ENERGY, EXERGY AND ENVIRONMENTAL IMPACT ANALYSES OF BARAPUKURIA COAL BASED THERMAL POWER PLANT

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ABSTRACT

Energy and exergy analyses of Barapukuria 2x125 MW coal-based thermal power plant have been carried out in order to evaluate the energetic and exergetic efficiencies and losses of different components and the overall system of the plant. This coal fired power plant, consuming about 2,000 metric tons of coal, generates about 170 MW to 180MW of electricity every day against the installation capacity of 250MW. The energy supply to demand narrowing down day by day around the world. The growing demand of power has made the power plants of scientific interest, but most of the power plants are designed by the energetic performance criteria based on The First Law of thermodynamics only. The real useful energy loss cannot be justified by The First Law of thermodynamics, because it does not differentiate between the quality and quantity of energy. So, this present study deals with the comparison of energy and exergy analyses of coal based thermal power plants. The entire plant cycle has been split up into three zones for the calculation: (1) only the turbo-generator with its inlets and outlets, (2) turbo-generator, condenser, feed pumps and the regenerative heaters, (3) the entire cycle with boiler, turbo-generator, condenser, feed pumps, regenerative heaters and the plant auxiliaries. The analyses have been done considering design data (50%, 80%, 100% and 106% loading condition) and operation data (57% and 67% loading condition) of this power plant. The overall energy efficiencies of the plant were 35.48%, 56.77%, 70.96% and 75.67% and the overall exergy efficiencies were 44.25%, 33.31%, 30.78 % and 30.21% for 50%, 80%, 100% and 106% (above 100%) loading condition for the design data. But the overall energy and exergy efficiencies of the power plant at operation data are 39.2%, 46.6% and 27.9 %, 27.2% for 57% and 67% loading condition which are lower than the design value. The environmental impact of the power plants is also studied by using specific emissions of CO_2, SOx, NOx and particulates. The distribution of the exergy losses in power plant components has been assessed to locate the process irreversibility. The comparison between the energy losses and the exergy losses of the individual components of the plant shows that the maximum energy losses of 49.92% occur in the condenser, whereas the maximum exergy losses of 68.27% occur in the boiler. The analyses have been also performed by inactivating heater one by one. Exergy analysis can be particularly effective in identifying ways to optimize the performance of existing operations and designing the plant while energy balance gives heat transfer between the system and its surrounding. Operation and maintenance decisions based on exergy analysis have proved to be more effective in reducing inefficiencies in operating power plant.

NOMENCLATURE

Symbols

B	Boiler
C	Condenser
D	Dearetor
G	Generator output power (kW)
P	Pump
h	Specific enthalpy (kJ/kg)
s	Specific entropy (kJ/kg K)
m	Mass flow rate
M	Molecular weight
n	Mass fraction
P	Power(kW)
T	Temperature (K)
e	Specific exergy(kJ/kg)
Cp	Specific heat of the fluid at constant pressure
En	Energy flow(kW)
Ex	Exergy flow(kW)
$\bullet_1$	Energy efficiency
$\bullet_2$	Exergy efficiency
$I_{destroyed}$	Irreversibility rate (kW)
W	Work done(kW)
Q	Heat transfer
Mpa	Mega-pascal
n_C	Mass fraction of carbon in coal
M_C	Molecular weight of carbon

Abbreviation

HPT	High pressure turbine
IPT	Intermediate pressure turbine
DFLP	Differential low pressure turbine
CW	Cooling water
CEP	Condensate extraction pump
BFP	Boiler feed pump
HTR	Heater
BPDB	Bangladesh Power Development Board
LHV	Lower Heating Value
MW	Mega Watt
Nm^3/h	Normal cubic Meter per hour

Subscripts

g	Flue gas
in	Inlet
j	Species /stream identification
out	Outlet
o	Reference state

Superscripts

a	air
f	fuel
g	Flue gas

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CHAPTER 1

INTRODUCTION

1.1 GENERAL

Energy consumption is the most important indicator showing the development stages of countries and living standards of communities. Population increment, urbanization, industrializing and technological development result directly in increasing energy consumption. This rapid growing trend brings about the crucial environmental problems such as contamination and greenhouse effect. Currently, approximately 80% of electricity in the world is produced from fossil fuels (coal, petroleum, fuel-oil, natural gas) fired thermal power plants, whereas 20% of the electricity is compensated from renewable sources such as hydraulic, wind, solar, geothermal and biogas [1]. The rate of depletion of fossil fuel reserves has necessitated the operation of power plants in the most efficient manner.

The general energy supply and environmental situation requires an improved utilization of energy sources. Therefore, the complexity of power-generating units has increased considerably. Plant owners are increasingly demanding a strictly guaranteed performance. This requires thermodynamic calculations of high accuracy. As a result, the expenditure for thermodynamic calculation during design and optimization has grown tremendously.

Energy and exergy analysis has increasingly attracted the interest to achieve the above goal. Generally, the performance of thermal power plants is evaluated through energetic performance criteria based on The First Law of Thermodynamics, including electrical power and thermal efficiency. In recent decades, the exergetic performance based on The Second Law of Thermodynamics has found as useful method in the design, evaluation, optimization and improvement of thermal power plants. The exergetic performance analysis can not only determine magnitudes, location and causes of irreversibilities in the plants, but also provides more meaningful assessment of plant individual component efficiency. These points of the exergetic performance analyses are the basic differences from energetic performance analysis. Therefore, it can be said that performing exergetic and energetic analyses together can give a complete depiction of system characteristics.

Energy is always conserved in every device or process. Unlike energy, exergy is not generally conserved but is destroyed. The majority of the causes of thermodynamic imperfection of thermal processes are not accounted for by energy or The First Law analysis. It is the exergy or second law analysis that accounts the irreversibilities like heat transfer through a finite temperature difference, chemical reactions, friction, mixing, and unrestrained expansion. The Second Law analysis of a power cycle enables us to identify the major sources of loss and shows avenues for performance improvement. Practical devices involving energy conversion and transfer always observe energy conservation law, but the quality of energy degrades i.e. work potential is lost or exergy is consumed (i.e., destroyed). Degradation of energy is equivalent to the irretrievable loss of exergy due to all real processes being irreversible. The loss of exergy or irreversibility provides a quantitative measure of process inefficiency [2].

