysis shall be

1. to calculate separately the cost of each product generated by a system having more than one product,
2. to understand the cost formation process and the flow of costs in the system,
3. to optimize specific variables in a single component, or
4. to optimize the overall system.

7.1 Exergoeconomic Analysis

The exergoeconomic analysis is a method that combines exergy analysis with economic analysis. Exergoeconomic analysis aims to predict the cost of exergy streams at different points in the plant, determine the cost of exergy destroyed, and predict the cost of the final product.

In analyzing the cost implications of the exergy flow and losses, the non- exergy cost of the components and the cost associated with the flow (stream) is calculated. This cost comprises the cost associated with the investment, operation, maintenance, and fuel costs of gas turbine power plants [Ahmadi & Dincer, 2011; Siahaya, 2009]. These monetary values are used in the cost balances to determine cost flow rates [Bejan et al., 1996].

7.1.1 Non-exergy cost parameter

The non-exergy cost parameters are to be determined by following equations given in table 7.1 below:

Table 7.1: The equations of the non-exergy cost parameters.

SL No	Particulars	Equations	
1	Anualization cost of the equipment ($\dot{C}$) ($/hr)	$\dot{C}$ = PW X CRF Where; PW = Present worth	7.1
		PW = PEC – (SV x PWF) Where; SV = Salvage value.	7.2
2	The present worth factor (PWF)	PWF = $\frac{1}{(1+i)^N}$ Where, N = time period, i= interest.	7.3
3	Capital recovery factor (CRF)	CRF = $\frac{i}{1-(1+i)^{-N}}$	7.4
4	Capital Cost of the Component/Capital cost rate	$Z_k = \frac{\varphi_k \ x \ \dot{C}}{3600 \ x \ n}$ Where; φ_k = Operation & Maintenance factor; n = operating hour.	7.5

7.1.2 Exergy cost parameter and cost balance equation

The plant is separated into seven subsystems to calculate the exergy cost parameter, i.e., Exergy inflows and outflows to these subsystems, exergy destruction in these subsystems, and exergy losses to the environment. In analyzing the Combined cycle power plant's exergy cost having air compressor, combustion chamber, gas turbine, HRSG, Steam turbine, Condenser & BFW pump, the code/numberings in every state of the power plant are taken to the figure 3.14.

Table 7.2: The equations of the exergy cost parameters.

SL No	Particulars	Equations	
1	Air Compressor	$\dot{C}_2=\dot{C}_1+\dot{C}_6+Z_{AC}$	7.6
		$\dot{C}_1 = 0$	7.7
2	Combustion Chamber	$\dot{C}_3=\dot{C}_2+\dot{C}_5+Z_{CC}$	7.8
3	Gas Turbine	$\dot{C}_4+\dot{C}_6+\dot{C}_7=\dot{C}_3+Z_{GT}$	7.9

SL No	Particulars	Equations	
		$\frac{\dot{C}_3}{\dot{E}_3} = \frac{\dot{C}_4}{E_4}$ (F-rule)	7.10
		$\frac{\dot{C}_6}{\dot{W}_{AC}} = \frac{\dot{C}_7}{W_{net,GT}}$ (P-rule)	7.11
		i.e; $\dot{C}_3 = \dot{C}_4$ or $\dot{E}_3 = E_4$	7.12
		i.e; $\dot{C}_6 = \dot{C}_7$	7.13
4	HRSG: (Including deaerator)	$\dot{C}_4$-$\dot{C}_8$+$\dot{C}_{23}$-$\dot{C}_9$+$\dot{C}_{18}$-$\dot{C}_{19}$+Z_{HRSG}=0	7.14
		$\frac{\dot{C}_8}{\dot{E}_8} = \frac{\dot{C}_4}{E_4}$ (F-rule)	7.15
		$\frac{\dot{C}_9 - \dot{C}_{23}}{E_9 - E_{23}} = \frac{\dot{C}_{19} - \dot{C}_{18}}{E_{19} - E_{18}}$ (P-rule)	7.16
5	Steam turbine	$\dot{C}_9$-$\dot{C}_{10}$+ $Z_{ST} = \dot{C}_{w2}$	7.17
6	Condenser, Condensate Pump & related Junction	$\dot{C}_{10}$-$\dot{C}_{16}$+$\dot{C}_{25}$-$\dot{C}_{24}$+$\dot{C}_I$-$\dot{C}_J$+$\dot{C}_{w3}$+$\dot{C}_{w4}$+$\dot{C}_{w5}$+Z_{cond} = 0	7.18
		Assuming that $\dot{C}_I = \dot{C}_J = \dot{C}_{25}$-$\dot{C}_{24}$=0	7.19
7	BFW Pump	$\dot{C}_{20} = \dot{C}_{w6}$+ $\dot{C}_{19}$+$Z_{BFW,pump}$	7.20
		$\frac{\dot{C}_{20}}{\dot{E}_{20}} = \frac{\dot{C}_{21}}{E_{21}} = \frac{\dot{C}_{23}}{\dot{E}_{23}}$	7.21

The cost of fuel for power is taken as \$4.45 SCF.

In evaluating the cost of the component, we need to calculate:

Average Cost per unit fuel of exergy, $C_F = \frac{\dot{C}_F}{E_F}$ 7.22

Average Cost of product, $C_P = \frac{\dot{C}_P}{E_P}$ 7.23

Cost rate of exergy destruction, $C_D = \dot{C}_F$ x E_D 7.24

Exergoeconomic factor, $f_k = \frac{Z_k}{Z_k + C_D}$ 7.25

7.2 Monthly economic analysis

Economic analysis of the Fenchuganj combined cycle power plant is done every month. For the economic analysis, reference tariffs are used which is specified for each Contract Year in the Term as set forth in levelized tariff table on Annexure 2. For each Contract Year and for each fuel, the Reference Tariff is composed of two components:

(A) Reference Capacity Price; and

(B) Reference Energy Price.

From and after the Commercial Operations Date, the Capacity Payment and Energy Payment payable of the plant for Dependable Capacity and Net Energy Output, respectively, in any period, during the Term shall be calculated based on the Reference Capacity Price and the Reference Energy Price.

(A) **Reference Capacity Price**:

The Reference Capacity Price for each Contract Year is made up of two components:

(i) the reference non-escalable component (the "Reference Non-Escalable Capacity Price"); and

(ii) the reference escalable component consisting of the reference foreign escalable capacity price (the "Reference Foreign Escalable Capacity Price") and the reference local escalable capacity price (the "Reference Local Escalable Capacity Price").

(B) **Reference Energy Price**:

The Reference Energy Price is made up of two components:

(i) the reference variable operation and maintenance component consisting of the reference foreign variable operation and maintenance component (the "Reference Foreign Variable Operation and Maintenance Price") and the local variable operation and maintenance component (the "Reference Local Foreign Variable Operation and Maintenance Price") (together defined as the "Reference Variable Operation and Maintenance Price"); and

(ii) the price payable for 1 (one) kWh of Net Energy Output for the Gas consumed at the net heat rate for various average plant factors (PF_h), which plant factors are set out in annexure 2 (such heat rates are herein referred to as the "Reference Heat Rates") (such prices are herein referred to as the "Reference Gas Price").

The Reference Non-Escalable Capacity Price, the Reference Foreign Escalable Capacity Price, the Reference Local Escalable Capacity Price, and the Reference Foreign Variable Operation and Maintenance Price, the Reference Local Variable Operation and Maintenance Price for each Contract Year are provided in annexure 2.

7.2.1 Capacity and Energy Payments

For the economic analysis of every month need to calculate two components of this plant. One is Capacity payment and another is Energy payment. These two components are described briefly below:

(i) Capacity Payment

(A) From and after the Commercial Operations Date, for each month, a Capacity Payment, for making available the Dependable Capacity in such month, which shall be calculated as follows:

$$CP_{mn} = CPNE_{mn} + CPE_{mn} \tag{7.26}$$

where:

CP_{mn} = Capacity Payment payable for Month "m" of Contract Year "n" in Taka.

$CPNE_{mn}$ = Non-Escalable Capacity Payment payable for Month "m" of Contract Year "n" in Taka.

CPE_{mn} = Escalable Capacity Payment payable for Month "m" of Contract Year "n" in Taka.

Calculation of CPNE

$$CPNE_{mn} = RNECP_n \text{ x } CE_m \text{ x } DC_m \tag{7.27}$$

Where:

$RNECP_n$ = Reference Non-Escalable Capacity Price as set forth in annexure 2, as applicable, for Contract Year "n" denominated in Dollars per kW-Month.

CE_m = the rate at which Sonali Bank sells one Dollar in exchange for Taka, as published in the Exchange Rates Bulletin or any latest official letter or circular to the "Authorised Dealers" (as that expression is used by Sonali Bank) or defined in any publication of Sonali Bank) in Bangladesh, on the first Day of the Month immediately following the Month "m" (each such date shall be defined as the "Indexation Date").

DC_m = the quotient of which the numerator is (a) sum of (i) each Dependable Capacity in effect during the Month "m" multiplied by (ii) the number of hours that each such Dependable Capacity was in effect during such

Month “m”, and the denominator is (b) the total number of hours in such Month “m”;

Calculation of CPE:

$$CPE_{mn} = CPE(US)_{mn} + CPE(Tk)_{mn} \qquad 7.28$$

where:

$CPE(US)_{mn}$ = Foreign Escalable Capacity Payment denominated in Dollars and payable in Taka for Month “m” of Contract Year “n” and calculated as follows:

$CPE(US)_{mn}$ = $RECP(US)_n \times CE_m \times FIIF_m \times DCm$ 7.29

$CPE(Tk)_{mn}$ = Local Escalable Capacity Payment denominated in Taka and payable in Taka for Month “m” of Contract Year “n” and calculated as follows:

$CPE(Tk)_{mn}$ = $RECP(Tk)_n \times LIIF_{qm} \times DCm$ 7.30

where:

$RECP(US)_n$ = Reference Foreign Escalable Capacity Price denominated in Dollars per kW-Month for Contract Year “n”.

$FIIF_m$ = Foreign Inflation Indexation Factor applicable to the Month “m” which shall be equal to:

$$PI(US)_q / PI(US)_b \qquad 7.31$$

where:

$PI(US)_q$ = the value of the United States Consumer Price Index as published in the publication of the International Monetary Fund entitled “International Financial Statistics” for the first month of the quarter “q” (i.e. January, April, July and October) in which the Month “m” appears:

$PI(US)_b$ = the value of the United States Consumer Price Index, as published in the publication of the International Monetary Fund entitled “International Financial Statistics”, for the month in which the Reference Date falls.

$RECP(Tk)_n$	=	Reference Local Escalable Capacity Price denominated in Taka per kW-Month for Contract Year "n".
$LIIF_m$	=	Local Inflation Indexation Factor applicable to the Month "m" and shall be calculated as follows:

$$LIIF_m = PI(Tk)_q / PI(Tk)_b \qquad 7.32$$

where:

$PI(Tk)_q$	=	the value of the Bangladesh Consumer Price Index as published in the publication of the International Monetary Fund entitled "International Financial Statistics" for the first month of the quarter "q" (i.e. January, April, July and October) in which the Month "m" appears;
$PI(Tk)_b$	=	the value of the Bangladesh Consumer Price Index as published in the publication of the International Monetary Fund entitled "International Financial Statistics", for the month in which the Reference Date falls.

(ii) Energy Payment

From and after the Commercial Operations Date, for each month, the Energy Payment in accordance for Net Energy Output generated and supplied to national grid during such month which shall be calculated as follows:

$$EP_{mn} = VOMP_{mn} + FP_{mn} \qquad 7.33$$

where:

EP_{mn}	=	Energy Payment payable in Taka for Month "m" of Contract Year "n".
$VOMP_{mn}$	=	Variable Operation and Maintenance Payment payable in Taka for Month "m" of Contract Year "n".
FP_{mn}	=	Fuel Payment payable in Taka in Month "m" of Contract Year "n".

(A) Calculation of VOMP

$$VOMP_{mn} = VOMP(US)_{mn} + VOMP(Tk)_{mn} \qquad 7.34$$

where:

$VOMP(US)_{mn}$ = Foreign Variable Operation and Maintenance Payment denominated in Dollars and payable in Taka for Month "m" of Contract Year "n" and calculated as follows:

$$VOMP(US)_{mn} = ((RVOMP(US)_n x\ CE_m\ x\ FIIF_m)\ x\ NEO_h\ x\ PFadj \qquad 7.35$$

where:

$RVOMP(US)_n$ = Reference Foreign Variable Operation and Maintenance Price denominated in Dollars per kWh for the Contract Year "n".

NEO_h = Net Energy Output measured in kWh during interval "h" in Month "m" in kWh.

h = each 30 minute interval ending on an hour or half-hour during which the Net Energy Output was made available by the Company for sale to BPDB and purchased by BPDB pursuant to this Agreement.

H = Total number of intervals "h" in Month "m".

PFadj = Power Factor Adjustment = 0.85/average power factor (lagging) of the Month 'm' at Delivery Point and shall be applicable if PFadj<1.

0.85 = the required power factor (lagging) at the Delivery Point.

$VOMP(Tk)_{mn}$ = Local Variable Operation and Maintenance Payment (in Taka) for Month "m" of Contract Year "n" and calculated as follows:

$$VOMP(Tk)_{mn} = ((RVOMP(Tk)_n x\ LIIF_m)\ x\ NEO_h\ x\ PFadj \qquad 7.36$$

where:

$RVOMP(Tk)_n$ = Reference Local Variable Operation and Maintenance Price (in Taka per kWh) for the Contract Year "n".

(2) Calculation of Fuel Payment for Operation on Gas

FP_{mn}(Gas) = Fuel Payment for Gas payable in Taka in Month "m" of Contract Year "n" is provided by the gas supplier, and it's calculated by the actual gas consumption on that month. The actual consumption of gas is measured to Net Energy Output in that month.

7.3 Result and Discussion

7.3.1 Monthly performance indicator parameter

The monthly costing of Fenchuganj Combined Cycle Power plant data was taken from July 2018 to June 2019. In that period, July 2018 to December 2018 plant was running in simple cycle mode operation, and from January 2019 to June 2019 plant was running in combined cycle mode. That period is chosen for showing the costing of combined cycle and simple cycle.

The monthly gross power produced, and auxiliary power consumption of the plant from July 2018 to December 2018 are shown in table 7.3.

Table 7.3: Monthly Energy generation report from July, 2018 to December, 2018.

Month	Gross Energy (kWh)	Auxiliary Consumption (kWh)	Net energy output, NEO (kWh)
Jul-18	21,388,760	535,273	20,853,487
Aug-18	21,151,354	508,509	20,642,845
Sep-18	20,037,082	467,828	19,569,254
Oct-18	20,230,682	430,402	19,800,280
Nov-18	19,761,378	395,970	19,365,408
Dec-18	19,783,576	404,578	19,378,998

The plant's monthly operating cost in simple cycle mode showing in table 7.4, table 7.5, and table 7.6. Table 7.4 shows the capacity payment of the plant in simple cycle operation mode.

Table 7.4: Capacity payment in Simple Cycle Operation Mode.

Month	Net Electrical Output	Non Escalable Capacity Payment (Taka)	Foreign Escalable Capacity Payment	Local Escalable Capacity Payment	Escalable Capacity Payment (Tk)	Capacity Payment
	(KWh)	RNECPn X CEm X DCm	CPE(US)mn	CPE(TK)mn	CPE(US)mn + CPE(TK)mn	
Jul-18	20,853,487	44,375,836	5,412,596	1,782,313	7,194,910	51,570,746
Aug-18	20,642,845	44,375,836	5,412,596	1,782,313	7,194,910	51,570,746
Sep-18	19,569,254	44,402,345	5,415,830	1,782,313	7,198,143	51,600,488
Oct-18	19,800,280	44,455,363	5,422,296	1,782,313	7,204,610	51,659,972
Nov-18	19,365,408	44,481,871	5,425,530	1,782,313	7,207,843	51,689,715
Dec-18	19,378,998	44,481,871	5,425,530	1,782,313	7,207,843	51,689,715

The variable operation and maintenance payment of the plant from July 2018 to December 2018 show in table 7.5 as below:

Table 7.5: Variable Operation and Maintenance payment in Simple Cycle Operation Mode.