The exergy consumption during a process is proportional to entropy creation, which accounts for inefficiencies due to irreversibilities. The potential for improvement in a given component is determined by its irreversibility rate under a given set of conditions in relation to the intrinsic irreversibility rate within the limits imposed by physical, technological, economic and other constraints.

The exergy consumption or order of destruction is a form of environmental damage. By preserving exergy through increased efficiency (i.e. degrading as little exergy as possible for a process), environmental damage is reduced [3]. Therefore, exergy analysis is as important as energy analysis for design, operation and maintenance of different equipment and systems of a power plant. It is important that the performance monitoring of an operative power station includes exergy analysis besides the conventional energy analysis. However, elaborate exergy analysis has not yet been practiced widely in power stations because of a lack of clearly defined codes and standards for this.

1.2 DEFINITION OF ENERGY AND EXERGY

Total energy consists of available energy plus unavailable energy. Considering flows of energy in a system, total energy is simply called energy and available energy is called exergy. Exergy flows to and from components however do not balance indicating a disappearance or "consumption" of exergy. This disappearance is really a conversion from available energy to unavailable energy. Consumption is a descriptive term indicating the loss of available energy. Components consume exergy by virtue of the ineffectiveness of their ability to transfer available energy. In order to compare the quality levels of various energy carriers, e.g. fuels, it is necessary to determine the equivalents of each energy quantity at a particular grade level. This can be done by using exergy concept, which overcomes the limitations of the first law of thermodynamics; and is based on both The First and The Second Laws of thermodynamics .

An exergy analysis can identify locations of energy degradation and rank them in terms of their significance. This knowledge is useful in directing the attention of process design, researchers, and practicing engineers to those components of the system being analyzed that offers the greatest opportunities for improvement. In order to perform the exergy analysis of the plant, the detail steam properties, mass, energy and exergy balances for the unit were conducted. The exergy values of each component are calculated by assuming that the component is in an open (control volume) system and there are only physical exergy associated with the material streams.

1.3 BACKGROUND

At present times, coal fired thermal power plants meet the growing energy demand with least fuel (coal) consumption. Electricity in particular, plays a vital role in developing the status of life. Coal plays a vital role in electricity generation worldwide. At present the electricity production by coal is about 41% of global electricity (Source: IEA 2011). But in Bangladesh the electricity production by coal is about 2.5% of total electricity production (Source: BPDB). Bangladesh has up to 2.7 billion short tons of high-quality coal reserves [4] and thus coal-based thermal power plants can play an important role for Bangladesh.

In Bangladesh, the first coal-fired power plant began commercial production at 250 MW at Barapukuria in Parbotipur in January, 2006. This coal fired power plant is operating on sub-critical steam conditions. An attempt has been made to predict the exergy, energy and environmental impact of this coal based thermal power plant. The environmental impact of the power plants is estimated in terms of specific emissions of CO_2, SOx, NOx and particulates.

1.4 OBEJECTIVES

The main objectives of this study are mentioned here:

i. Analyze the coal fired thermal power plant based on energy, exergy, and environmental impact analysis under Barapukuria 2x125 MW coal fired thermal power plant.

ii. Calculate the exergy and energy efficiency using the data from the plant at different loads as 50%, 80%, 100% & above 100%.

iii. Study the environmental impact of the power plant in terms of specific emissions of CO_2, SOx, NOx, and particulates.

1.5 SCOPE OF THE PRESENT WORK

In the present work, several visits to Barapukuria 2x125 MW thermal power plant, the only coal fired thermal power plant in Bangladesh, have been performed for collecting necessary data. A vigorous study of different equipment, machinery, operating conditions and flow diagrams of this power plant has also been performed during the site visit. The entire plant cycle have been split up into three zones for the analysis: (1) only the turbo-generator with its inlets and outlets, (2) turbo-generator, condenser, feed pumps and the regenerative heaters, (3) the entire cycle with boiler, turbo-generator, condenser, feed pumps, regenerative heaters and the plant auxiliaries. The mass, energy and exergy balance equations have been used to calculate energy and exergy efficiency of different parts of the plant. The load variation will be studied with the design data at 50, 80, 100 and above 100% and the operating data at 57% and 67% of full load. The environmental impact of 2x125 MW Barapukuria Coal Fired Thermal Power Plant will also be analyzed in this study.

CHAPTER 2

LITERATURE REVIEW

In the literature, there exist a number of papers concerning energetic and exergetic performances of coal-fired thermal power plants. For instance, Datta et al. [5] presented work on exergy analysis of a coal-based thermal power plant using the design data from a 210 MW thermal power plant under operation in India. In this the exergy efficiency is calculated using the operating data from the plant at different conditions, viz. at different loads, different condenser pressures, with and without regenerative heaters and with different settings of the turbine governing. The load variation is studied with the data at 100, 75, 60 and 40% of full load. Effects of two different condenser pressures, i.e. 76 and 89 mmHg (abs.), are studied. It is observed that the major source of irreversibility in the power cycle is the boiler, which contributes to exergy destruction of the order of 60%. Part load operation increases the irreversibilities in the cycle and the effect is more pronounced with the reduction of the load. Increase in the condenser back pressure decreases the exergy efficiency. Successive withdrawal of the high pressure heaters shows a gradual increment in the exergy efficiency for the control volume excluding the boiler.

Rosen [6] presented energy and exergy-based comparison of coal-fired and nuclear steam power plants. The results are reported of energy and exergy-based comparisons of coal-fired and nuclear electrical generating stations. A version of a process-simulation computer code, previously enhanced by the author for exergy analysis, is used. Overall energy and exergy efficiencies, respectively, are 37% and 36% for the coal-fired process, and 30% and 30% for the nuclear process.

Ganapathy et al. [7] determined the energy losses and the exergy losses of the individual

components of the lignite fired thermal power plant. This paper deals with an exergy analysis performed on an operating 50MWe unit of lignite fired steam power plant at Thermal Power Station-I, Neyveli Lignite Corporation Limited, Neyveli, Tamil Nadu, India. The exergy losses occurred in the various subsystems of the plant and their components have been calculated using the mass, energy and exergy balance equations. The distribution of the exergy losses in several plant components during the real time plant running conditions has been assessed to locate the process irreversibility. The First Law efficiency (energy efficiency) and The Second Law efficiency (exergy efficiency) of the plant have also been calculated. The comparison between the energy losses and the exergy losses of the individual components of the plant shows that the maximum energy losses of 39% occur in the condenser, whereas the maximum exergy losses of 42.73% occur in the combustor.
The real losses of energy which has a scope for the improvement are given as maximum exergy losses that occurred in the combustor.

Suresh et al. [8] provides insight to find out efficiency improvement in various components of a power generating system. The results of energy and exergy analysis carried out on a 62.5 MWe coal-based thermal power plant are presented in this paper. The performance of the plant was estimated by a component-wise modeling followed by a system simulation. A flow-sheet computer program, “Cycle-Tempo” was used for this study. The detailed break-up of exergy losses for the considered plant has been presented. The maximum exergy loss was found to be in steam generation unit followed by turbine. The variation of overall energy and exergy efficiency with parameters such as temperature gain of cooling water across the condenser, excess air and condenser pressure are presented. The off-design simulation resulted in an overall energy efficiency of 28.5% at 40% load factor compared to 31.5% at design rating.

Kiran and Karuna [9] examined to identify the magnitude, location and source of thermodynamic inefficiencies in thermal power plant. It is hoped that this examination as it includes both energy and exergy analyses will yield new insights into the performance of steam power plant. Exergy analysis can be particularly effective in identifying ways to optimize the performance of existing operations and designing the plant while energy balance gives heat transfer between the system and its surrounding.

Rudra et al. [10] examined to increase coal-fired steam power plant efficiency by advance steam parameters. This paper presents study of coal based thermal power plant using sub-critical, supercritical and ultra-supercritical steam conditions.

Boiler efficiency therefore has a great influence on heating- related energy savings. It is therefore important to maximize the heat transfer to the water and minimize the heat losses in the boiler. Heat can be lost from boilers by a variety of methods, including hot flue gas losses, radiation losses and, in the case of steam boilers, blow-down losses [11] etc. To optimize the operation of a boiler plant, it is necessary to identify where energy wastage is likely to occur. A significant amount of energy is lost through flue gases as all the heat produced by the burning fuel cannot be transferred to water or steam in the boiler. As the temperature of the flue gas leaving a boiler typically ranges from 150 to 250 ^{o}C, about 10–30% of the heat energy is lost through it. Since most of the heat losses from the boiler appear as heat in the flue gas, the recovery of this heat can result in substantial energy saving [12].This indicates that there is huge savings potentials of a boiler energy savings by minimizing its losses. Having been around for centuries, the technology involved in a boiler can be seen as having reached a plateau, with even marginal increase in efficiency painstakingly hard to achieve [13].

Bejan [14] draw outlines the fundamentals of the methods of exergy analysis and entropy generation minimization (or thermodynamic optimization-the minimization of exergy destruction). The paper begins with a review of the concept of irreversibility, entropy generation, or exergy destruction. Examples illustrate the accounting for exergy flows and accumulation in closed systems, open systems, heat transfer processes, and power and refrigeration plants.

Tapan et al. [15] presented a 500 MWe steam turbine cycle to identify the components that offer significant work potential saving opportunity. Criteria of performance relevant to the individual components are formulated. Exergy flows, exergy consumption due to irreversibilities and rational performance parameters for the turbine cycle and its components are computed by using plant operation data under different conditions.

Kaushik et al. [16] presented the comparison of energy and exergy analyses of thermal power plants stimulated by coal and gas. This article provides a detailed review of different studies on thermal power plants over the years. This review would also throw light on the scope for further research and recommendations for improvement in the existing thermal power plants.

Keeping in view the facts stated above, it can be expected that performing an analysis based on the same definition of performance criteria will be meaningful for performance comparisons, assessments and improvement for thermal power plants. Additionally, considering both the energetic and exergetic performance criteria together can guide the ways of efficient and effective usage of fuel resources by taking into account the quality and quantity of the energy used in the generation of electric power in thermal power plants. The purpose of this study presented here is to carry out energetic and exergetic performance analyses, at the design conditions, for the existing coal fired thermal power plant in order to identify the needed improvement. For performing this aim, thermodynamic models for the considered power plant have been summarized on the basis of mass, energy and exergy balance equations. The thermodynamic model simulation results are compared. In the direction of the comprehensive analysis results, the requirements for performance improvement are evaluated.

CHAPTER 3

DESCRIPTION OF THE PLANT

In this chapter a brief description of the process flow system of Barapukuria 2x125 MW coal fired thermal power plant has been given. Barapukuria power plant site is located at about 32 km southeast of the district headquarter of Dinajpur, about 9 km southwest of Fulbari upazila, over 11 km south of Parbatipur upazila and about 1 km north of the coal mine. The project is located at 88°57•2.4• E longitude and 25°33•18.2• N latitude. The area is situated at approximately 30 m above sea level. The schematic arrangement of equipments of this power plant is shown in figure 3.1. Coal fired thermal power plant generally operates on Rankine cycle.

The main components of the power plant are high, intermediate and low pressure turbines (HPT, IPT and DFLP), a boiler (B), number of pumps (P), a dearetor (D), a generator (G), a condenser (C), low and high pressure feed water heaters. The thermodynamic models of this power plant are based on fundamental mass, energy and exergy balances. Using the mass, energy and exergy balance equations for each component of the power plant, it is possible to compute energy and exergy flows at each node of the plants, energy and exergy efficiencies and irreversibilities of the component and so on.