Month	VOMP(US)mn	VOMP(TK)mn	Variable O&M Payment (Tk)	Fuel Payment (Taka)	Energy Payment	Total Payment (Tk)
Jul-18	870,973	520,691	1,391,664	31,525,989	32,917,653	84,488,398
Aug-18	862,175	515,431	1,377,606	32,823,792	34,201,398	85,772,144
Sep-18	817,824	488,625	1,306,448	32,693,760	34,000,208	85,600,696
Oct-18	828,466	494,393	1,322,860	33,380,453	34,703,312	86,363,285
Nov-18	810,754	483,535	1,294,289	33,235,560	34,529,849	86,219,564
Dec-18	811,323	483,874	1,295,197	34,141,862	35,437,060	87,126,774

The performance indicator parameter such as Total payment, per unit fuel cost, per unit generation cost, per unit VOMP cost, is calculated using the data from Tables 7.2 and 7.3 for that month. This per unit generation cost is helpful to identify the plant is cost-efficient on that month or not. Table 7.4 shows the performance indicator parameter in the simple cycle as below:

Table 7.6: Performance Indicator in Simple Cycle Operation Mode

Month	Net Electrical Output (KWh)	Capacity Payment	Energy Payment	Total Payment (Tk)	Per Unit Fuel Price (Tk/kWh)	Per Unit Generation Cost (Tk/kWh)
Jul-18	20,853,487	51,570,746	32,917,653	84,488,398	1.51	4.05
Aug-18	20,642,845	51,570,746	34,201,398	85,772,144	1.59	4.16
Sep-18	19,569,254	51,600,488	34,000,208	85,600,696	1.67	4.37
Oct-18	19,800,280	51,659,972	34,703,312	86,363,285	1.69	4.36
Nov-18	19,365,408	51,689,715	34,529,849	86,219,564	1.72	4.45
Dec-18	19,378,998	51,689,715	35,437,060	87,126,774	1.76	4.50

January 2019 to June 2019 of the Fenchuganj combined cycle power plant data is used to calculate the plant's performance indicator parameter during that period. Table 7.7 shows the monthly gross power produced and auxiliary power consumption of the plant from January 2019 to June 2019 below:

Table 7.7: Monthly Energy generation report from January, 2019 to June, 2019.

Month	Gross Energy (kWh)	Auxiliary Consumption (kWh)	Net Energy Output, NEO (kWh)
Jan-19	29,968,451	634986	29,333,465
Feb-19	28,783,187	609587	28,173,600
Mar-19	30,861,124	622286	30,238,838
Apr-19	29,328,275	597395	28,730,880
May-19	31,468,924	628636	30,840,288
Jun-19	30,387,011	603491	29,783,520

The plant's monthly operating cost in combined cycle mode is shown in table 7.8, table 7.9, and table 7.10 using data from table 7.7. Table 7.8 shows the capacity payment of the plant in simple cycle operation mode.

Table 7.8: Capacity payment in Combined Cycle Operation Mode.

Month	Net Electrical Output(KWh)	Non Escalable Capacity Payment	Foreign Escalable Capacity Payment	Local Escalable Capacity Payment	Escalable Capacity Payment (Tk)	Capacity Payment
Jan-19	29,333,465	44,481,871	5,425,530	1,782,313	7,207,843	51,689,715
Feb-19	28,173,600	44,614,416	5,441,696	1,782,313	7,224,010	51,838,426
Mar-19	30,238,838	44,667,433	5,448,163	1,782,313	7,230,476	51,897,910
Apr-19	28,730,880	44,773,469	5,461,096	1,782,313	7,243,410	52,016,879
May-19	30,840,288	44,773,469	5,461,096	1,782,313	7,243,410	52,016,879
Jun-19	29,783,520	44,773,469	5,461,096	1,782,313	7,243,410	52,016,879

Table 7.9 shows the variable operation and maintenance payment of the plant from January 2019 to June 2019 as below:

Table 7.9: Variable Operation & Maintenance payment in Combined Cycle Operation Mode.

Month	VOMP(US)mn	VOMP(TK)mn	Variable O&M Payment (Taka)	Fuel Payment (Taka)	Energy Payment	Total Payment (Tk)
Jan-19	1,228,078	732,427	1,960,505	34,613,488	36,573,994	88,263,708
Feb-19	1,183,033	703,467	1,886,500	34,653,528	36,540,028	88,378,454
Mar-19	1,271,263	755,034	2,026,297	36,286,606	38,312,903	90,210,813
Apr-19	1,210,735	717,381	1,928,116	35,913,600	37,841,716	89,858,595
May-19	1,299,627	770,051	2,069,678	39,475,569	41,545,246	93,562,125
Jun-19	1,255,094	743,665	1,998,759	38,718,576	40,717,335	92,734,213

The performance indicator parameter such as Total payment, per unit fuel cost, per unit generation cost, per unit VOMP cost, is calculated using the data from table 7.8 and 7.9 for that month. This per unit generation cost is helpful to identify the plant is cost-efficient on that month or not. Table 7.10 shows the performance indicator parameter in the combined cycle as below:

Table 7.10: Performance Indicator parameter in Combined Operation Mode.

Month	Net Electrical Output (KWh)	Capacity Payment	Energy Payment	Total Payment (Tk)	Per Unit Fuel Price (Tk/kWh)	Per Unit Generation Cost (Tk/kWh)
Jan-19	29,333,465	51,689,715	36,573,994	88,263,708	1.18	3.01
Feb-19	28,173,600	51,838,426	36,540,028	88,378,454	1.23	3.14
Mar-19	30,238,838	51,897,910	38,312,903	90,210,813	1.20	2.98
Apr-19	28,730,880	52,016,879	37,841,716	89,858,595	1.25	3.13
May-19	30,840,288	52,016,879	41,545,246	93,562,125	1.28	3.03
Jun-19	29,783,520	52,016,879	40,717,335	92,734,213	1.30	3.11

In simple cycle operation, the average costing of per unit generation cost from July 2018 to December 2018 is 4.32 Tk/kWh and per unit fuel cost is 1.66 Tk/kWh. In combined cycle operation, the average costing of per unit generation cost from January 2019 to June 2019 is 3.07 Tk/kWh and per unit fuel cost is 1.24 Tk/kWh.

7.3.2 Exergoeconomic parameter

The exergoeconomic parameter analysis is described below by calculating both the non-exergoeconomic and exergy cost parameter and cost balance equation.

(A) The non-exergoeconomic parameter:

Non-exergoeconomic parameters are capital recovery factor, present worth factor, annualization cost of the equipment, purchase equipment cost, and capital cost. For calculating those parameters described above, assume the interest value is 17%, N is the number of years for a return on Capital investment is 10 years, the Operating plant hours is 7884 hours, and is the factor of operating and maintenance cost which value is 1.06. The rate of gas price is taken as 4.45 TK/SCM. Table 7.11 shows the non-exergoeconomic parameter of the plant below:

Table 7.11: Non-exergoeconomic parameter.

Component	PEC	Salvage Value (SV)	PW	CRF	PWF	Annualization Cost of the Equipment	Capital Cost Z_k
	$	$				C (S/yr)	($/S)
AC	19,590,000	1,959,000	19,182,455	0.2147	0.2080	4,117,640	0.1538
CC	348,249	34,825	341,004	0.2147	0.2080	73,199	0.0027
GT	12,695,000	1,269,500	12,430,897	0.2147	0.2080	2,668,374	0.0997
HRSG	15,300,000	1,530,000	14,981,703	0.2147	0.2080	3,215,921	0.1201
ST	26,590,000	2,659,000	26,036,829	0.2147	0.2080	5,588,977	0.2087
COND	500,000	50,000	489,598	0.2147	0.2080	105,095	0.0039
PUMP	200,000	20,000	195,839	0.2147	0.2080	42,038	0.0016

(B) Exergy cost parameter and cost balance equation

The whole plant is separated into seven subsystems to calculate the exergy cost parameter. Exergy inflows and outflows to these subsystems, exergy destruction in these subsystems and exergy losses to the environment are calculated. In analyzing the exergy cost of the Combined gas power plant having air compressor, combustion chamber, gas turbine, HRSG, Steam turbine, Condenser & BFW pump, the code/numberings in every state of the power plant are taken to the figure 3.14. Table 7.10 shows the Cost formation of different component of the plant below:

Table 7.12: Cost formation of different component of the plant.

State	Substance	E	C	C	C
		kW	$/HR	$/Gj	$/kWhr
Ref	Air	0	0	0	0
1	Air	1.63	0	0	0
2	Air	31,663.05	1,574.83	13.82	0.05
3	Comb. Gas	99,307.50	3,076.10	8.60	0.03
4	Comb. Gas	30,586.10	1,360.85	12.36	0.04
5	Natural Gas	93,098.01	1,491.43	4.45	0.02
6	W_{AC}	35,021.07	1,021.21	8.10	0.03
7	W_{GT}	31,527.88	1,052.80	9.28	0.03
8	Comb. Gas	2,164.57	108.783	13.96	0.05
9	Water	19287.31	1,552.55	22.36	0.08
10	Water	1802.71	145.111	22.36	0.08
11	Water	1781.50	0	0	0
12	Water	1760.29	0	0	0
13	Water	3571.59	287.499	22.36	0.08
14	Water	15.04	0	0	0
15	Water	15.04	0	0	0
16	Water	1171.38	94.292	22.36	0.08
17	Water	1.17	0	0	0
18	Water	520.82	125.256	66.81	0.24
19	Water	541.33	164.000	84.15	0.30
20	Water	401.49	173.652	120.14	0.43
21	Water	13.49	3.560	73.28	0.26
22	Water	12.13	3.160	72.39	0.26
23	Water	401.49	93.151	64.45	0.23
24	Water	3.29	0	0	0
25	Water	8.78	0	0	0
26	Water	58.49	0	0	0

Since the costs change year by year, costs should be levelized, used as inputs in thermoeconomic analysis. Average Cost per unit fuel of exergy (CF), Average Cost of the product (CP), Cost rate of exergy destruction (CD), and exergoeconomic factor (fk) for each of the components need to calculate for exergoeconomic evaluation. Table 7.13 shows the exergoeconomic parameter of each component of the combined cycle power plant bellow:

Table 7.13: Exergoeconomic parameter of each components of the plant.

Component	Cp ($/Gj)	Cf ($/Gj)	Ed (kW)	Ed (MW)	Cd ($/hr)	Z ($/h)	f (%)
Air Compressor	13.82	8.10	3,359.66	3.36	27.21	553.61	95.31
Combustion Chamber	12.36	4.45	25,453.56	25.45	113.27	9.84	7.99
Gas Turbine	9.28	8.60	2,172.45	2.17	18.69	358.76	95.05
HRSG	22.36	12.36	9,004.04	9.00	111.28	432.38	79.53
Steam turbine	24.87	22.36	4,009.42	4.01	89.65	751.44	89.34
Condenser	22.36	22.36	492.13	0.49	11.00	14.13	56.22
BFWP	120.14	84.15	63.90	0.06	5.38	5.65	51.24

From above table 7.13, the maximum exergy destruction cost is in the combustion chamber & the minimum exergy cost has occurred in the BFW pump. The following figures, 7.1, 7.2, and 7.3, show the graphical representation of exergy destruction cost, capital investment cost, and exergoeconomic factor (fk) of each plant component.

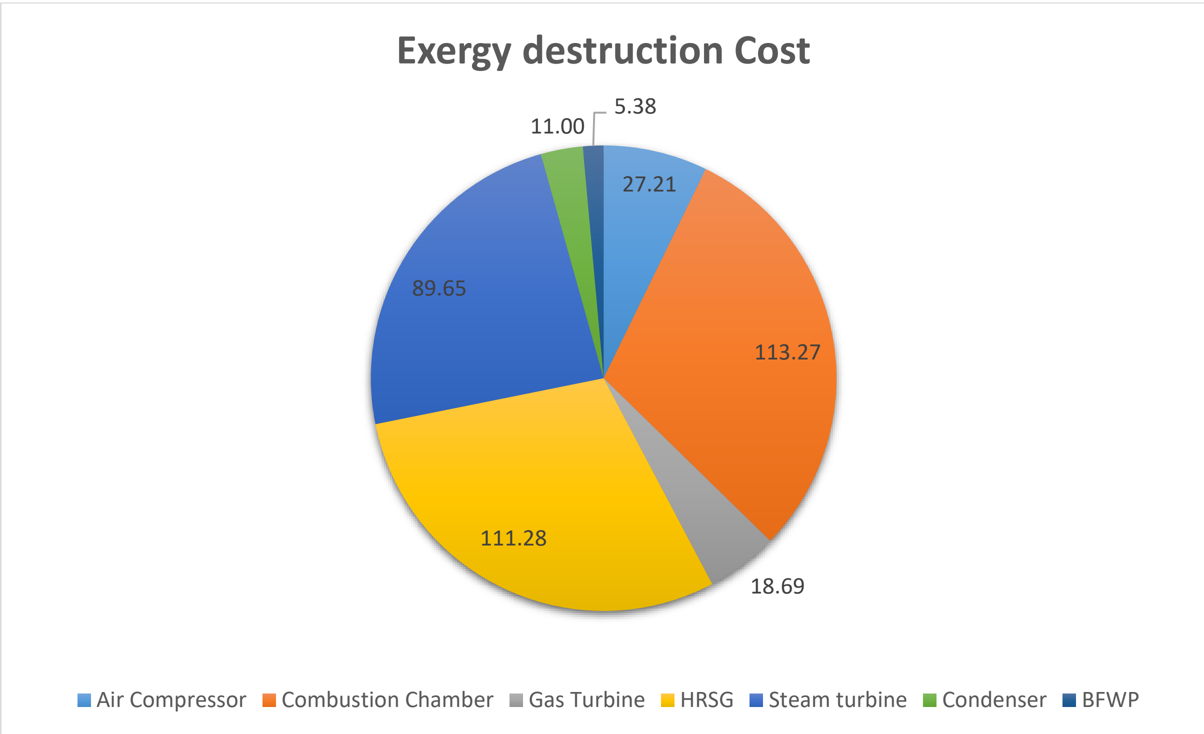

Figure 7.1: Exergy destruction Cost of each component of FCCP.

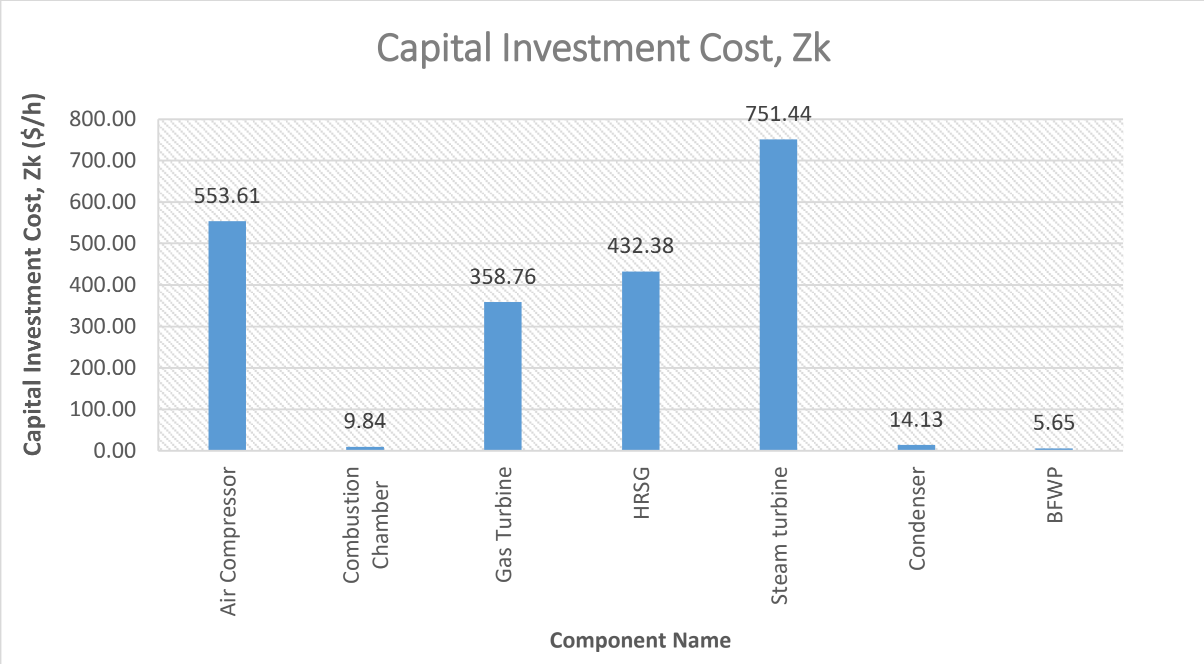

Figure 7.2: Capital Investment Cost of each component of FCCP.

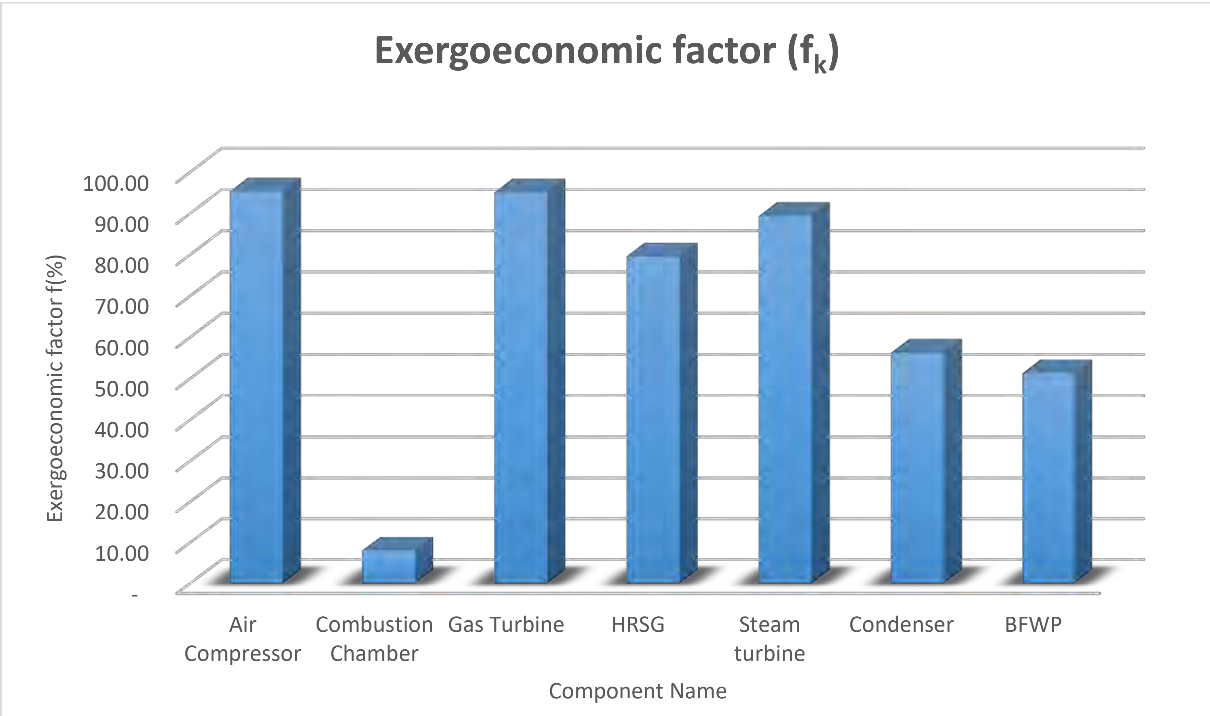

Figure 7.3: Exergoeconomic Factor of each component of FCCP.

The exergoeconomic factor showing in figure 7.3 varies with the capital investment cost & cost rate of exergy destruction. The lower cost rate of exergy destruction in the component is the higher exergoeconomic factor. That shows that if the component's destruction cost is low, then the component exergoeconomic factor is higher.

7.4 Comparing with other research works

Presently in Bangladesh Power Development Board’s power plant, only energy analysis is done every month to achieve the KPI. However, doing exergy analysis can identify the exergy destruction of every component of the plants. Furthermore, exergoeconomic analysis aims to predict the cost of exergy streams at different points in the plant, determine the cost of exergy destroyed, and predict the cost of the final product. It will also help identify every component's exergoeconomic factor, improving the low exergoeconomic factor component. If we do exergy and exergo economic analysis, we can make better improvement of our plants.

As far as I know, exergy and exergo economic analysis are not occurring in our country. For that reason, for comparing the exergoeconomic component of Fenchuganj combined cycle power plant data with other power plants by using other research work, several studies on exergy costing analyses of thermal power plants have been carried out several researchers. Those research works were done by the power plants located outside Bangladesh.

For comparing the Fenchuganj combined cycle power plant data with other power plants using other research work, several studies on exergy costing analyses of thermal power plants have been carried out by several researchers.

An exergy balance for the components of the gas turbine plants and also the overall plants is at this point performed and the net exergy flow rates crossing the boundary of each component of the plants, together with the exergy destruction, cost of exergy destruction, Capital investment cost, exergoeconomic factor of each component are calculated and are presented in Table 7.14 to Table 7.16 for simple cycle operation. The two most important performance criteria are exergy efficiency and exergy destruction rate. The exergoeconomic performance data of the Fenchuganj Combined Cycle Power Plant & selected gas turbine power plants in Nigeria in the simple cycle are given in Table 7.14:

Table 7.14 Cost of Exergy Destruction Rate in different power plants in simple cycle operation.

Plant Name	FCCP (35 MW)	DEL1 (25 MW)	DEL2 (25 MW)	AES1 (33 MW)	AES2 (33 MW)	AES3 (33 MW)	AF1 (75 MW)	AF2 (75 MW)	DEL3 (100 MW)	DEL4 (100 MW)	AF3 (138 MW)	AF4 (138 MW)
Component	**Cd ($/hr)**	**Cd ($/hr)**	**Cd ($/hr)**	**Cd ($/hr)**	**Cd ($/hr)**	**Cd ($/hr)**	**Cd ($/hr)**	**Cd ($/hr)**	**Cd ($/hr)**	**Cd ($/hr)**	**Cd ($/hr)**	**Cd ($/hr)**
AC	27	31	41	124	171	214	190	138	498	424	210	606
CC	1132	1272	1339	2762	3493	3575	3193	2849	5642	5353	3638	4615
GT	19	61	67	11	25	111	206	208	730	445	604	1045
Total	1178	1364	1447	2897	3689	3900	3589	3195	6870	6222	4452	6266

Where,

FCCP means the Fenchuganj Combined Cycle Power plant and its installed capacity of 35 MW, located in Fenchuganj, Sylhet, Bangladesh.

AES 1 means AES barges station 1 and its installed capacity of 33 MW, located in Nigeria.

AES 2 means AES barges station 1 and its installed capacity of 33 MW, located in Nigeria.

AF 1 means Afam gas turbine station 1 and its installed capacity of 75 MW, located in Nigeria.

AF 2 means Afam gas turbine station 2 and its installed capacity of 75 MW, located in Nigeria.

AF 3 means Afam gas turbine station 3 and its installed capacity of 138 MW, located in Nigeria.

AF 4 means Afam gas turbine station 4 and its installed capacity of 138 MW, located in Nigeria.

DEL 1 means Delta gas turbine station 1 and its installed capacity of 25 MW, located in Nigeria.

DEL 2 means Delta gas turbine station 2 and its installed capacity of 25 MW, located in Nigeria.

DEL 3 means Delta gas turbine station 3 and its installed capacity of 100 MW, located in Nigeria.

DEL 4 means Delta gas turbine station 4 and its installed capacity of 100 MW, located in Nigeria.

Table 7.14 shows the cost of exergy destruction in different plants in simple cycle operation. The total cost of exergy destruction of the selected gas turbine plants varies between 1178 $/hr to 6870 $/hr. FCCP unit has the least total cost of exergy destruction and DEL3 unit has the highest total cost of exergy destruction. But, the component-wise exergy destruction is different in different plants. Table 7.14 also showed that the lowest cost of exergy destruction in the air compressor happens in the FCCP unit and the highest cost of exergy destruction in the AF4 unit. The least cost of exergy destruction in the combustion chamber happens in the FCCP unit and the highest one is the DEL3 unit. The least cost of exergy destruction in gas turbine happens in AES1 unit and highest in DEL3 unit. Table 7.14 represents that, above three components of simple cycle operation, the maximum cost of exergy destruction in a simple cycle occurs in the combustion chamber.

Table 7.15 Capital investment cost in different power plants in simple cycle operation.

Plant Name	FCCP (35 MW)	DEL1 (25 MW)	DEL2 (25 MW)	AES1 (33 MW)	AES2 (33 MW)	AES3 (33 MW)	AF1 (75 MW)	AF2 (75 MW)	DEL3 (100 MW)	DEL4 (100 MW)	AF3 (138 MW)	AF4 (138 MW)
Component	**Z ($/h)**	**Z ($/h)**	**Z ($/h)**	**Z ($/h)**	**Z ($/h)**	**Z ($/h)**	**Z ($/h)**	**Z ($/h)**	**Z ($/h)**	**Z ($/h)**	**Z ($/h)**	**Z ($/h)**
AC	553	168	217	126	145	123	403	520	509	511	674	777
CC	10	14	18	13	14	13	46	51	44	44	69	79
GT	359	105	136	87	100	87	292	365	324	325	482	556
Total	922	287	371	226	259	223	741	936	877	880	1225	1412

Table 7.15 shows the Capital investment cost in different plants in simple cycle operation. The selected gas turbine plants' total capital investment cost varies between 223 $/hr to 1412 $/hr. The maximum total Capital investment cost is AF4 unit. It was found that the minimum capital investment was made in combustion chamber investment for all the plants.

Table 7.16 Exergoeconomic factor in different power plants in simple cycle operation.

Plant Name	FCCP (35 MW)	DEL1 (25 MW)	DEL2 (25 MW)	AES1 (33 MW)	AES2 (33 MW)	AES3 (33 MW)	AF1 (75 MW)	AF2 (75 MW)	DEL3 (100 MW)	DEL4 (100 MW)	AF3 (138 MW)	AF4 (138 MW)
Component	**f (%)**	**f (%)**	**f (%)**	**f (%)**	**f (%)**	**f (%)**	**f (%)**	**f (%)**	**f (%)**	**f (%)**	**f (%)**	**f (%)**
AC	95	84	83	50	46	36	68	79	51	55	76	56
CC	8	1	1	0.45	0.41	0.36	1	2	0.77	0.81	2	1
GT	95	63	67	88	80	44	58	64	31	42	44	35

Table 7.16 shows the maximum exergoeconomic factor found in gas turbine and the lowest exergoeconomic factor found in the combustion chamber.

For the Combined Cycle Comparison, Garri 2 Combined cycle power plant and Guddu 747 MW Combined cycle power plant located in Pakistan is also considered to compare with Fenchuganj Combined Cycle Power plant.

The cost of exergy destruction is maximum found in Guddu 747 MW power plant located in Pakistan and minimum cost of exergy destruction is found in FCCP. The maximum exergoeconomic factor found in gas turbine and the lowest exergoeconomic factor found in the combustion chamber.

7.5 Economic dispatch order according to economic evaluation

Power Division, Ministry of Power, Energy and Mineral Resources, Bangladesh Power Development Board, Power Grid Company of Bangladesh, and many other power generation companies are working together for reliable and uninterrupted operation of national grid in

Bangladesh 24-hour basis. For doing the smooth operation of the national grid system, dispatch instruction is one of the significant factors. Dispatch instruction means that an instruction is issued directly by the Control Center to the power generating station in accordance with the dispatch principles and guidelines established for the Grid System.

The concerned Government authorities make an economic list for dispatch instruction purposes; sometimes, it is called the economic dispatch to maintain the proper dispatch instruction method every month. Economic dispatch means the distribution of the country's total energy needs among available sources of generation based upon the variable cost where plants will be dispatched by the grid from the lowest cost to the highest cost, considering capabilities of the Grid System at the time of dispatch. An economic dispatch list is made for both the base load and Peak load power plants. Upto October 2020 in Bangladesh has 47 base load power plants, and rest are peak load power plants. The Directorate makes the economic dispatch list of System Operation of Bangladesh Power Development Board and the concerned authorities of GOB. This economic dispatch list is maintained carefully every month for the plants' economic operation in Bangladesh and to reduce the operating cost of power plants.

From January 2020 to October 2020 in Bangladesh have a total of 45 nos. Baseload power plants. Among their economic dispatch merit order Fenchuganj Combined Cycle Power Plant holds on 25th position all over the year. The average fuel consumption is 1.19 Tk/kWh all over the year. Where the plant which is holding position no 1 has the Fuel consumption is 0.86 Tk/kWh. In the year 2019 all over the year, the Fuel consumption is higher than in 2020. In 2019, the Fuel consumption was 1.61 Tk/kWh, and the position in the economic dispatch list is 31st all over the year.

Fenchuganj Combined Cycle Power Plant is a baseload gas-based power plants. In Bangladesh, gas is the primary fuel. In that sense, the economic dispatch order list its position needs to be in a higher position, but due to degradation of efficiency of gas turbine and the exergy destruction rate is high the Fuel Consumption and the variable operation and maintenance cost is higher than the other power plants. One another thing is that the plant is already running for 15 years. According to a gas turbine's standard economic life cycle, its efficiency needs to well till 22 years. Because of poor maintenance and lower capitalization cost of combustion chamber plant efficiency also decrease.

For the combined cycle, HRSG is the second component for the whole plant for the lower economic performance. Some of the heat pipes of HRSG are leaked, and the heat transfer process is disturbed in that cause heat loss occurred and lower steam production causes high rate steam production for the operation of the steam turbine.

Fenchuganj Combined Cycle Power Plant is a Government power plant, and it is running continuously. Significant change or troubleshooting cannot be done on the machine to avoid load shedding. However, if any fault occurs in any machine, it is removed rapidly, and the machine is online on time: scheduled outage and Maintenance outage took for the regular maintenance of the plant. There will be a major overhauling will be carried out on 17 contract year of the plant. During the significant overhauling period of 17 years of the life cycle, it is possible to overcome the leakage problem in HRSG and the plant's other equipment. Gas turbine also has some major overhauling provision on that period. Hopefully, the efficiency will increase more than present, but it is not possible to reach 35 MW of the gas turbine's installed capacity, but it can maintain the present derated capacity of 28 MW to 29 MW.

Chapter 8: Conclusion

Fired thermal power plant is one of the major power generations in Bangladesh. A detailed energetic, exergetic and exergoeconomic analysis of the Fenchuganj Combined Cycle Power Plant has been carried out in this thesis. The thermodynamic data were extracted from the power station. The preset study leads to the following conclusions:

8.1 Conclusions

The Fenchuganj Combined Cycle Gas power plant's performance was analyzed based on the first and second laws of thermodynamics in between the temperature range of 25^{O}C to 35^{O}C. The effects of the gas turbine's inlet temperature, the excess air factor, and the compressor pressure ratio on both simple and combined cycle performance were investigated.

Information about net electrical output, enthalpy difference of process, stack temperature, fuel utilization efficiency, and the power-to-heat ratio is obtained from energy analysis. The average range of thermal efficiency of the gas turbine is 30.23%, the overall efficiency of the plant in simple cycle operation is 30.46%, and the plant's combined cycle efficiency is 43.21%. The average Heat Rate in simple cycle operation is 11641.32 kJ/kWh, and the average Heat Rate in combined cycle operation is 8369.55 kJ/kWh.

The plant's efficiency and power output reduce with an increase in temperature, while the Heat Rate increases with temperature. The plant showed a better behavior when the ambient temperature is 25^{O}C; it's power output & efficiency increases.

An inlet air cooler can increase the gas turbine's power output by decreasing the inlet air temperature. To install an air cooling process in the plant for reducing the inlet air temperature, an investment of approximately $240,000.00 is needed initially. After installation, the plant's capacity will increase by 2 MW, which will earn approximately USD 18,950.00 additionally.

Increasing the work ratio and mass flow rate of the working fluid increase the Power Output. Installing an additional air compressor booster to supply more air to the combustion chamber may increase the work output of the simple cycle. It will help to burn the fuel more. To install an air compressor, spend approximately $380,000.00 investment is required, and which increases 3 to 4 MW more output.

In the air compressor (air compressor), the generated heat to produce compressed air is 2256 kWh/day. The percentage of waste heat to the environment is 361.01 kWh/day, for leakage 423.87 kWh/day, actuation is 92 kWh/day, and conveying & others is 1369.13 kWh/day.

The total Operating Cost of the cooling tower system is USD 177,769.87. It can be reduced by proper utilization of chemical dosing, changing the water treatment pipeline which is already damaged, reduce the excess use of raw water.