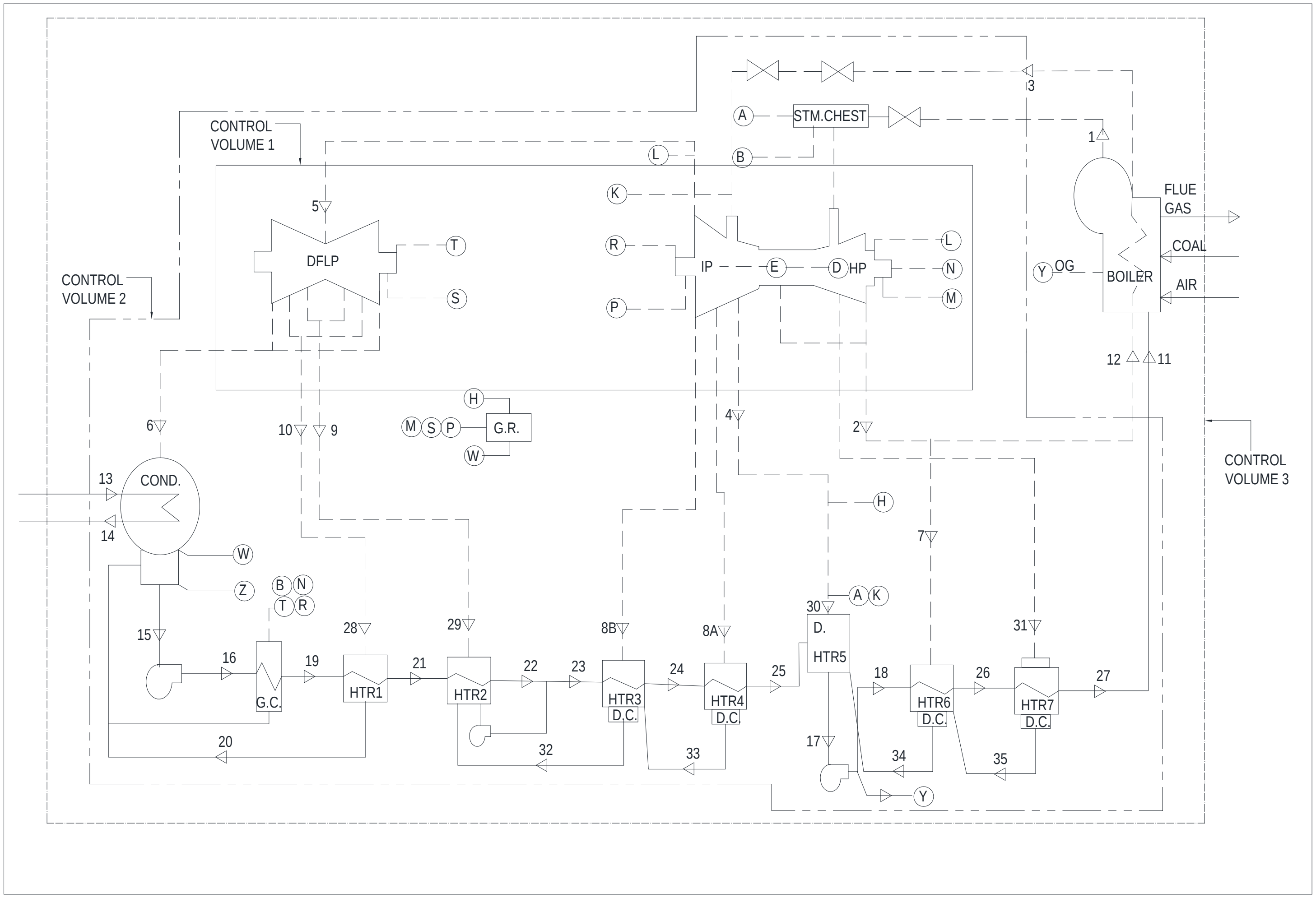

Figure 3.1 Flow diagram of 2x125 MW Barapukuria coal based thermal Power Plant

3.1 DESCRIPTION OF THE FLOW SYSTEM OF THE PLANT

Figure 3.1 describes the total cycle arrangement while Appendix B1 and Appendix B2 summarizes the salient flow and pressure, temperature and mass flow rate of design and operating data at different loading condition. The Power Plant has three turbines: high, intermediate and low pressure (HP, IP and LP) and is connected to the generator. The energy and exergy flows are computed using the plant operation design and operating data at different unit loads (electrical power output at the generator terminal). Steam flows to HP Turbine (point 1) with high energy and high exergy, after producing work on expansion in HP turbine, cold reheat steam (point 12) with low energy and exergy flows back to boiler for reheating, hot reheat steam (point 3) with high energy and exergy flows to IP Turbine and then LP Turbine, where further expansion takes place and work is produced. Wet steam (vapor fraction = 0.92) is exhausted from LP Turbine to condenser at a very low pressure, of the order of 86 kPa (abs). A large quantity of circulating water (CW) flows to the condenser (point 13) almost at ambient temperature, takes away heat of condensation and flows back to the river (point 14). The condensate exits the condenser (point 15) with low energy and almost negligible exergy and is pumped by the condensate extraction pump (CEP) to the deaerator through LP heaters (HTR1 to HTR4). Deaerator feeds (point 17) to BFP, which raise the pressure of feed water (flow 18) to sufficiently high value to flow through high pressure heaters (HTR6 and HTR7) and back to the boiler (point 11) for generation of steam and the cycle continues.

Thus, energy and exergy flows associated with the flow of the working fluid to the control region of the turbine cycle through three streams (point 1, 3 and 13), and from the control region through three streams (point 11, 12 and 14). Heat rejection to the environment is made possible at a low temperature by maintaining a low back pressure at the condenser. Final feed water (point 11) temperature rises across feed heaters by transferring heat from turbine extraction steam and facilitates high temperature heat addition in boiler.

Figure 3.2: Overall view of the Power Plant

3.2 THE STEAM GENERATION SECTION

This section includes furnace to burn the coal, boiler to produce high pressure steam at desired temperature, an economizer, and a superheater. Safety valves are also located at suitable points to avoid excessive boiler pressure. Heat produced due to burning of coal is utilized in converting water contained in boiler drum into steam at suitable pressure and temperature which is then passed through superheater. This boiler is a super-high pressure water tube reheat type boiler. Some salient features of this boiler are given below:

Model	SG-400/14.42-M772
Max. Cont. Rating (MCR)	400 t/h
Main Steam Outlet Press	14.42 mpa
Main Steam Outlet Temp.	538^0C
Reheated Steam Flow	325 t/h
Reheated Steam Press (in/Out)	2.31/2.21 mpa
Reheated Steam Temp (in/Out)	290 ^{o}C/538^{o}C
Fuel	Bituminus Coal
Combustion	4 corner firing
Feed Water Temp	248 ^{o}C
Rated Power	125 MW
Manufacturer	Shanghai Boiler Works Ltd.

Figure 3.3: Boiler sub-section

3.3 THE POWER PRODUCTION SECTION

The steam produced in the Steam Generation section is passed through a series of turbines (High pressure (H.P), Intermediate pressure (I.P) and Low pressure (L.P)) which are attached to a generator. Extraction steam from several points on the turbines preheats feed water in several low and high-pressure heat exchangers and one open deaerating heat exchanger. Then the low-pressure steam goes to the condenser.

Figure 3.4: Turbine sub-section

3.4 THE CONDENSATION SECTION

Cooling water from deep tube-well condenses the steam exhausted from the turbines. The flow rate of cooling water is adjusted so that a specified temperature rise in the cooling water is achieved across the condenser.

3.5 THE PREHEATING SECTION

Air taken from the atmosphere is first passed through the air pre-heater, where it is heated by flue gases. This hot air is then passed through the furnace. The flue gases after passing over boiler and superheater tubes flow through dust collector, economizer and air pre-heater before their exhaust to the atmosphere through chimney.

3.6 THE FEED HEATERS SECTION

A feed water heater is basically a heat exchanger where heat is transferred from the steam to the feed water either by the two fluid streams (open feed-water heaters) or without mixing them (closed feed-water heaters). Feed heaters in the turbine cycle do not involve the production or the input of work. For such systems the definition of second-law efficiency or exergy efficiency or rational efficiency refers to the accomplishment of the goal of the process relative to the process input, in terms of exergy changes or transfers. Loss of exergy in heat exchangers is caused by different forms of irreversibilities due to: heat transfer over a finite temperature difference, pressure loss, thermal interaction with the environment (which is neglected in this case), and stream wise conduction in the walls of the heat exchanger.

3.7 THE DEAERATOR SECTION

The primary purpose of deaerator, an open type feed heater is deaeration of feed water. Steady flow mixing process takes place in it, involving heat exchange with streams of water and steam. The irreversibilities are due to energy transfer between streams which initially are not in thermal or mechanical equilibrium, heat transfer with finite temperature gradients and viscous dissipation during mixing which results pressure drop between the inlets to the mixing space and the outlet.

3.8 ANALYSIS OF COAL:

The considered coal for this paper represents the typical coal of Bangladesh (Baropukuria coal mine) with lower heating value 22.7 MJ/Kg .(Source: Boiler Operation and Maintenance Manual of 2x 125 MW Barapukuria coal based Thermal Power Plant)

Proximate analysis (Source: BPDB)

Fixed Carbon : 48.4%

Volatile Matter : 29.2%

Ash : 12.4%

Total Moisture : 10.0%

Total Sulpher : 0.53%

Ultimate analysis (Dry ash free basis) (Source: BPDB)

Carbon: 83.0%

Hydrogen: 5.1%

Oxygen: 9.4%

Nitrogen: 1.7%

Sulpher: 0.77%

Total: 99.97%

3.9 DATA COLLECTION

In this analysis design and operating data of the power plant have been used. At 50%, 80%, 100% loading condition of design data are collected from BPDB Officials which are supplied by the Consultant FICHTNER of the Power Plant . Operating data of 67% and 57% loading condition are collected from the site visit on 13/04/2013 and 14/04/2013.

3.10 ENVIRONMENTAL IMPACT ASSESSMENT STUDY

The project consists of a coal fired power plant with two units of 125 MW each and two forced draft cooling tower stations of the wet type and two stacks of 100 m each. There is an ash pond of about 250 m^3. This ash pond is located within the area of the power plant site. The coal for firing the power plant come from the neighbour coal mine and is transported by conveyor belts of a length of about 1 km. The cooling water comes from deep wells (30-80 m).

Figure 3.5: Stack of the Power plant

The power plant authority operates 14 deep tube wells installed in Sherpur village adjacent to plant site for withdrawing the required water from the underground aquifer. The continuous withdrawal of huge amount of water from the ground water system, the water lable may fall. Naturally, the crisis becomes acute during the dry spells of the year when the precipitation rate is very low or absent.

During power plant operation, the following parts of the process could substantially influence the environment:

- Flue gas / turbine exhaust emissions
- Waste water discharges
- Residues from cleaning equipment and from operating supplies in the form of slude.
- Solid wastes
- Chemicals/consumables
- Oil deliveries in open water

Table 3.1: Some baseline data of the 2x 125 MW Barapukuria coal based Thermal Power Plant (Source: BPDB)

Coal flow	55 t/h
Ash production	6.6 t/h
Flue gas flow	700,000 Nm^3/h
Flue gas temperature	150 ^oC
Emission SOx on average	0.600 t/h
Emission SOx max	0.735 t/h
Emission NOx max	500 mg/Nm^3
Particulate matter	50 mg/Nm^3
Stack height	95m
Circulating cooling water	14,000 m^3/h

3.10.1 EMISSION ASPECTS

Sources of the emission of the power plant that have a major impact on the environment are:

- Flue gas
- Waste Heat
- Noise

The following major substances are generally contained in the flue gas of a power plant: H_2O, N_2, O_2, CO_2, CO, SO_2, SOx, NOx and particulate matter.

The combustion process of the pulverized coal in the boiler is a complicated non-linear phenomenon. The pollutants emitted from thermal power plants depend largely upon the characteristics of the fuel burned, temperature of the furnace and actual air used.

Carbon dioxide (CO_2) Emissions

CO_2 emissions are estimated based on the carbon content as obtained from the elemental analysis of the coal and the excess air used at the power plants. A small percentage of the carbon in the coal remains un-burnt due to factors, such as reactivity of the coal particles, air to fuel ratio, flame turbulence, fuel residence time etc. A small portion of the un-burnt carbon goes with the fly ash and the remaining un-burnt carbon goes in the bottom ash. Exact portion of un-burnt carbon can only be determined by experimental measurements.