The Exergy balance of a thermodynamic system allows the determination of the Exergy destruction or the estimate of the energy losses due to actual transformations' irreversibility. It leads to the quantitative measurement of the Exergy efficiency. The results show that the combustion chamber's efficiency is the lowest compared to that of the compressor and the expansion in the turbine because it is where the highest exergy destruction takes place in the simple cycle. For the combined cycle, the highest exergy destruction occurs in the combustion chamber, and the second-highest component is the HRSG. The combustion chamber's exergy destruction is 25MW and in HRSG is 9 MW, and the total exergetic efficiency in the combined cycle is 47.35% in the operating conditions. The change in ambient temperature has a direct impact on the exergy performance of the gas turbine. Exergy destruction increases with air temperature increase, while the exergy efficiency decreases, and the plant's exergy efficiency and its units increase when the temperature decreases.

The exergetic analysis results identify the combustion chamber as the largest magnitude of overall exergy destruction, avoidable exergy destruction. By reducing the air-fuel ratio, and further preheating the combustion air, exergy destruction could be reduced in the combustion chamber. These results are helpful for future improvements for the plant. Additionally, the cost of products and cost formation is discussed. The plant owner may develop an opinion from these results and check the company's economic policy. Exergy analysis of each of the component work can be done more briefly and using the exergy analysis tool to improve the plant's performance in our country.

The plant's exergoeconomic analysis showed a lower cost of power compared to that of energy costing, which is due to the non-consideration of other cost factors involved in power production. It performs in terms of exergy cost and unit exergy cost. The exergoeconomic analysis results indicated that the combustion chamber was also the most cost-effective component to be improved due to a low cost of capital investment and high cost of exergy destruction.

Comparing the Fenchuganj Combined Cycle Power Plant with other plants also represents that the combustion chamber is the plant's highest exergy destruction component. The heat energy loss in the combustion chamber decreases with an increase in air mass flow rate. This implies that a high mass flow rate of air can minimize the combustion chamber's energy losses as this would introduce more air for combustion. The component with the highest exergy improvement potential is the combustion chamber.

This high improvement potential in the combustion chamber is due to the irreversibility of combustion and the large temperature difference between the air entering the combustion chamber and the flame temperature. These immense losses mean that a large amount of energy present in the fuel to generate useful work is wasted. Exergy improvement potential can be afforded in the combustion chamber by preheating the reactants and reducing the heat loss and the excess air entering the combustion chamber. The lower improvement potential in the air compressor, compared with the combustion chamber, is due to relatively heat loss from the air compressor through friction compared to the large temperature difference between the air entering the combustion chamber and the flame temperature. These results have made it possible to determine the critical points of the gas turbine system stating hierarchy on its components so that the measure is applied in the places where they will be most effective.

8.2 Recommendations for Future Works

Fenchuganj Combined Cycle Power Plant is a baseload gas-based power plant and running continuously. Plant’s standard economic life cycle is 20 years, and it is almost past 15 years. Significant changes on that machine may not be economical and user friendly. However, some future works can be taken to maintain the plant efficiency with life cycle degradation and keep the plant available in desired capacity with the grid. Some recommendations to improve the plant performance are given below:

i. An Inlet air cooling system, designed with specific inlet air temperature can be installed to balance the inlet air temperature in the gas turbine. It will increase power production and reduce the cost of production.

ii. Installing an additional air compressor before the gas turbine air compressor will increase the airflow to the combustion chamber and reduce the Air compressor work in the gas turbine. By installing an additional air compressor, 2 to 4 MW capacity can be increased.

iii. A pre-heater could be introduced before the combustion chamber to reduce the rate of exergy destruction in the combustion chamber.

iv. the Exergy analysis of a combined cycle power plant with carbon capture utilization. This paper considers retrofitting an existing combined cycle power plant arrangement to get carbon dioxide from gas turbine exhaust and convert it into methane to run additional gas turbine for power augmentation. Here exergy analysis of the considered combined cycle estimates the exergy efficiency and exergy destruction of significant components. The study shows the comparative results for different power plants with and without carbon capture arrangement.

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Annexure 1: Nomenclature

SYMBOLS	NAME	UNITS
$	DOLLAR	
B	BLOW DOWN RATE	kg/sec
C	ANNUALIZED COST	
CH_4	METHANE	
C_2H_6	ETHANE	
C_3H_8	PROPANE	
C_4H_{10}	BUTANE	
CO_2	CARBON DI OXIDE	
C_P	SPECIFIC HEAT CAPACITY	kg/kJK
D	DEPRECIATION	
ε/Ex	EXERGY	MW
η	EFFICIENCY	
c	COST PER UNIT OF EXERGY	$/GJ
H_2	HYDROGEN GAS	
H_2O	WATER	
H_2S	HYDROGEN SULPHIDE	
I	INTEREST	
ISO	INTERNATIONAL ORGANIZATION FOR STANDARDIZATION	
f	EXERGOECONOMIC FACTOR (%)	
ε	EXERGETIC EFFICIENCY (%)	
η	EFFICIENCY (%)	
m	MASS FLOW RATE	kg/s
N	NUMBER OF YEARS	yr
n	HOURS	hr
N_2	NITROGEN GAS	
$Ø_k$	MAINTENANCE FACTOR	
P	PRESSURE	Bar
PEC	PURCHASING EQUIPMENT COST	$
PW	PRESENT WORTH	
PWF	PRESENT WORTH FACTOR	
ρ	DENSITY	kg/m^3
R	GAS CONSTANT	kJ/kgK
r_p	PRESSURE RATIO	
S	ENTROPY	kJ/kgK
SV	SALVAGE VALUE	
T	TEMPERATURE	K
Z_k	CAPITAL COST	$/sec

ABBREVIATIONS

AC	AIR COMPRESSOR
CC	COMBUSTION CHAMBER
CT	COMBUSTION TURBINE
CCPP	COMBINED CYCLE POWER PLANT
GT	GAS TURBINE
HPST	HIGH PRESSURE STEAM TURBINE
HRSG	HEAT RECOVERY STEAM GENERATOR
IGV	INLET GUIDE VANES
ST	STEAM TURBINE
SPECO	SPECIFIC EXERGY COST ANALYSIS
TIT	TURBINE INLET TEMPERATURE
TOT	TURBINE OUTLET TEMPERATURE

SUBSCRIPT AND SUPERSCRIPT

a	AIR
GT	GAS TURBINE
AC	AIR COMRESSOR
CC	COMBUSTION CHAMBER/COMBUSTION
CHE	CHEMICAL
PH	PHYSICAL
D/dest	DESTRUCTION
f	FUEL
g	FLUE GAS/EXHAUST GAS/PRODUCT OF COMBUSTION
K	COMPONENT
O	OVERALL
P	PRESSURE
Ref	REFERENCE/AMBIENT CONDITION
T	TEMPERATURE
th	THERMAL
W	WORK

Annexure 2: Tariff

REFERENCE TARIFF

Contract Year	Reference Capacity Price			Reference Energy Charge	
	Reference Non-Escalable Capacity Price (Dollar/kW-Month) **($RNECP_n$)**	Reference Foreign Escalable Capacity Price (Dollar/kW-Month) **($RECP(US)_n$)**	Reference Local Escalable Capacity Price (Taka/kW-Month) **($RECP(Tk)_n$)**	Reference Foreign Variable Operation and Maintenance Price (Dollar/kWh) **($RVOMP(US)_n$)**	Reference Local Variable Operation and Maintenance Price (Taka/kWh) **($RVOMP(Tk)_n$)**
1	12.049483	1.469696	40.507122	0.000728	0.024962
2	12.051574	1.469696	40.507122	0.000728	0.024962
3	12.053791	1.469696	40.507122	0.000728	0.024962
4	12.056142	1.469696	40.507122	0.000728	0.024962
5	12.058637	1.469696	40.507122	0.000728	0.024962
6	12.061287	1.469696	40.507122	0.000728	0.024962
7	12.064104	1.469696	40.507122	0.000728	0.024962
8	12.067100	1.469696	40.507122	0.000728	0.024962
9	12.070287	1.469696	40.507122	0.000728	0.024962
10	12.073682	1.469696	40.507122	0.000728	0.024962
11	12.074333	1.469696	40.507122	0.000728	0.024962
12	12.242161	1.469696	40.507122	0.000728	0.024962
13	7.904138	1.469696	40.507122	0.000728	0.024962
14	7.904138	1.469696	40.507122	0.000728	0.024962
15	7.904138	1.469696	40.507122	0.000728	0.024962
16	7.904138	1.469696	40.507122	0.000728	0.024962
17	7.904138	1.469696	40.507122	0.000728	0.024962
18	7.904138	1.469696	40.507122	0.000728	0.024962
19	7.904138	1.469696	40.507122	0.000728	0.024962
20	7.904138	1.469696	40.507122	0.000728	0.024962

Reference Heat Rates

Plant Factor (PF_h)	Reference Heat Rates (kJ/kWh) (HHV)	
(% of Dependable Capacity)	A. Facility Operating in Simple Cycle Mode (applicable to Simple Cycle Operation Period)	B. Facility Operating in Combined Cycle Mode (applicable following Commercial Operations Date)
100%	10,780	7,175
95%	10,834	7,217
90%	10,962	7,271
85%	11,091	7,332
80%	11,213	7,412
75%	11,408	7,507
70%	11,660	7,603
65%	11,924	7,721
60%	12,341	7,861
55%	12,640	8,024
50%	13,111	8,253
45%	13,622	8,639
40%	14,270	9,190
35%	15,105	9,715
30%	16,172	10,370
25%	17,689	11,702

Annexure 3: Power Plant Operational Data

The collected data from the Fenchuganj combined cycle power plant at different times and seasons are shown in this section. Load factor is one of the major issues from taken this data from the power plant. Majority percentage of the data was collected through the distributive control system (DCS) & the rest of the data was collected manually by using a fire gun, anemometer, multimeter, tachometer, Etc. The power plant's operating data are both collected in the winter season & summer season (2018-2019). Data from DCS system is more accurate than the onsite measuring data.

1. A Generator and exciter data

Date	Name time	GENERATOR AND EXCITER									
		Active	Re active	PF	AMPS	VOLTS	FLD AMPS	FLD VOLTS	RTD (1-9) MAX	Cold Gas	Hot Gas
	unit	MW	MVAr	Cosφ	KA	KV	A	V	˚C		
12.06.18	0	31	9	0.96	1.68	11.11	3	20	75	39	67
	2	31	8	0.96	1.69	11.13	3	20	75	39	67
	4	31	8	0.96	1.68	11.11	3	20	75	39	67
	6	31	4	0.99	1.68	11.13	2.95	19.2	75	40	67
	8	31	5	0.98	1.68	11.11	2.95	19.2	75	40	67
	10	31	3	0.98	1.68	11.11	3	20	75	40	67
	12	31	4	0.98	1.57	11.13	3	20	76	40	68
	14	31	5	0.98	1.6	11.13	3	20	76	40	68
	16	31	7	0.98	1.58	11.13	3	20	76	40	68
	18	31	7	0.97	1.6	11.13	3	20	76	40	68
	20	31	8	0.97	1.68	11.13	3	20	76	40	68
	22	31	9	0.97	1.68	11.13	3	20	76	40	68
13.06.18	0	31	4	0.99	1.68	11.13	2.95	19.2	75	40	67
	2	31	5	0.98	1.68	11.11	2.95	19.2	75	40	67
	4	31	3	0.98	1.68	11.11	3	20	75	40	67
	6	31	4	0.98	1.57	11.13	3	20	76	40	68
	8	31	5	0.98	1.6	11.13	3	20	76	40	68
	10	31	9	0.96	1.68	11.11	3	20	75	39	67
	12	31	8	0.96	1.69	11.13	3	20	75	39	67
	14	31	8	0.96	1.68	11.11	3	20	75	39	67
	16	31	4	0.99	1.68	11.13	2.95	19.2	75	40	67
	18	31	5	0.98	1.68	11.11	2.95	19.2	75	40	67
	20	31	3	0.98	1.68	11.11	3	20	75	40	67
	22	31	4	0.98	1.57	11.13	3	20	76	40	68
14.06.18	0	31	5	0.98	1.6	11.13	3	20	76	40	68
	2	31	7	0.98	1.58	11.13	3	20	76	40	68
	4	31	7	0.97	1.6	11.13	3	20	76	40	68
	6	31	8	0.97	1.68	11.13	3	20	76	40	68
	8	31	9	0.97	1.68	11.13	3	20	76	40	68

Date	Name time	GENERATOR AND EXCITER									
		Active	Re active	PF	AMPS	VOLTS	FLD AMPS	FLD VOLT S	RTD (1-9) MAX	Cold Gas	Hot Gas
	unit	MW	MVAr	Cosφ	KA	KV	A	V	˚C		
14.06.18	10	31	9	0.96	1.68	11.11	3	20	75	39	67
	12	31	8	0.96	1.69	11.13	3	20	75	39	67
	14	31	8	0.96	1.68	11.11	3	20	75	39	67
	16	31	7	0.98	1.58	11.13	3	20	76	40	68
	18	31	3	0.98	1.68	11.11	3	20	75	40	67
	20	31	4	0.98	1.57	11.13	3	20	76	40	68
	22	31	5	0.98	1.6	11.13	3	20	76	40	68

1.B Combustion and gas turbine

Date	COMBUSTION AND TURBINE														
	VSR -1	VG C-1	VG C-2	Speed	FSR	TTX M	TTR XB	Sprea d	Spr ead -1	Spr ead -2	Spr ead -3	FP G1	FT G	FPG 2	FQG
	%			rpm	%	˚C						bar	˚C	bar	kg/s
12.06.18	47	36	20	5225	61.5	578	578	180	27	20	18	23	57	20	2.7
	48	36	20	5210	61.6	577	577	180	27	22	18	23	55	20	2.7
	47	37	20	5215	61.5	577	577	180	25	22	20	23	55	20	2.7
	43	35	19	5275	61	576	594	180	27	22	19	23	55	20.8	2.76
	43	36	19	5270	61.8	576	594	180	27	22	19	23	55	20.8	2.7
	48	36	20	5220	60	579	579	135	30	25	22	22	56	20	2.7
	48	36	20	5210	60	578	579	135	30	25	20	22	56	20	2.7
	50	36	20	5220	60	578	580	134	32	25	17	22	56	20	2.7
	48	36	20	5189	60	578	579	180	32	25	17	22	56	20	2.7
	48	37	20	5211	60	579	580	180	32	25	17	22	56	20	2.7
	49	37	20	5189	60	578	580	180	31	25	17	22	56	20	2.7
	50	37	20	5211	60	578	580	180	31	20	18	22	56	20	2.7
13.06.18	43	35	19	5275	61	576	594	180	27	22	19	23	55	20.8	2.76
	43	36	19	5270	61.8	576	594	180	27	22	19	23	55	20.8	2.7
	48	36	20	5220	60	579	579	135	30	25	22	22	56	20	2.7
	48	36	20	5210	60	578	579	135	30	25	20	22	56	20	2.7
	50	36	20	5220	60	578	580	134	32	25	17	22	56	20	2.7
	47	36	20	5225	61.5	578	578	180	27	20	18	23	57	20	2.7
	48	36	20	5210	61.6	577	577	180	27	22	18	23	55	20	2.7
	47	37	20	5215	61.5	577	577	180	25	22	20	23	55	20	2.7
	43	35	19	5275	61	576	594	180	27	22	19	23	55	20.8	2.76
	43	36	19	5270	61.8	576	594	180	27	22	19	23	55	20.8	2.7
	48	36	20	5220	60	579	579	135	30	25	22	22	56	20	2.7
	48	36	20	5210	60	578	579	135	30	25	20	22	56	20	2.7
14.06.18	50	36	20	5220	60	578	580	134	32	25	17	22	56	20	2.7
	48	36	20	5189	60	578	579	180	32	25	17	22	56	20	2.7
	48	37	20	5211	60	579	580	180	32	25	17	22	56	20	2.7
	49	37	20	5189	60	578	580	180	31	25	17	22	56	20	2.7
	50	37	20	5211	60	578	580	180	31	20	18	22	56	20	2.7
	47	36	20	5225	61.5	578	578	180	27	20	18	23	57	20	2.7
	48	36	20	5210	61.6	577	577	180	27	22	18	23	55	20	2.7
	47	37	20	5215	61.5	577	577	180	25	22	20	23	55	20	2.7
	48	36	20	5189	60	578	579	180	32	25	17	22	56	20	2.7
	48	36	20	5220	60	579	579	135	30	25	22	22	56	20	2.7
	48	36	20	5210	60	578	579	135	30	25	20	22	56	20	2.7
	50	36	20	5220	60	578	580	134	32	25	17	22	56	20	2.7