Sulfer dioxide (SO_2) Emissions

SO_2 emissions from coal combustion mainly depends on the sulfur content in the coal unlike the emissions of CO_2 and NO which depends on the operating conditions and the design of the plant.

Emissions of oxides of Nitrogen

The formation of NO is influenced by the concentration of oxygen (which depends on the excess air) in the system and the flame temperature. NO emissions are estimated based on equilibrium reaction calculated at an average gas temperature of 1200 K. This is a theoretical ideal. In reality the gas temperature in the boiler varies from 1000K to 2500 K and the reaction also occurs in several phases. The estimates take into account the excess air used at the individual power plants. These estimates may be of limited value in describing details of NO formation but useful in establishing a baseline for NO emissions.

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3.10.2 AIR QUALITY

The most harmful substances which are emitted by a coal fired power plant are SOx, NOx and particulate matter. The emission limit values of SOx, NOx and particulate matter are given in Table 3.1 which are not to be exceeded by the power plant. The stack heights has been calculated according to the Bangladesh Environmental Conservation Rules as (Source: Study report on Environmental Impact Assessment of Barapukuria 2x125 MW coal based thermal power plant, BPDB)

$$H = 14 * (SO2\ emission\left(\frac{kg}{h}\right))^{0.3}$$

As shown in Table 3.1 the average SOx emission will be on average 600 kg/h. From that a stack height of

$$H = 14 * (600)^{.3}$$

$$H = 95.4m$$

In order to determine the ground level concentrations of SOx, NOx and particulate matter a propagation calculation has been studied on the basis of the meteorological situation prevailing at the sites within a radius of 1,425 m. The stack height was 95 m and the baseline data for calculation are shown in Table 3.1 (Source: Study report on Environmental Impact Assessment of Barapukuria 2x125 MW coal based thermal power plant, BPDB).The Bangladesh Air Quality Standards as shown in Table 3.3 are not to be exceeded neither by NOx, nor by SOx nor by particulate matter concentrations.

Table 3.2: Standards value for air (Source: BPDB)

Parameter	Maximum Standard value as per Bangladesh Air Quality Standards [mg/m3]
NOx	1
SOx	2
Particulate matter	0.5

Table 3.3: The maximum average concentrations can be located in a distance of about 4 km away from the Power Plant site. (Source: BPDB)

Parameter	Standard value [mg/m3]
NOx	0.1340 mg/m^3 max. average concentration
SOx	0.2300 mg/m^3 max. average concentration
Particulate matter	0.0075 mg/m^3 max. average concentration

3.10.3 POLLUTION CONTROL

For controlling the environmental pollutions caused due to the coal fired thermal power plant are described below:

(a) **Air pollution:** The air quality has been determined in the plant site at preconstruction phase. To conserve the same air quality, a chimney/stack of 100 meter high has been constructed for controlling the emission of SO_2 and NO_x. From this stack, flue gas has been released at a rate of 30 metre/second. So, from environmental requirement point, the stack height and emission rate has been controlled the SO_2 and NO_x emission is in the acceptable limit.

(b) **Ash Emission:** In the exit of the stack, there exists 99% efficient and low temperature Electrostatic Precipitator. For this there does not exist any possibility to mix the flying ash particle with the air. Moreover, there are huge ash storage has been made to store the emission ash particle to make it environment friendly. In this storage, the ash particle has been stored under water, so the adverse effect will not happen.

(c) **Water Pollution:** The polluted water from the cooling system of power plant and other sources are treated in the chemical water treatment plant and discharged in the nearest natural Canal. Furthermore, the water has been used for agriculture and the local people are benefitted.

(d) **Sound pollution:** The main source of noise in the power plant is turbine. In the plan to make the sound pollution beyond the standard limit, there introduce different sound insulator and items in the design and planning of the power plant. By taking this process into account, the maximum sound will be felt is 45 decibel in the power plant territory. Also, the equipment in the power plant has shock absorber and designed for resonance frequency, then it is free from the vibration effect due to the equipment.

CHAPTER 4

MATHEMATICAL MODELING

This chapter deals with the mathematical modeling of the proposed power plant for calculating overall and component energy and exergy efficiency.

The specific physical exergy of the stream was evaluated from the following equation:

$$e_{in} = (h_{in} - h_o) - T_o(s_{in} - s_o) = \Delta h - T_o \Delta s \quad (4.1)$$

The energy rate of a stream was obtained from its specific value as:

$$En = m_{in}(h_{in} - h_o) \quad (4.2)$$

4.1 OVERALL ENERGY AND EXERGY EFFICIENCY

For calculating the overall efficiency of the plant the control volume (3) shown in Fig. 4.1 is considered. Following additional considerations are made for this case, while including the entire plant into the control volume (3) of interest:

(i) Total auxiliary power consumption (P_{aux}) in the plant is considered in a consolidated manner as a percentage of the generated power.

(ii) Unaccounted heat loss from the system due to radiation and convection is neglected.

(iii) Exergy loss through ash is neglected.

(iv) Atmospheric pressure and temperature are same with those for reference environment, i.e.25°C temperature and 1 atmospheric pressure.

(v) The relative humidity of the ambient air is assumed to be 80%.

(vi) The kinetic and potential exergies have been neglected.

(vii) Incoming fuel temperature is 25°C.