1.C Lube oil data on gas turbine

Date	Name Time	LUBE OIL									
		LT-OT-1.2A	96QV-1	96QA-2	Header Temp	LT-BT1D	LT-B1D	LT-B2D	LT-RG CD-1	LT-G1 D	LT-G2D
	unit	˚C	bar	bar	˚C						
12.06.18	0	70	2.1	3.6	51	63	64	80	76	65	65
	2	70	2.2	3.6	51	63	64	80	76	65	65
	4	70	2.1	3.6	51	63	64	80	76	64	64
	6	70	2.3	3.6	52	64	65	80	77	65	65
	8	70	2.3	3.6	52	64	65	80	77	65	65
	10	70	2.3	3.6	53	64	65	80	77	65	65
	12	71	2.4	3.6	53	65	65	80	77	65	65
	14	71	2.6	3.6	53	64	65	80	77	65	65
	16	70	0	3.6	52	63	64	81	78	64	65
	18	70	0	3.6	52	63	62	81	77	65	64
	20	70	0	3.6	52	63	63	80	77	64	64
	22	70	0	3.6	52	63	63	80	77	63	62
13.06.18	0	70	2.1	3.6	51	63	64	80	76	64	64
	2	70	2.3	3.6	52	64	65	80	77	65	65
	4	70	2.3	3.6	52	64	65	80	77	65	65
	6	70	2.3	3.6	53	64	65	80	77	65	65
	8	71	2.4	3.6	53	65	65	80	77	65	65
	10	71	2.6	3.6	53	64	65	80	77	65	65
	12	70	0	3.6	52	63	64	81	78	64	65
	14	70	0	3.6	52	63	62	81	77	65	64
	16	70	0	3.6	52	63	63	80	77	64	64
	18	70	0	3.6	52	63	63	80	77	63	62
	20	70	2.1	3.6	51	63	64	80	76	65	65
	22	70	2.2	3.6	51	63	64	80	76	65	65
14.06.18	0	70	2.1	3.6	51	63	64	80	76	64	64
	2	70	2.2	3.6	51	63	64	80	76	65	65
	4	70	2.1	3.6	51	63	64	80	76	64	64
	6	70	2.3	3.6	52	64	65	80	77	65	65
	8	70	2.3	3.6	52	64	65	80	77	65	65
	10	70	2.3	3.6	53	64	65	80	77	65	65
	12	71	2.4	3.6	53	65	65	80	77	65	65
	14	71	2.6	3.6	53	64	65	80	77	65	65
	16	70	0	3.6	52	63	64	81	78	64	65
	18	70	0	3.6	52	63	62	81	77	65	64
	20	70	0	3.6	52	63	63	80	77	64	64
	22	70	0	3.6	52	63	63	80	77	63	62

1.D Vibration, Air compressor and cooling water on gas turbine

Date	Name Time	VIBRATION												COMPRESSOR				COOLING WATER	
		Max Vib	BB1	BB2	BB4	BB5	BB6	BB7	BB8	BB9	BB10	BB11	BB12	CPD	CTD	INLET Air T	IGV	WT-TL-1	WT-TL-2
	unit	m/sec												bar	˚C	˚C	Ang	˚C	
12.06.2018	0	4	3	3	3	3	2	2	2	2	2	1	1	9	357	30	70	41	41
	2	4	4	3	3	3	2	2	2	2	2	1	1	9	359	30	70	41	41
	4	4	3	3	3	3	2	2	2	2	2	1	1	9	360	30	70	41	41
	6	3	3	3	3	3	3	2	2	1	1	1	1	9	361	31	70	41	41
	8	3	3	3	3	3	3	2	2	1	1	1	1	9	361	31	70	41	41
	10	5	4	4	4	3	3	2	2	2	1	1	1	9	363	32	80	42	42
	12	5	4	4	4	3	3	2	2	2	1	1	1	9	364	32	85	42	42
	14	5	4	4	4	3	3	2	2	2	1	1	1	9	365	33	85	42	42
	16	4	4	4	4	3	3	2	2	2	1	1	1	9	358	32	88	42	42
	18	4	4	4	4	3	3	2	2	2	1	1	1	9	357	31	85	42	41
	20	4	4	4	4	3	3	2	2	2	1	1	1	9	358	30	85	41	41
	22	4	4	4	4	3	3	2	2	2	1	1	1	9	356	29	85	41	41
13.06.2018	0	4	3	3	3	3	2	2	2	2	2	1	1	9	360	30	70	41	41
	2	3	3	3	3	3	3	2	2	1	1	1	1	9	361	31	70	41	41
	4	3	3	3	3	3	3	2	2	1	1	1	1	9	361	31	70	41	41
	6	5	4	4	4	3	3	2	2	2	1	1	1	9	363	32	80	42	42
	8	5	4	4	4	3	3	2	2	2	1	1	1	9	364	32	85	42	42
	10	5	4	4	4	3	3	2	2	2	1	1	1	9	365	33	85	42	42
	12	4	4	4	4	3	3	2	2	2	1	1	1	9	358	32	88	42	42
	14	4	4	4	4	3	3	2	2	2	1	1	1	9	357	31	85	42	41
	16	4	4	4	4	3	3	2	2	2	1	1	1	9	358	30	85	41	41
	18	4	4	4	4	3	3	2	2	2	1	1	1	9	356	29	85	41	41
	20	4	3	3	3	3	2	2	2	2	2	1	1	9	357	30	70	41	41
	22	4	4	3	3	3	2	2	2	2	2	1	1	9	359	30	70	41	41
14.06.2018	0	4	3	3	3	3	2	2	2	2	2	1	1	9	360	30	70	41	41
	2	4	4	3	3	3	2	2	2	2	2	1	1	9	359	30	70	41	41
	4	4	3	3	3	3	2	2	2	2	2	1	1	9	360	30	70	41	41
	6	3	3	3	3	3	3	2	2	1	1	1	1	9	361	31	70	41	41
	8	3	3	3	3	3	3	2	2	1	1	1	1	9	361	31	70	41	41
	10	5	4	4	4	3	3	2	2	2	1	1	1	9	363	32	80	42	42
	12	5	4	4	4	3	3	2	2	2	1	1	1	9	364	32	85	42	42
	14	5	4	4	4	3	3	2	2	2	1	1	1	9	365	33	85	42	42
	16	4	4	4	4	3	3	2	2	2	1	1	1	9	358	32	88	42	42
	18	4	4	4	4	3	3	2	2	2	1	1	1	9	357	31	85	42	41
	20	4	4	4	4	3	3	2	2	2	1	1	1	9	358	30	85	41	41
	22	4	4	4	4	3	3	2	2	2	1	1	1	9	356	29	85	41	41

1.E Operating parameter of Gas turbine

The following data are taken from the plant as average data to temperature.

T_1	T_2	T_3	T_4	P_1	P_2	P_3	P_4	γ_a	γ_g	C_{pa}	C_{pg}	m_a	m_f	m_g
˚C	˚C	˚C	˚C	bar	bar	bar	bar			Kj/Kg K	Kj/Kg K	kg/s	kg/s	kg/s
25	344	1091	585	1.013	9	8.9	1.005	1.4	1.33	1.005	1.148	108.22	1.779	110.00
25	347	1091	585	1.013	9	8.9	1.005	1.4	1.33	1.005	1.148	106.20	1.779	107.98
26	348	1078	578	1.013	9	8.9	1.005	1.4	1.33	1.005	1.148	106.22	1.779	108.00
26	348	1091	585	1.013	9	8.9	1.005	1.4	1.33	1.005	1.148	105.66	1.779	107.44
27	348	1091	585	1.013	9	8.9	1.005	1.4	1.33	1.005	1.148	105.55	1.779	107.33
27	348	1091	585	1.013	9	8.9	1.005	1.4	1.33	1.005	1.148	104.59	1.779	106.37
28	350	1080	578	1.013	9	8.8	1.005	1.4	1.33	1.005	1.148	107.59	1.778	109.37
28	348	1091	585	1.013	9	8.9	1.005	1.4	1.33	1.005	1.148	105.58	1.779	107.36
29	350	1075	580	1.013	9	8.7	1.005	1.4	1.33	1.005	1.148	103.55	1.707	105.26
29	349	1089	576	1.013	9	9	1.004	1.4	1.33	1.005	1.148	101.50	1.771	103.27
30	359	1097	570	1.013	9	8.9	0.901	1.4	1.33	1.005	1.148	100.50	1.778	102.28
30	352	1093	584	1.013	9	9	1.004	1.4	1.33	1.005	1.148	104.15	1.776	105.93
31	361	1112	594	1.013	9	9	1.002	1.4	1.33	1.005	1.148	103.20	1.777	104.98
31	361	1105	594	1.013	9	8.8	1.001	1.4	1.33	1.005	1.148	102.50	1.779	104.28
32	346	1091	588	1.013	9	8.8	1.012	1.4	1.33	1.005	1.148	101.30	1.778	103.08
32	348	1078	579	1.013	9	8.7	1.013	1.4	1.33	1.005	1.148	103.50	1.778	105.28
33	352	1080	574	1.013	9	8.7	1.006	1.4	1.33	1.005	1.148	101.05	1.667	102.72
33	356	1083	578	1.013	9	9	1.001	1.4	1.33	1.005	1.148	102.95	1.778	104.73
34	358	1090	585	1.013	9	8.9	1.007	1.4	1.33	1.005	1.148	103.55	1.779	105.33
34	357	1089	584	1.013	9	8.9	1.005	1.4	1.33	1.005	1.148	102.11	1.770	103.88
35	354	1079	579	1.013	9	8.9	1.007	1.4	1.33	1.005	1.148	103.55	1.775	105.33
35	355	1082	580	1.013	9	8.9	1.005	1.4	1.33	1.005	1.148	102.11	1.774	103.88

1.F Performance Indicator calculated data of Gas Turbine

The following table shows the performance indicator parameter of gas turbine:

Case	LHV	W_{AC}	W_{GT}	$W_{GT,NET}$	Work Ratio	Q_{in}	η_{GT}	SFC	$HR_{s/c}$
	kJ/kg	KW	kW	kW		kW	%		kJ/kWHr
Case No. 1	50768.57	34,694.79	63,886.48	29,191.69	0.46	90,317.28	32.32	0.22	11,138.18
	50768.57	34,367.38	62,713.28	28,345.90	0.45	90,317.28	31.38	0.23	11,470.52
	50768.57	34,373.85	61,974.34	27,600.49	0.45	90,317.28	30.56	0.23	11,780.31
	50768.57	34,192.63	62,399.66	28,207.02	0.45	90,317.28	31.23	0.23	11,527.00
	50768.57	34,050.96	62,335.77	28,284.81	0.45	90,317.28	31.32	0.23	11,495.29
	50768.57	33,741.26	61,778.21	28,036.95	0.45	90,317.28	31.04	0.23	11,596.92
	50768.57	34,817.20	63,028.23	28,211.03	0.45	90,256.36	31.26	0.23	11,517.59
	50768.57	33,954.53	62,353.19	28,398.66	0.46	90,317.28	31.44	0.23	11,449.21
	50768.57	33,405.75	59,763.92	26,358.17	0.44	86,661.95	30.41	0.23	11,836.29
	50768.57	32,642.40	60,818.77	28,176.37	0.46	89,911.13	31.34	0.23	11,487.64
	50768.57	33,229.82	61,835.76	28,605.94	0.46	90,266.51	31.69	0.22	11,359.86
	50768.57	33,703.98	61,877.00	28,173.01	0.46	90,164.98	31.25	0.23	11,521.45
	50768.57	34,226.28	62,449.02	28,222.74	0.45	90,215.75	31.28	0.23	11,507.62
	50768.57	33,994.13	61,219.39	27,225.26	0.44	90,317.28	30.14	0.24	11,942.67
	50768.57	31,967.24	59,499.37	27,532.13	0.46	90,266.51	30.50	0.23	11,802.92
	50768.57	32,869.53	60,308.71	27,439.18	0.45	90,266.51	30.40	0.23	11,842.90
	50768.57	32,396.12	59,667.07	27,270.95	0.46	84,631.20	32.22	0.22	11,172.05
	50768.57	33,419.11	60,659.88	27,240.77	0.45	90,266.51	30.18	0.23	11,929.16
	50768.57	33,717.95	61,099.28	27,381.33	0.45	90,317.28	30.32	0.23	11,874.60
	50768.57	33,146.44	60,229.49	27,083.05	0.45	89,860.37	30.14	0.24	11,944.64
	50768.57	33,197.61	60,470.32	27,272.71	0.45	90,114.21	30.26	0.23	11,895.08
	50768.57	32,838.58	59,819.26	26,980.68	0.45	90,063.44	29.96	0.24	12,017.06

1.G Performance Indicator parameter of Gas Turbine

The following table shows the Overall efficiency of gas turbine in Simple Cycle:

Case	Power output in GT	Overall efficiency in Simple Cycle
	kW	%
Case No. 02	28,753.82	31.84
	27,920.71	30.91
	27,186.48	30.10
	27,783.92	30.76
	27,860.54	30.85
	27,616.40	30.58
	27,787.86	30.79
	27,972.68	30.97
	25,962.80	29.96
	27,753.72	30.87
	28,176.85	31.22
	27,750.42	30.78
	27,799.40	30.81
	26,816.89	29.69
	27,119.14	30.04
	27,027.60	29.94
	26,861.88	31.74
	26,832.15	29.73
	26,970.61	29.86
	26,676.80	29.69
	26,863.62	29.81
	26,575.97	29.51

1.H HRSG Operating Data

Date	Time, Hrs	HP Pump Flow t/h	HP Curr Amps	LP Pump Flow t/h	Exhaust Temp ⁰C	HP feed water press	LP Feed water Press	Feed water temp ⁰C	Eco Outlet Temp ⁰C
12.06.2018	0	218	66	114	559	10.5	4.2	100	279
	1	219	66	114	559	10.5	4.2	100	279
	2	219	66	113	558	10.3	4.2	100	278
	3	218	66	113	558	10.4	4.2	100	278
	4	218	66	113	558	10.4	4.2	100	278
	5	218	66	113	558	10.3	4.2	100.9	278
	6	218	66.4	112	559	10.3	4.2	100.9	278
	7	218	66.4	113	559	10.3	4.2	100	278
	8	218	66	113	559	10.3	4.2	100.9	278
	9	208	66	113	560	10.3	4.2	100	279
	10	209	66	112	560	10.3	4.2	100	279
	11	207	66	113	560	10.3	4.2	100	279
	12	206	66	113	561	10.3	4.2	100	279
	13	205	66	113	561	10.3	4.2	100	279
	14	204	66	112	562	10.2	4.2	100	279
	15	207	66	113	560	10.3	4.2	100	279
	16	206	66	113	561	10.3	4.2	100	279
	17	205	66	113	561	10.3	4.2	100	279
	18	218	66.4	113	559	10.3	4.2	100	278
	19	218	66	113	559	10.3	4.2	100.9	278
	20	218	66	113	558	10.3	4.2	100.9	278
	21	218	66.4	112	559	10.3	4.2	100.9	278
	22	206	66	113	561	10.3	4.2	100	279
	23	205	66	113	561	10.3	4.2	100	279
14.06.2018	0	204	66	112	556	10.2	4.2	101	280
	1	206	67	112	556	10	4.2	101	280
	2	206	66	111	560	10	4.2	101	280
	3	208	66	112	553	10	4.2	100	280
	4	208	66	112	555	10	4.2	100	280
	5	215	67	111	561	10	4.2	101	280
	6	215	67	112	561	9.9	4.2	101.4	280
	7	215	66.7	111	561.5	9.9	4.2	101.4	280
	8	215	66.7	111	561.5	9.9	4.2	101	281
	9	200	65	111	560	10	4.2	101	280
	10	206	66	112	558	10	4.2	101	280
	11	205	66	110	558	10	4.2	101	280
	12	200	65	110	560	10	4.2	101	280

Date	Time, Hrs	HP Pump Flow t/h	HP Curr Amps	LP Pump Flow t/h	Exhaust Temp ⁰C	HP feed water press	LP Feed water Press	Feed water temp ⁰C	Eco Outlet Temp ⁰C
14.06.2018	13	200	65	110	555	10	4.2	101	279
	14	202	65	111	556	10	4.2	101	279
	15	202	66	111	556	10	4.2	101	279
	16	202	66	111	555	10	4.2	101	279
	17	203	66	111	556	10	4.2	100	279
	18	204	66	111	575	10	4.2	100	280
	19	204	66	111	553	10	4.2	100	280
	20	204	66	111	556	10	4.2	100	280
	21	203	66	111	557	10	4.2	100	280
	22	203	66	112	557	10.3	4.2	100	280
	23	206	66	112	557	10.3	4.2	101	280
13.06.2018	0	200	65	114	560	10	4.2	100	279
	1	200	65	114	560	10	4.2	100	279
	2	200	65	115	558	10	4.2	100	279
	3	200	65	114	560	10	4.2	100	279
	4	206	65	114	560	10	4.2	100	279
	5	205	65	112	560	10	4.2	100	279
	6	205	65	112	558	10	4.2	100	279
	7	206	66	111	558	10	4.2	100	279
	8	206	66	111	560	10	4.2	100	280
	9	204	66	111	560	10	4.2	100	280
	10	204	66	111	559	10	4.2	100	280
	11	204	66	111	559	10	4.2	100	280
	12	204	66	110	560	10	4.2	100	280
	13	200	66	110	560	10	4.2	100	280
	14	200	66	110	563	10	4.2	100	280
	15	218	65.3	111	563	10.1	4.2	100.3	278
	16	218	65	111	559	10.1	4.2	100.3	278.2
	17	221	66	113	559	10.4	4.2	101	279
	18	221	66	113	559.6	10.4	4.2	101.1	279.5
	19	221	66.3	113	560	10.4	4.2	101.1	279.5
	20	221	66	113	560.5	10.4	4.2	100.8	279.2
	21	222	66.1	113	559	10.4	4.2	100.8	279.2
	22	222	66	113	559	10.4	4.2	100.8	279.3
	23	200	65.4	114	559	10	4.2	100	279

1.I Operating parameter of HRSG at different operating load

GT load	Lean Gas				Rich Gas			
	100%	75%	50%	25%	100%	75%	50%	25%
Exhaust Gas flow, kg/s	133.81	105.83	89.47	80.14	133.81	105.83	89.58	80.14
Exhaust gas temp	565.6	602.8	615.6	566.7	565.6	602.8	615.6	566.7
- Argon , % vol	0.86	0.87	0.85	0.85	0.86	0.85	0.86	0.85
- Nitogen , % vol	71.58	71.49	71.59	71.98	71.66	71.58	71.67	72
- Oxygen , % vol	13.04	12.79	13.09	14.23	13.09	12.85	13.14	14.25
- Carbon Dioxide , % vol	3.11	3.22	3.09	2.57	3.2	3.32	3.18	2.6
- Water , % vol	11.42	11.64	11.38	10.37	11.2	11.41	11.15	10.3
HP Steam Output, kg/sec	19.7	17.8	15.5	12.27	19.7	17.8	15.6	12.27
HP Superheater exit pressure, bara	69	58	51	39	69	58	51	39
HP Superheater exit temp., °C	468	468	468	468	468	468	468	468
HP Feed Water temp. @ Eco. Inlet, °C	104	104	104	104	104	104	104	104
LP Steam Output, kg/sec	2.9	1.9	1.5	1.3	2.9	1.9	1.5	1.3
LP Superheater exit pressure, bara	4	4	4	4	4	4	4	4
LP Superheater exit temp., °C	Sat	Sat	Sat	Sat	Sat	Sat	Sat	Sat
LP Feed Water temp. @ drum Inlet, °C	104	104	104	104	104	104	104	104

1.J Operating parameter of HRSG at different operating load at design condition

GT load	Design Gas			
	100%	75%	50%	25%
Exhaust Gas flow, kg/s	133.94	105.83	89.47	80.14
Exhaust gas temp. °C	565.60	602.80	615.60	567.20
- Argon , % vol	0.86	0.87	0.85	0.88
- Nitogen , % vol	71.60	71.52	71.62	71.99
- Oxygen , % vol	13.06	12.81	13.11	14.24
- Carbon Dioxide , % vol	3.15	3.26	3.12	2.60
- Water , % vol	11.34	11.55	11.30	10.30
HP Steam Output, kg/sec	19.7	17.8	15.5	12.3
HP Superheater exit pressure, bara	69	58	51	39
HP Superheater exit temp., °C	468	468	468	468
HP Feed Water temp. @ Eco. Inlet, °C	104	104	104	104
LP Steam Output, kg/sec	2.8	1.9	1.5	1.3
LP Superheater exit pressure, bara	4	4	4	4
LP Superheater exit temp., °C	Sat	Sat	Sat	Sat
LP Feed Water temp. @ drum Inlet, °C	104	104	104	104

1.K Steam Turbine Operational Data

Date	Time, Hrs	Load	Main Stream P	Main Stream T	Main Stream m	Lube oil Press	EH oil Press	Vacc Pres	Lube oil temp	Cond Temp	DA Press
		MW	MPa	°C	Kg/s	MPa			°C		MPa
12.06.18	0	13.5	5.2	452	15	0.141	3.5	-0.09	36	42	0.06
	1	13.5	5.2	453	15	0.141	3.5	-0.09	36	42	0.06
	2	13.5	5.2	451	14.72	0.141	3.5	-0.09	36	42	0.06
	3	13.5	5.1	453	14.72	0.142	3.5	-0.09	36.5	42.3	0.06
	4	13.5	5.1	452	14.72	0.142	3.5	-0.09	36.5	42.3	0.06
	5	13.5	5.1	452	14.72	0.142	3.5	-0.09	36	42.9	0.059
	6	13.3	5.1	452	14.72	0.141	3.5	-0.09	6.8	42.9	0.06
	7	13.5	5.1	452	14.72	0.141	3.5	-0.09	36.8	42.9	0.06
	8	13.3	5.1	452	14.72	0.141	3.5	-0.09	36.8	42.9	0.06
	9	13.5	5.1	453	14.72	0.135	3.5	-0.09	37	43	0.06
	10	13.5	5.1	451	15	0.136	3.5	-0.09	37	43	0.06
	11	13.5	5.2	452	15	0.139	3.5	-0.09	37	43.5	0.06
	12	13.5	5.2	452	15	0.139	3.5	-0.09	37.5	43.5	0.06
	13	13.5	5.3	454	14.72	0.134	3.5	-0.09	37.7	43.7	0.06
	14	13.2	5.2	452	14.72	0.134	3.5	-0.09	37	43	0.06
	15	13.4	5.2	453	14.72	0.134	3.5	-0.09	37	43	0.06
	16	13.5	5.2	452	14.72	0.134	3.5	-0.09	37	43	0.06
	17	13.5	5.2	452	14.72	0.134	3.5	-0.09	37	43	0.06
	18	13.4	5.2	452	14.72	0.134	3.5	-0.09	37	43	0.06
	19	13.3	5.2	452	14.72	0.134	3.5	-0.09	37	43	0.06
	20	13.4	5.2	452	14.72	0.134	3.5	-0.09	37	43	0.06
	21	13.3	5.2	450	14.72	0.134	3.5	-0.09	37	43	0.06
	22	13.2	5.2	450	14.72	0.134	3.5	-0.09	37	43	0.06
	23	13.5	5.2	451	14.72	0.134	3.5	-0.09	37	43	0.06
13.06.18	0	13.6	5.3	452	15	0.142	3.5	-0.9	37	43	0.059
	1	13.6	5.3	450	15	0.142	3.5	-0.9	37	43	0.06
	2	13.6	5.3	450	15	0.141	3.5	-0.9	37	43	0.06
	3	13.6	5.3	450	15	0.141	3.5	-0.9	37	43	0.06
	4	13.6	5.3	450	15	0.139	3.5	-0.9	37	43	0.06
	5	13.6	5.3	452	15.16	0.142	3.5	-0.9	37.1	43.2	0.06
	6	13.6	5.3	452	15.16	0.139	3.5	-0.9	37	43.2	0.06
	7	13.7	5.3	452	15	0.142	3.5	-0.9	37	43	0.06
	8	13.7	5.3	452	15	0.139	3.5	-0.9	37.1	43.2	0.06
	9	13.6	5.3	450	15	0.139	3.5	-0.9	37	43	0.06
	10	13.6	5.3	450	15	0.142	3.5	-0.9	37	43	0.06
	11	13.6	5.3	450	15	0.142	3.46	-0.9	37	43	0.06
	12	13.6	5.3	452	15	0.139	3.46	-0.9	37	43	0.06

Date	Time, Hrs	Load	Main Stream			Lube oil Press	EH oil Press	Vacc Pres	Lube oil temp	Cond Temp	DA Press
			P	T	m						
		MW	MPa	°C	Kg/s	MPa			°C		MPa
13.06.18	13	13.6	5.3	452	15	0.139	3.46	-0.9	37	43	0.06
	14	13.5	5.3	451	15.27	0.138	3.46	-0.9	37	43	0.06
	15	13.5	5.3	452	15.27	0.139	3.46	-0.9	37	43	0.06
	16	13.5	5.3	452	15.27	0.141	3.46	-0.9	37	43	0.06
	17	13.5	5.3	452	15.27	0.141	3.46	-0.9	37	43	0.06
	18	13.5	5.3	452	15.27	0.141	3.46	-0.9	37	43	0.06
	19	13.5	5.3	451	15.27	0.141	3.46	-0.9	37	43	0.06
	20	13.5	5.3	452	15.27	0.141	3.46	-0.9	37	43	0.06
	21	13.5	5.3	452	15.27	0.141	3.46	-0.9	37	43	0.06
	22	13.5	5.3	452	15.27	0.141	3.46	-0.9	37	43	0.06
	23	13.5	5.3	452	15.27	0.141	3.46	-0.9	37	43	0.06
14.06.18	0	13	5.3	452	15	0.134	3.5	-0.9	36	42	0.057
	1	13	5.3	450	15	0.134	3.5	-0.9	36	42	0.05
	2	13	5.3	450	15	0.135	3.5	-0.9	36	42	0.05
	3	13	5.3	450	15	0.135	3.5	-0.9	36	42	0.05
	4	13	5.3	450	15	0.135	3.5	-0.9	36	42	0.05
	5	13.6	5.3	450	15	0.136	3.5	-0.9	36	42	0.06
	6	13.7	5.3	452	15	0.139	3.5	-0.9	36	42	0.06
	7	13.6	5.3	452	15	0.14	3.5	-0.9	36	42	0.06
	8	13.6	5.3	452	15	0.14	3.5	-0.9	36	42	0.051
	9	13.6	5.3	453	15	0.141	3.5	-0.9	36	42	0.06
	10	13.6	5.4	453	15	0.141	3.5	-0.9	36	42	0.06
	11	13.7	5.4	453	15	0.139	3.5	-0.9	37	43	0.06
	12	13.6	5.4	453	15	0.139	3.5	-0.9	37	43	0.06
	13	13.8	5.4	452	15	0.139	3.46	-0.9	37	43	0.06
	14	13.7	5.4	454	15	0.138	3.46	-0.9	37	43.5	0.06
	15	13.3	5.2	453	14.72	0.139	3.46	-0.9	37	43.4	0.06
	16	13.3	5.2	453	14.72	0.139	3.46	-0.9	37	44	0.06
	17	13.5	5.3	454	14.72	0.141	3.46	-0.9	37	44.1	0.06
	18	13.4	5.3	454	15	0.141	3.46	-0.9	37.6	44.1	0.06
	19	13.4	5.3	454	15	0.141	3.46	-0.9	37.6	44	0.06
	20	13.5	5.3	454	15	0.141	3.46	-0.9	37.6	44	0.06
	21	13.4	5.3	454	15	0.141	3.46	-0.9	37.6	44	0.06
	22	13.4	5.3	454	14.72	0.141	3.46	-0.9	37.6	44	0.06
	23	13.5	5.3	454	15	0.141	3.46	-0.9	37.6	44	0.06

1.L Operating parameter of steam turbine

The following table shows the performance indicator parameter of steam turbine:

T_9	T_{10}	P_9	P_{10}	h_9	h_{10}	Load	Initial steam flow in ST, $m_{s,in}$	Exhaust flow in ST $m_{s,out}$
˚C	˚C	bar	bar	kJ/kg	kJ/kg	%	kg/s	kg/s
468	53.3	69	0.145	3329	2334.2	100	15.50	15.30
468	53.3	69	0.145	3329	2334.2	95	14.73	14.54
468	53.3	69	0.145	3329	2334.2	90	13.95	13.77
468	53.3	59	0.145	3344	2360.2	85	13.18	13.01
468	53.3	59	0.145	3344	2360.2	80	12.40	12.24
468	53.3	59	0.145	3344	2360.2	75	11.63	11.48
468	53.3	59	0.145	3344	2360.2	70	10.85	10.71
468	53.3	55	0.145	3355	2373.2	65	10.08	9.95
468	53.3	55	0.145	3355	2374.2	60	9.30	9.18
468	53.3	50	0.145	3362	2382.5	55	8.53	8.42
468	53.3	50	0.145	3362	2383.1	50	7.75	7.65

1.M Performance Indicator parameter of steam Turbine

The following table shows the performance indicator parameter of steam turbine:

Generator end power	$W_{cond.pump}$	$W_{BFW,pump}$	W_{CTF}	$W_{CW,pump}$	$W_{make\ up\ pump}$	W_{pump}	$W_{net,ST}$
W_{ST} (kW)	kW	kW	kW	kW	kW	kW	kW
15,419.40	116.98	510	110	110	3	849.98	14,569.42
14,648.43	116.98	510	110	110	3	849.98	13,798.45
13,877.46	116.98	510	110	110	3	849.98	13,027.48
12,961.57	116.98	510	110	110	3	849.98	12,111.58
12,199.12	116.98	510	88	110	3	827.98	11,371.14
11,436.68	116.98	510	83	110	3	822.48	10,614.19
10,674.23	116.98	510	77	110	3	816.98	9,857.25
9,891.64	116.98	510	72	110	3	811.48	9,080.15
9,121.44	116.98	510	66	110	3	805.98	8,315.46
8,350.24	116.98	510	61	110	3	800.48	7,549.75
7,586.48	116.98	510	55	110	3	794.98	6,791.49

1.N The performance indicator parameter of combined cycle

The following table shows the performance indicator parameter of combined cycle:

Steam Consumption /rating	$W_{net,GT}$	$W_{net,ST}$	$Q_{in,GT}$	$\eta_{combined\ cycle}$	HR in CC
kg/kW.h	kW	kW	kW	(%)	kJ/kWHr
3.57	29,191.69	43,761.11	90,317.28	48.45	7429.936
3.57	28,605.94	42,404.38	90,317.28	46.95	7667.656
3.58	28,398.66	41,426.14	90,317.28	45.87	7848.721
3.60	28,345.90	40,457.48	90,317.28	44.79	8036.64
3.61	28,284.81	39,655.95	90,317.28	43.91	8199.079
3.61	28,222.74	38,836.93	90,317.28	43.00	8371.986
3.61	28,211.03	38,068.27	90,266.51	42.17	8536.228
3.62	28,207.02	37,287.17	90,266.51	41.31	8715.047
3.62	28,176.37	36,491.83	90,114.21	40.50	8889.968
3.63	28,173.01	35,722.77	90,063.44	39.66	9076.239
3.63	28,036.95	34,828.44	89,911.13	38.74	9293.556

1.O Feed water pump Operating Data

Date	Hours	FEED WATER PUMP						
		Speed	Suction Pressure	HP Discharge Pressure	LP Discharge Pressure	D-End Bearing Temp	ND-End Bearing Temp	Cooling Water Pressure
	Units	RPM	MPa	MPa	MPa	°C	°C	MPa
12.06.2018	0:00	3012	0.22	0.11	9.8	3.6	54	56
	2:00	3017	0.22	0.11	9.8	3.6	55	57
	4:00	3009	0.22	0.11	9.7	3.7	55	57
	6:00	3022	0.22	0.11	9.7	3.7	56	57
	8:00	3019	0.22	0.11	9.7	3.6	56	58
	10:00	3009	0.22	0.11	9.7	3.7	55	57
	12:00	3017	0.23	0.11	9.6	3.8	55	57
	14:00	3002	0.22	0.11	9.5	3.8	54	56
	16:00	3017	0.22	0.11	9.8	3.6	55	57
	18:00	3026	0.23	0.11	9.7	3.7	54	56
	20:00	3016	0.22	0.11	9.8	3.6	54	56
	22:00	3022	0.22	0.11	9.8	3.7	55	57
13.06.2018	0:00	3022	0.22	0.11	9.7	3.8	54	56
	2:00	3022	0.22	0.13	9.7	3.7	55	55
	4:00	3018	0.22	0.11	9.8	3.8	54	56
	6:00	3016	0.21	0.12	9.8	3.8	54	56