The composition of the environment (mole %) is as follows [18]:

CO_2: 0.03

H_2O (g): 3.12

N_2: 75.65

O_2: 20.30

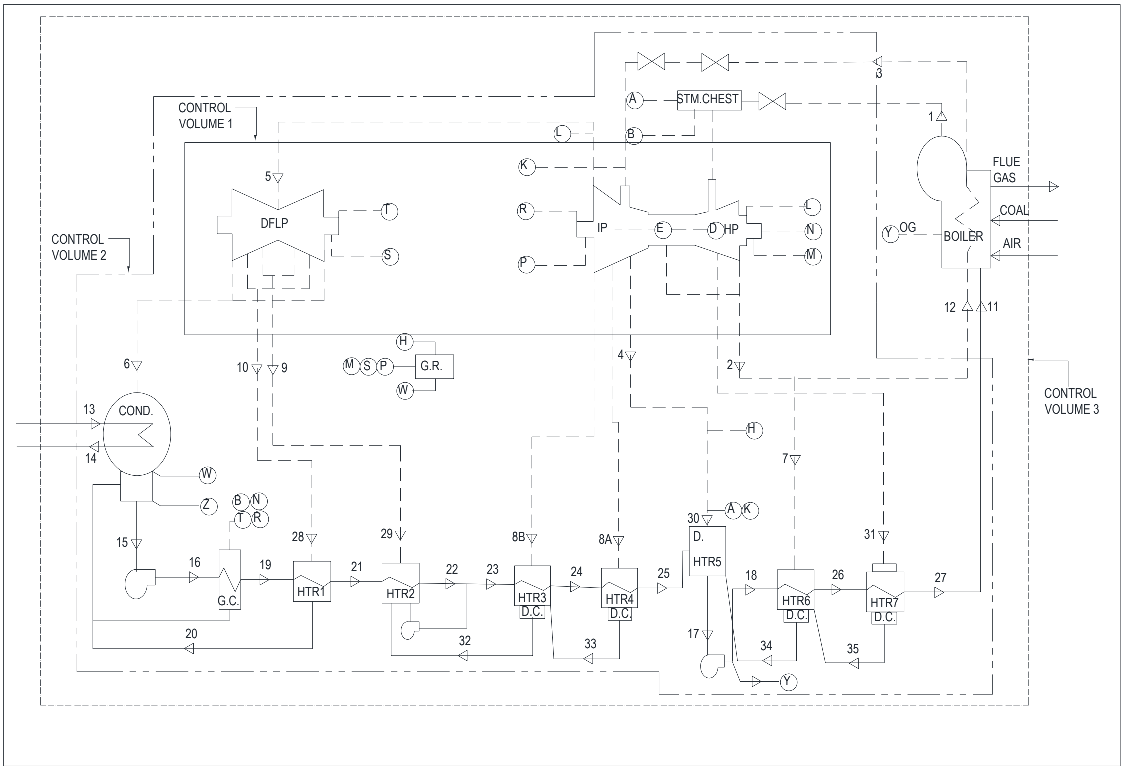

Figure 4.1 Schematic diagram of Control volume 3

4.1.1 OVERALL ENERGY EFFICIENCY

Considering the control volume 3, fuel (coal), air and circulating water comes into and circulating water and flue gas comes out the control volume 3 (Fig 4.1).The considered coal for this power plant represents the typical coal of Bangladesh (Barapukuria coal mine) with lower heating value 22.7 MJ/Kg (Source: BPDB).

The rate of energy entering the control volume with fuel:

$$En_i^f = m_{in}^f * LHV \tag{4.3}$$

where m_{in}^f is the mass flow rate of coal at the inlet and LHV is the Lower Heating Value of coal.

The total energy flow rate entering the control volume 3 is written as

$$En_{in} = En_{in}^f + En_{13} \tag{4.4}$$

Considering the energy input with fuel and circulating water.

The following composition (by mass) of coal has been considered for the analysis (Source: BPDB):

83% C, 5.1% H_2, 9.4% O_2, 1.7% N_2, Ash 12.4%, H_2O 10%

Considering the complete combustion of the coal in oxygen and that the hydrogen in coal (as H_2) is fully oxidized during combustion, The stoichiometric equation becomes as follows,

$$C + 0.37H_2 + 1.18\ O_2 = CO_2 + 0.37H_2O \tag{4.5}$$

The mass flow rate of flue gas can be written as (Neglecting the ash in flue gas)

$$m_g = (0.876\, m_{in}^{f} + m_{in}^{a}) \tag{4.6}$$

m_{in}^{a} is the designed value of the plant corresponding to different boiler loading.
Mass flow rate of moisture in flue gas:

m_g^{H2O}= moisture from coal + moisture from combustion of hydrogen in coal+ moisture from air = (moisture % in coal) x m_{in}^{f} + (% of H_2 in coal x m_{in}^{f}) + moisture in air

Considering the air temperature and the relative humidity, the mass flow rate of moisture in air is calculated from the psychrometric chart.

Mass flow rate of dry flue gas:

$$m_g^{dry} = m_g - m_g^{H2O} \tag{4.7}$$

The oxygen and carbon dioxide percentages at the air-heater inlet are used for calculating the mass flow rate of oxygen and CO_2 in the flue gas as

$$m_g^{O_2} = n_g^{O_2} X\ m_g^{dry} \tag{4.8}$$

$$m_g^{CO_2} = n_g^{CO_2} X\ m_g^{dry} \tag{4.9}$$

The mass flow rate of N_2 ($m_g^{N_2}$) in the flue gas is obtained by the difference between the equation 4.7 to summation of equation 4.8 and 4.9.

The energy flow rate leaving the control volume with flue gas is

$$En_{out}^{g} = \sum m_j^g h_j^g \tag{4.10}$$

In the equation (4.10), Energy summation is obtained from all the components of the flue gas (O_2, CO_2 and N_2).

Specific enthalpy calculation of flue gas respectively O_2, CO_2 and N_2 can be made by the following equation

$$h = mc_p \Delta T \tag{4.11}$$

The total energy flow rate leaving the control volume 3 including that with circulating water

$$En_{out} = En_{out}^{g} + En_{14} \tag{4.12}$$

The net power output from the control volume is

$$P_{net} = G - P_{aux} \tag{4.13}$$

The energy efficiency or The First Law efficiency of the control volume 3 is written as follows

$$\eta_1 = \frac{P_{net}}{En_{in} - En_{out}} \tag{4.14}$$

4.1.2 OVERALL EXERGY EFFICIENCY

Considering the control volume 3, fuel (coal), air and circulating water comes into and circulating water and flue gas comes out the control volume 3 (Fig 4.1).