Date	Hours	FEED WATER PUMP						
		Speed	Suction Pressure	HP Discharge Pressure	LP Discharge Pressure	D-End Bearing Temp	ND-End Bearing Temp	Cooling Water Pressure
	Units	RPM	MPa	MPa	MPa	°C	°C	MPa
13.06.2018	8:00	3017	0.21	0.11	9.8	3.8	54	56
	10:00	3009	0.22	0.12	9.7	3.7	55	57
	12:00	3012	0.22	0.12	9.5	3.7	54	56
	14:00	3017	0.23	0.11	9.6	3.8	56	58
	16:00	3012	0.22	0.12	9.6	3.8	54	56
	18:00	3017	0.23	0.11	9.5	3.8	54	56
	20:00	3007	0.22	0.13	9.7	3.7	56	58
	22:00	3009	0.22	0.12	9.7	3.7	56	58
14.06.2018	0:00	3022	0.24	0.11	9.8	3.6	54	56
	2:00	3022	0.23	0.11	9.6	3.5	55	56
	4:00	3029	0.22	0.11	9.6	3.5	55	57
	6:00	3021	0.22	0.11	9.5	3.7	55	57
	8:00	3012	0.22	0.11	9.6	3.8	55	58
	10:00	3011	0.22	0.11	9.7	3.7	56	58
	12:00	3009	0.21	0.11	9.8	3.8	54	56
	14:00	2998	0.22	0.11	9.6	3.8	56	58
	16:00	2998	0.23	0.11	9.6	3.7	55	57
	18:00	2995	0.22	0.11	9.6	3.6	55	57
	20:00	2990	0.22	0.11	9.6	3.6	54	57
	22:00	2992	0.22	0.11	9.6	3.6	56	56

1.P CCCW Pump Operating Data

Date	Time, Hrs	CCCW Pump			CW Comn HDR Press	CW Comn HDR Temp, Tw2	CCCW Outlet press	Heat Exchanger water temp, Tw1	CCCW Exchanger Outlet press
		EH Oil Level, Temp	Suction Pressure	Delivery Pressure					
		^{0}C	Mpa	Mpa					
12.06.2018	0:00	38	0.03	0.45	0.22	31	0.32	38.1	0.3
	2:00	38	0.03	0.45	0.22	31	0.32	38.1	0.3
	4:00	38	0.03	0.45	0.22	32	0.32	38.1	0.3
	6:00	38	0.03	0.45	0.22	31	0.32	38.1	0.3
	8:00	38	0.03	0.45	0.22	31	0.32	38.1	0.3
	10:00	38	0.03	0.45	0.22	32	0.32	38.1	0.3
	12:00	38	0.03	0.45	0.22	32	0.32	38.1	0.3
	14:00	38	0.03	0.45	0.22	31.7	0.32	38.1	0.3
	16:00	37	0.03	0.45	0.22	32	0.32	38.1	0.3
	18:00	37	0.03	0.45	0.22	32	0.32	38.1	0.3
	20:00	37	0.03	0.45	0.22	32	0.32	38.1	0.3
	22:00	37	0.03	0.45	0.22	32	0.32	38.1	0.3
Date	Time,	CCCW Pump			CW	CW	CCCW	Heat	CCCW

	Hrs	EH Oil Level, Temp	Suction Pressure	Delivery Pressure	Comn HDR Press	Comn HDR Temp, Tw2	Outlet press	Exchanger water temp, Tw1	Exchanger Outlet press
		⁰C	Mpa	Mpa					
13.06.2018	0:00	38	0.03	0.45	0.22	31.3	0.32	38.1	0.3
	2:00	38	0.03	0.45	0.22	31	0.32	38.1	0.3
	4:00	38	0.03	0.45	0.22	32	0.32	38.1	0.3
	6:00	38	0.03	0.45	0.22	32.2	0.32	38.1	0.3
	8:00	38	0.03	0.45	0.22	32	0.32	38.1	0.3
	10:00	38	0.03	0.44	0.22	32	0.32	38.1	0.3
	12:00	38	0.03	0.44	0.22	32	0.32	38.1	0.3
	14:00	38	0.03	0.44	0.22	30	0.32	38.1	0.3
	16:00	38	0.03	0.44	0.22	32	0.32	38.1	0.3
	18:00	38	0.03	0.44	0.22	32	0.32	38.1	0.3
	20:00	38	0.03	0.44	0.22	32	0.32	38.1	0.3
	22:00	38	0.03	0.44	0.22	32	0.32	38.1	0.3
14.06.2018	0:00	38	0.03	0.45	0.22	30	0.32	38.1	0.3
	2:00	38	0.03	0.44	0.22	30	0.32	38.1	0.3
	4:00	38	0.03	0.44	0.22	32	0.32	38.1	0.3
	6:00	38	0.03	0.45	0.22	31	0.32	38.1	0.3
	8:00	38	0.03	0.45	0.22	31	0.32	38.1	0.3
	10:00	38	0.03	0.45	0.22	32	0.32	38.1	0.3
	12:00	38	0.03	0.45	0.22	31.5	0.32	38.1	0.3
	14:00	38	0.03	0.45	0.22	32	0.32	38.1	0.3
	16:00	37	0.03	0.45	0.22	32	0.32	38.1	0.3
	18:00	39	0.03	0.47	0.22	32.9	0.32	38.1	0.3
	20:00	38	0.04	0.44	0.22	32.7	0.32	38.1	0.3
	22:00	39	0.03	0.44	0.22	32.8	0.32	38.1	0.3

1.Q Condenser

The following table shows the condenser data taken from the site:

Tin	Tout	Tsat	Delta T
29	41	43	12
28	42	43.5	14
28.5	40	42	11.5
30	41	43	11
30.5	39.8	42.5	9.3
31	42	43.2	11
31.5	41.5	42.6	10
31.4	42.58	44	11.18
30.6	41.6	42.6	11
30.4	39	41.5	8.6
31	39.5	43	8.5

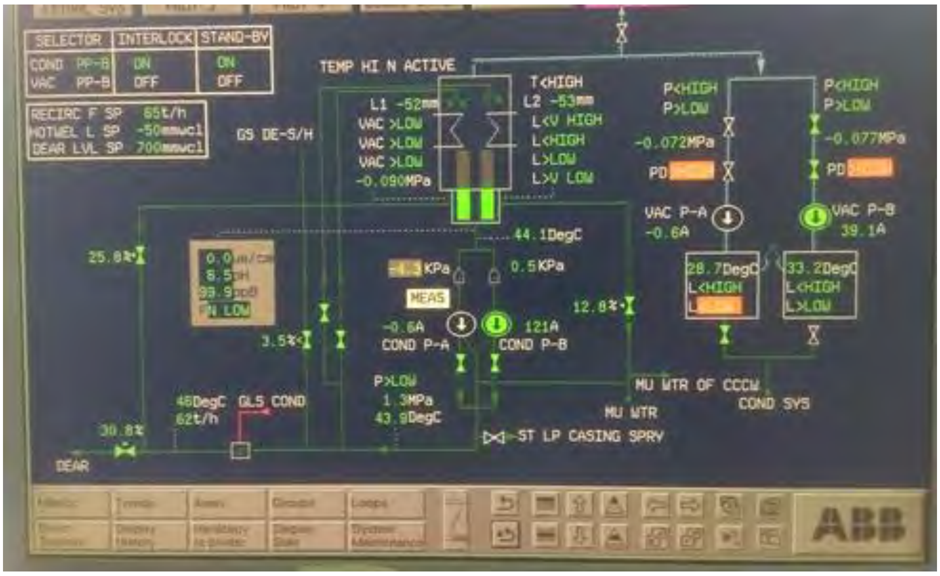

Figure 1: Condensate Pump Flow system.

1.R Key indicators of Condenser performance

mass flow of water, $m_{water,inlet}$	mass flow of water, $m_{water,outlet}$	h_{in}	S_{in}	h_{out}	S_{out}	Condenser Heat rejection, Q_{cond}
Kg/s	Kg/s	kJ/kg	kJ/kg-K	kJ/kg	kJ/kg-K	kW
15.3	15.1	121.22	0.404	171.2	0.586	789.18
15.2	15.02	117.04	0.39	175.4	0.556	908.13
15.15	14.92	119.13	0.397	167.5	0.572	760.81
15.21	14.96	125.4	0.4369	171.2	0.543	728.21
15.11	14.8	127.49	0.417	166.3	0.535	625.94
11.25	10.98	129.5	0.423	175.4	0.556	551.25
11.2	10.95	131.6	0.43	173.3	0.549	499.98
11.12	11.01	131.2	0.429	177.8	0.564	532.78
11.01	11.05	127.8	0.418	173.7	0.551	500.48
7.89	7.6	127	0.415	162.9	0.524	320.08
7.5	7.1	129.5	0.423	165	0.531	318.05

1.S Lube Oil System

Date	Time, Hrs	Lube Oil System						
		Main Pump Suction Pressure, Mpa	Main Pump Delivery Pressure, Mpa	Lube Oil Level	Lube Oil filter Diff Press	Cooler Inlet Temp, ºC	Cooler Outlet Temp, ºC	Lube Oil Tank Level
12.06.2018	0:00	0.12	1.34	-64	2	56	40	80
	2:00	0.12	1.34	-63	2	56	38	80
	4:00	0.14	1.36	-64	2	56	39	80
	6:00	0.13	1.34	-63	2	56	38	80
	8:00	0.12	1.32	-64	2	56	40	80
	10:00	0.12	1.32	-64	2	56	40	80
	12:00	0.13	1.34	-64	2	56	40	80
	14:00	0.12	1.34	-63	2	56	40	80
	16:00	0.13	1.34	-63	2	56	40	80
	18:00	0.12	1.32	-64	2	56	40	80
	20:00	0.12	1.32	-64	2	56	40	80
	22:00	0.14	1.34	-64	2	56	40	80
13.06.2018	0:00	0.12	1.31	-63	2	56	38	80
	2:00	0.12	1.31	-63	2	56	39	80
	4:00	0.12	1.32	-65	2	56	38	80
	6:00	0.12	1.32	-64	2	56	40	80
	8:00	0.12	1.32	-64	2	56	40	80
	10:00	0.14	1.34	-63	2	56	41	80
	12:00	0.14	1.36	-64	2	56	41	80
	14:00	0.12	1.32	-65	2	56	40	80
	16:00	0.12	1.34	-64	2	56	40	80
	18:00	0.12	1.34	-63	2	56	38	80
	20:00	0.14	1.36	-64	2	56	39	80
	22:00	0.13	1.34	-63	2	56	38	80
14.06.2018	0:00	0.12	1.32	-64	2	56	40	80
	2:00	0.12	1.32	-64	2	56	40	80
	4:00	0.13	1.34	-64	2	56	40	80
	6:00	0.12	1.34	-63	2	56	40	80
	8:00	0.13	1.34	-63	2	56	40	80
	10:00	0.12	1.32	-64	2	56	40	80
	12:00	0.12	1.32	-64	2	56	40	80
	14:00	0.14	1.34	-64	2	56	40	80
	16:00	0.12	1.32	-64	2	56	40	80
	18:00	0.12	1.32	-65	2	56	40	80
	20:00	0.12	1.32	-64	2	56	40	80
	22:00	0.12	1.32	-65	2	56	40	80

1.T Demi Water, Cooling Tower Operating data

Date	Time, Hrs	Demi Water		Clarified water before demi water	Cooling water Basin		Cooling Tower water (make-up)	
		pH	Conductivity µs/cm	pH	Conductivity µs/cm	pH	Conductivity µs/cm	pH
11.06.18	9	7.32	2.16	7.46	36.16	7.16	28.16	7.32
	12	7.16	3.42	7.32	72.16	7.52	56.16	7.36
	15	7.42	1.12	7.16	76.16	7.32	66.12	7.39
	18	7.36	1.28	7.49	99.16	7.12	78.16	7.14
12.06.18	9	7.16	2.16	7.32	98.16	7.42	76.16	7.46
	12	7.32	1.16	7.16	1.12	7.46	96.16	7.42
	15	7.46	2.42	7.42	1.28	7.42	776.12	7.16
	18	7.12	3.16	7.51	3.16	7.32	71.12	7.33
13.03.18	9	7.17	1.96	7.16	4.16	7.18	61.16	7.33
	12	7.28	2.16	7.32	3.16	7.32	1.96	7.49
	15	7.36	2.42	7.16	2.12	7.16	2.16	7.56
	18	7.16	1.16	7.32	23.16	7.34	19.16	7.32
14.06.18	9	7.42	2.12	7.42	71.16	7.16	56.16	7.52
	12	7.13	3.42	7.36	79.12	7.42	66.16	7.36
	15	7.39	2.96	7.52	88.16	7.36	76.42	7.42
	18	7.49	1.42	7.16	96.16	7.39	88.28	7.56

1.U Daily Testing report of Pre-HRSG and HRSG Water

Date	Time	Feed Water Before HRSG	Condensate Water	HRSG (HP/LP)	
	Hrs	pH	pH	pH	Conductivity µs/cm
11.06.18	9	9.32	9.56	9.16	121.16
	15	9.14	9.42	9.36	128.1
12.06.18	9	9.46	9.16	9.42	96.16
	15	9.16	9.32	9.36	126.16
13.06.18	9	9.32	9.42	9.12	92.14
	15	9.16	9.36	9.32	126.16
14.06.18	9	9.42	9.16	9.16	99.16
	15	9.32	9.28	9.12	94.16

1.V Air compressor Cycle time taking for 24 hr's

Hr	Compressor 1		Compressor 2	
	t, s	T, s	t, s	T, s
1	184	0.86	198	56
2	185	55	200	60
3	195	45	210	65
4	188	52	190	60
5	200	46	196	68
6	210	40	210	55
7	205	50	220	59
8	195	48	190	69
9	220	55	188	75
10	200	65	205	49
11	186	70	200	55
12	188	65	210	58
13	195	60	195	62
14	200	58	188	72
15	185	51	199	59
16	205	60	205	56
17	203	55	220	45
18	210	48	180	78
19	205	52	189	69
20	186	68	203	54
21	194	65	198	60
22	204	49	208	51
23	198	44	200	63
Average:	189.21	50.07	191.75	58.25

The following calculations shows the air compressor performance:

Volume rate of flow,

$$q_{FAD} = q_N x \frac{T_{FAD}}{T_N} x \frac{P_N}{P_{FAD}}$$

Or, q_N= (q_{FAD} x T_Nx P_{FAD}) / (T_{FAD} x P_N)

$= \frac{12.1x(273+0)x1.1}{1.031x(273+26)} = 11.99\ m^3$/min

Mass Flow, $\dot{m}$(kg/min) = q_N (m^3/min) x ρ (kg/m^3)

= 11.99 x 1.225 = 14.68775 kg/min

Isothermal Power, (kw) = $\frac{P_1 Q_f \log r}{36.7}$

Where, P_1=Absolute inlet pressure (kg/cm^2) = 1.031 kg/cm^2

r=compression ratio = 7.6/1.031=7.502

Q_f= Free air delivered (m^3/hr)=12.1x60= 726 m^3/hr

So, Isothermal Power, kw = $\frac{1.031\ x\ 726\ x \log 7.502}{36.7}$ =17.84 kw

Isothermal efficiency = $\frac{17.84}{58}\ x\ 100$ = 30.7%

Specific Power Consumption = $\frac{75}{726}$ = 0.11 kwh/ m^3

Compressor Idle Time, $t_I = \frac{V_R(P_{max} - P_{min})}{q} = \frac{10\ x\ (7.6-6.6)}{12.1}$ = 0.83 min

Compressor Running Time, $t_R = \frac{V_R(P_{max} - P_{min})}{\dot{V} - L_B} = \frac{10\ x\ (7.6-6.6)}{12.1-10.98}$ = 3.36 min

Motor Cycle Speed, A= $\frac{60}{t_I+t_R} = \frac{60}{4.19}$ = 14.32/hr

% leakage = $\frac{T}{T+t}$ x 100 = $\frac{.83}{3.36+0.83}$ x 100 = 19.80%

System leakage quantity, q (m^3/min) = $\frac{T}{T+t}$ x Q = 2.396 m^3/min

Leakage quantity per day = q x 24 x 60 = 2.396x24x60=3451.53 m^3/day

Energy lost due to leakage/day = 0.11 x 3451.5322 = 379.67 kwh/day

Leakage quantity per day = q x 24 x 60 = 2.81744x24x60=4057.1189 m^3/day

Energy lost due to leakage/day = 0.11 x 4057.1189 = 446.283 kwh/day

Average Energy lost due to leakage/day = (446.283+401.452)/2 = 423.8675 kwh/day

1.W Operating parameter of Circulating water pump, Cooling tower fan in summer season

Time	CW Pump				Cooling Tower Fan			
Hrs	Curr, I Amp	Voltage V	Power, W_{ele}, Kw	Mass flow of water, M_w, Kg/hr	Curr, I Amp	Voltage, V	Power, W_{ele}, Kw	Mass flow of air, M_a, Kg/hr
0	88.8	6300	794.54	10329007.85	165	400	98.31	1966166400
1	88.2	6300	789.17	10259217.26	165	400	98.31	1966166400
2	86.5	6300	773.96	10061477.24	166	400	98.90	1978082560
3	87.1	6300	779.33	10131267.84	167	400	99.50	1989998720
4	87.2	6300	780.22	10142899.6	166	400	98.90	1978082560
5	86.3	6300	772.17	10038213.71	165	400	98.31	1966166400
6	85.9	6300	768.59	9991686.65	164	400	97.71	1954250240
7	85.5	6300	765.01	9945159.588	164	400	97.71	1954250240
8	86.8	6300	776.64	10096372.54	169	400	100.69	2013831040
9	87.1	6300	779.33	10131267.84	165	400	98.31	1966166400
10	87.8	6300	785.59	10212690.2	168	400	100.10	2001914880
11	87.4	6300	782.01	10166163.13	166	400	98.90	1978082560
12	88.2	6300	789.17	10259217.26	167	400	99.50	1989998720
13	89.1	6300	797.22	10363903.15	168	400	100.10	2001914880
14	88.5	6300	791.85	10294112.56	168	400	100.10	2001914880
15	87.9	6300	786.49	10224321.96	168	400	100.10	2001914880
16	86.5	6300	773.96	10061477.24	169	400	100.69	2013831040
17	86.9	6300	777.54	10108004.31	169	400	100.69	2013831040
18	88.5	6300	791.85	10294112.56	170	400	101.29	2025747200
19	88.6	6300	792.75	10305744.32	170	400	101.29	2025747200
20	87.5	6300	782.91	10177794.9	171	400	101.88	2037663360
21	88.5	6300	791.85	10294112.56	171	400	101.88	2037663360
22	88.1	6300	788.28	10247585.49	169	400	100.69	2013831040
23	87.2	6300	780.22	10142899.6	169	400	100.69	2013831040
24	86.8	6300	776.64	10096372.54	166	400	98.90	1978082560
		Total:	19567.31				2493.46	

1.X Operating parameter of Closed Cycle Circulating water pump in summer season

Time	CCCW Pump			
Hrs	Curr, I Amp	Voltage, V	Power, W_{ele}, Kw	Mass flow of water in closed cycle, M_{cccw}, Kg/hr
0	177	400	100.55	703870.94
1	176	400	99.98	699894.27
2	173	400	98.28	687964.25
3	172	400	97.71	683987.58
4	170	400	96.57	676034.24
5	171	400	97.14	680010.91
6	175	400	99.41	695917.60
7	176	400	99.98	699894.27
8	177	400	100.55	703870.94
9	177	400	100.55	703870.94
10	177	400	100.55	703870.94
11	178	400	101.12	707847.62
12	179	400	101.68	711824.28
13	176	400	99.98	699894.27
14	176	400	99.98	699894.27
15	175	400	99.41	695917.60
16	177	400	100.55	703870.94
17	178	400	101.12	707847.62
18	177	400	100.55	703870.94
19	176	400	99.98	699894.27
20	176	400	99.98	699894.27
21	177	400	100.55	703870.94
22	177	400	100.55	703870.94
23	178	400	101.12	707847.62
24	179	400	101.68	711824.28
		Total	2499.62	

1.Y Cooling tower performance indicator parameter in summer season

The performance indicator parameter of cooling tower in summer season is shown in table below:

Time, Hrs	Tw2 ^0C	CCCW Outlet Press, MPa	Tw1 ^0C	CCCW Exchanger Outlet press, MPa	Ta ^0C	WBT, ^0C	Approach	Range	η	ε	Heat Load, D
0	33	0.32	39.2	0.3	29	28	5	6.2	55.35	0.49	64039849
1	29.8	0.32	37.5	0.3	30	27	2.8	7.7	73.33	0.37	78995973
2	30.5	0.32	37.8	0.3	31	27	3.5	7.3	67.59	0.51	73448784
3	30.6	0.32	37.9	0.3	30	28	2.6	7.3	73.74	0.33	73958255
4	30.8	0.32	38.1	0.3	28	28	2.8	7.3	72.28	0.28	74043167
5	31.5	0.32	38.1	0.3	28	28	3.5	6.6	65.35	0.35	66252211
6	31.6	0.32	38.1	0.3	29	28	3.6	6.5	64.36	0.40	64945963
7	31.7	0.32	38.1	0.3	29	28	3.7	6.4	63.37	0.41	63649021
8	31	0.32	38.4	0.3	28	28	3	7.4	71.15	0.29	74713157
9	31.1	0.32	38.4	0.3	28	28	3.1	7.3	70.19	0.30	73958255
10	31	0.32	38.1	0.3	29	28	3	7.1	70.30	0.33	72510100
11	30	0.32	37.8	0.3	29	27	3	7.8	72.22	0.34	79296072
12	31.3	0.32	37.3	0.3	30	27	4.3	6	58.25	0.59	61555304
13	31	0.32	38.1	0.3	29	28	3	7.1	70.30	0.33	73583712
14	32	0.32	38.1	0.3	30	27	5	6.1	54.95	0.62	62794087
15	32.2	0.32	38.1	0.3	31	27	5.2	5.9	53.15	0.73	60323500
16	32	0.32	38.1	0.3	32	28	4	6.1	60.40	0.66	61375011
17	32	0.32	38.1	0.3	33	28	4	6.1	60.40	0.78	61658826
18	32	0.32	38.1	0.3	34	28	4	6.1	60.40	0.98	62794087
19	30	0.32	38.1	0.3	35	28	2	8.1	80.20	0.65	83476529
20	32	0.32	38.1	0.3	36	27	5	6.1	54.95	2.38	62084549
21	32	0.32	38.1	0.3	37	27	5	6.1	54.95	4.55	62794087
22	32	0.32	39	0.3	38	28	4	7	63.64	4.00	71733098
23	32	0.32	38.1	0.3	29	28	4	6.1	60.40	0.44	61871688
24	30	0.32	38.1	0.3	30	28	2	8.1	80.20	0.25	82157487
Avg:	31.32	0.32	38.1	0.3	30.8	27.68	3.64	6.8	65.26	0.85	69120511

1.Z Cooling tower performance indicator parameter in summer season

The Heat loss by water, Volume of air required, heat gain by air and Mass of air required in cooling tower system in summer season is given in table below:

Cpw	Heat Loss By Water, HL Kj/Hr	Volume Of air required, V, Kj/hr	Heat Gain By air, HG, Kj/Hr	Mass Of Air Required M_{air}, Kg/hr
4.186	406429932.6	8186147.737	406429933	8830233.82
4.186	403683784.4	8130835.928	403683784	8770570.08
4.186	395903031.2	7974119.136	395903031	8601522.81
4.186	398649179.4	8029430.945	398649179	8661186.55
4.186	399106870.7	8038649.58	399106871	8671130.51
4.186	394987648.5	7955681.866	394987648	8581634.9
4.186	393156883	7918807.327	393156883	8541859.07
4.186	391326117.5	7881932.787	391326118	8502083.24
4.186	397276105.3	8001775.04	397276105	8631354.68
4.186	398649179.4	8029430.945	398649179	8661186.55
4.186	401853018.9	8093961.389	401853019	8730794.25
4.186	400022253.5	8057086.849	400022253	8691018.42
4.186	403683784.4	8130835.928	403683784	8770570.08
4.186	407803006.7	8213803.641	407803007	8860065.69
4.186	405056858.5	8158491.832	405056858	8800401.95
4.186	402310710.3	8103180.023	402310710	8740738.21
4.186	395903031.2	7974119.136	395903031	8601522.81
4.186	397733796.6	8010993.675	397733797	8641298.64
4.186	405056858.5	8158491.832	405056858	8800401.95
4.186	405514549.9	8167710.467	405514550	8810345.91
4.186	400479944.8	8066305.484	400479945	8700962.38
4.186	405056858.5	8158491.832	405056858	8800401.95
4.186	403226093	8121617.293	403226093	8760626.12
Avg:	400559543.3	8067908.725	400559543	8702691.76

1.AA Operating parameter of Circulating water pump, Cooling tower fan in winter season

Time	CW Pump				Cooling Tower Fan			
Hrs	Curr, I Amp	Voltage, V	Power, Wele, Kw	Mass flow of water, Mw, Kg/hr	Curr, I Amp	Voltage, V	Power, Wele, Kw	Mass flow of air, Ma, Kg/hr
0	86.2	6300	771.25	10026581.95	163	400	97.11	1942334080
1	84.9	6300	759.64	9875368.99	160	400	95.32	1906585600
2	84.5	6300	756.06	9828841.93	158	400	94.13	1882753280
3	83	6300	742.64	9654365.44	159	400	94.73	1894669440
4	83.2	6300	744.43	9677628.97	153	400	91.15	1823172480
5	83.1	6300	743.53	9665997.21	152	400	90.56	1811256320
6	83	6300	742.64	9654365.44	155	400	92.35	1847004800
7	83.2	6300	744.43	9677628.97	151	400	89.96	1799340160
8	83.1	6300	743.53	9665997.21	157	400	93.54	1870837120
9	82.5	6300	738.16	9596206.62	150	400	89.37	1787424000
10	84	6300	751.59	9770683.10	161	400	95.92	1918501760
11	83	6300	742.64	9654365.44	160	400	95.32	1906585600
12	84.9	6300	759.64	9875368.99	158	400	94.13	1882753280
13	84.5	6300	756.06	9828841.93	159	400	94.73	1894669440
14	80.5	6300	720.27	9363571.31	153	400	91.15	1823172480
15	80	6300	715.80	9305412.48	162	400	96.52	1930417920
16	79	6300	706.85	9189094.82	163	400	97.11	1942334080
17	78	6300	697.90	9072777.17	164	400	97.71	1954250240
18	83.2	6300	744.43	9677628.98	158	400	94.13	1882753280
19	83.1	6300	743.53	9665997.21	158	400	94.13	1882753280
20	82.5	6300	738.16	9596206.62	159	400	94.73	1894669440
21	82.6	6300	739.06	9607838.38	153	400	91.15	1823172480
22	81.5	6300	729.22	9479888.96	162	400	96.52	1930417920
23	81	6300	724.74	9421730.14	163	400	97.11	1942334080
24	81.6	6300	730.11	9491520.73	164	400	97.71	1954250240
		Total:	18486.45				2356.4	

1.BB Operating parameter of Closed Cycle Circulating water pump in winter season

CCCW Pump			
Curr, I Amp	Voltage, V	Power, Wele, Kw	Mass flow of water in closed cycle, Mcccw, Kg/hr
177	400	100.55	703870.94
176	400	99.98	699894.27
170	400	96.57	676034.24
175	400	99.41	695917.60
177	400	100.55	703870.94
178	400	101.12	707847.62
177	400	100.55	703870.94
171	400	97.14	680010.91
172	400	97.71	683987.58
172	400	97.71	683987.58
173	400	98.28	687964.25
173	400	98.28	687964.26
174	400	98.84	691940.93
175	400	99.41	695917.60
176	400	99.98	699894.27
175	400	99.41	695917.60
177	400	100.55	703870.94
178	400	101.12	707847.62
177	400	100.55	703870.94
172	400	97.71	683987.58
172	400	97.71	683987.58
173	400	98.28	687964.25
173	400	98.28	687964.25
178	400	101.12	707847.62
179	400	101.68	711824.28
	Total	2482.58	

1.CC Cooling tower performance indicator parameter in winter season

Time, Hrs	Tw2 ^{0}C	CCCW Outlet Press, MPa	Tw1 ^{0}C	CCCW Exchanger Outlet press, MPa	Ta, ^{0}C	WBT, ^{0}C	Approach	Range	η	ε	Heat Load, D
2	30.5	0.32	37.8	0.3	16	27	3.5	7.3	67.59	0.16	73194048.21
3	30.6	0.32	37.9	0.3	16	28	2.6	7.3	73.73	0.12	73194048.21
4	30.8	0.32	38.1	0.3	15	28	2.8	7.3	72.27	0.12	73194048.21
5	31.5	0.32	38.1	0.3	18	28	3.5	6.6	65.34	0.17	66175440.85
6	31.6	0.32	38.1	0.3	19	28	3.6	6.5	64.35	0.18	65172782.66
7	31.7	0.32	38.1	0.3	21	28	3.7	6.4	63.36	0.21	64170124.46
8	31	0.32	38.4	0.3	22	28	3	7.4	71.15	0.18	74196706.41
9	31.1	0.32	38.4	0.3	23	28	3.1	7.3	70.19	0.21	73194048.21
10	31	0.32	38.1	0.3	25	28	3	7.1	70.29	0.22	71188731.83
11	30	0.32	37.8	0.3	26	27	3	7.8	72.22	0.25	78207339.19
12	31.3	0.32	37.3	0.3	27	27	4.3	6	58.25	0.41	60159491.68
13	31	0.32	38.1	0.3	28	28	3	7.1	70.29	0.29	71188731.83
14	32	0.32	38.1	0.3	29	27	5	6.1	54.95	0.54	61162149.88
15	32.2	0.32	38.1	0.3	28	27	5.2	5.9	53.15	0.51	59156833.49
16	32	0.32	38.1	0.3	27	28	4	6.1	60.39	0.36	61162149.88
17	32	0.32	38.1	0.3	26	28	4	6.1	60.39	0.33	61162149.88
18	32	0.32	38.1	0.3	26	28	4	6.1	60.39	0.33	61162149.88
19	30	0.32	38.1	0.3	23	28	2	8.1	80.19	0.13	81215313.77
20	32	0.32	38.1	0.3	21	28	4	6.1	60.39	0.23	61162149.88
21	32	0.32	38.1	0.3	20	28	4	6.1	60.39	0.22	61162149.88
22	32	0.32	38.1	0.3	19	28	4	6.1	60.39	0.20	61162149.88
23	32	0.32	38.1	0.3	17	28	4	6.1	60.39	0.18	61162149.88
24	30	0.32	38.1	0.3	16	27	3	8.1	72.97	0.13	81215313.77
Avg:	31.2	0.32	38.0	0.3	21	27.72	3.56	6.74	65.45	0.24	67659374.98

1.DD Cooling tower performance indicator parameter in winter season

The Heat loss by water, Volume of air required, heat gain by air and Mass of air required in cooling tower system in summer season is given in table below:

Winter time			
Heat Loss By Water, HL Kj/Hr	Volume Of air required, V, Kj/hr	Heat Gain By air, HG, Kj/Hr	Mass Of Air RequiredMair, Kg/hr
251827632	2881998.813	244533529	3446479
248029768	2838534.794	240845668	3394502
246861194	2825161.249	239710942	3378509
242479043	2775010.458	235455718	3318535
243063329	2781697.23	236023081	3326532
242771186	2778353.844	235739400	3322534
242479043	2775010.458	235455718	3318535
243063329	2781697.23	236023081	3326532
242771186	2778353.844	235739400	3322534
241018325	2758293.527	234037310	3298544
245400477	2808444.319	238292534	3358518
242479043	2775010.458	235455718	3318535
248029768	2838534.794	240845668	3394502
246861194	2825161.249	239710942	3378509
235175457	2691425.806	228363679	3218580
233714740	2674708.875	226945271	3198588
230793306	2641275.014	224108455	3158606
227871871	2607841.153	221271639	3118624
243063329	2781697.23	236023081	3326532
242771186	2778353.844	235739400	3322534
241018325	2758293.527	234037310	3298544
241310469	2761636.914	234320992	3302543
238096891	2724859.667	231200494	3258562
241780439	2767015.404	234777349	3308974