The exergy of coal comprises of both chemical and thermo-mechanical components and is defined on the basis of a single step irreversible reaction in consideration of the ultimate analysis of coal. The following composition (by mass) of coal has been considered for the analysis (Source: BPDB):

83% C, 5.1% H_2, 9.4% O_2, 1.7% N_2, Ash 12.4%, H_2O 10%

Considering the complete combustion of the coal, the stoichiometric equation can be written as equation (4.5). By using the stoichiometric equation, the specific exergy of coal per mole of carbon (neglecting thermo-mechanical component) can be evaluated as follows,

$$e_{in}^{-f} = \left[\bar{g}C + 0.37\bar{g}H_2 + 1.18\bar{g}O_2 - \bar{g}CO_2 - 0.37H_20\right](T_o, P_o)$$
$$+ \bar{R}T_o \ln\left[\frac{\left(y_{O_2}^e\right)^{1.18} x\left(y_{H_2}^e\right)^{0.37}}{\left(y_{CO_2}^e\right) x\left(y_{H_20(g)}^e\right)^{0.37}}\right]$$

(4.15)

Where, the fuel is assumed to enter the system at room temperature.

The variables with bar (e.g. $\bar{g}$) designate the Gibbs functions of formation of the respective substances. The variables (e.g. e) represent the molar values of the respective quantities. For the specific molar entropy calculations in the above equation, the reference mole fractions of O_2, CO_2 and H_2O are taken from Moran and Shapiro [17]. Each term in the equation (4.15) has been explained in Moran and Shapiro [17].

The rate of exergy entering the control volume with fuel:

$$Ex_{in}^{f} = m_{in}^{f} n_c \frac{e_{in}^{-f}}{M_c} \tag{4.16}$$

where m_{in}^{f} is the mass flow rate of coal at the inlet, n_C is the mass fraction of carbon in coal and M_C is the molecular weight of carbon.

Exergy entering the control volume with air at reference temperature is zero. Considering the exergy input with fuel and circulating water, the total exergy flow rate entering the control volume is

$$Ex_{in} = Ex_{in}^{f} + Ex_{13} \tag{4.17}$$

By using the equation (4.6) ,(4.7), (4.8) and (4.9) the mass flow rate of flue gas, mass flow rate of moisture in flue gas, mass flow rate of dry flue gas and mass flow rate of O_2, N_2 and CO_2 in the flue gas have been calculated.

The exergy flow rate leaving the control volume with flue gas is

$$Ex_{out}^{g} = \sum m_j^{g} e_j^{g} \tag{4.18}$$

Where, the summation applies over all the components of the flue gas (O_2, CO_2 and N_2). In the equation (4.18), specific enthalpy and specific entropy calculation of flue gas respectively O_2, CO_2 and N_2 can be made by the following equation

$$hj = mjc_p \Delta T \tag{4.19}$$

$$ej = (hj - h_o) - To(sj - s_o) \tag{4.20}$$

The total exergy flow rate leaving the control volume including that with circulating water

$$Ex_{out} = Ex_{out}^{g} + Ex_{14} \tag{4.21}$$

The net power output from the control volume is

$$P_{net} = G - P_{aux} \tag{4.22}$$

The exergy efficiency or The Second Law efficiency of the control volume 3 is evaluated accordingly using the following Equation

$$\eta_2 = \frac{P_{net}}{Ex_{in} - Ex_{out}} \tag{4.23}$$

4.2 ENERGY ANALYSIS FOR THE COMPONENT

In an open flow system there are three types of energy transfer across the control surface namely work transfer, heat transfer, and energy associated with mass transfer and/or flow. The First Law of thermodynamics or energy balance for the steady flow process of an open system is given by:

$$\sum Q_k + m\left(h_j + \frac{\dot{C}_i^2}{2} + gZ_i\right) = m\left(h_o + \frac{C_o^2}{2} + gZ_o\right) + W \tag{4.24}$$

where Q_k is heat transfer to system from source at temperature $T_{k,}$ and W is the net work developed by the system. The other notations C is the bulk velocity of the working fluid, Z, is the altitude of the stream above the sea level, g is the specific gravitational force.

To analyze the possible realistic performance, a detailed energy analysis of the coal fired thermal power plant system has been carried out by ignoring the kinetic and potential energy change.

To calculate specific enthalpy and specific entropy, thermodynamic property tables are used for water and steam.

4.2.1 BOILER SUB-SECTION

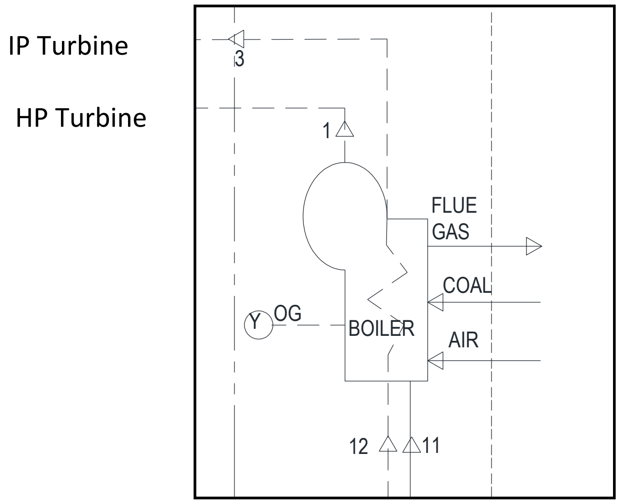

Figure 4.2 Boiler Sub-section

The energy balance for boiler:

The energy balance for the combustion/boiler is give by:

$$0 = En_{in}^{f} - \mathrm{m}_1(\mathrm{h}_1 - \mathrm{h}_{11}) - \mathrm{m}_3(\mathrm{h}_3 - \mathrm{h}_{12}) - \text{Energy loss}$$

$$\text{Energy loss} = En_{in}^{f} - \mathrm{m}_1(\mathrm{h}_1 - \mathrm{h}_{11}) - \mathrm{m}_3(\mathrm{h}_3 - \mathrm{h}_{12}) \tag{4.25}$$

The First Law efficiency is defined as

$$\eta_{1,\text{Boiler}} = 1 - \left(\frac{\text{Energy loss}}{\text{energy Input}}\right)$$

$$= \frac{\mathrm{m}_1(\mathrm{h}_1 - \mathrm{h}_{11}) - \mathrm{m}_3(\mathrm{h}_3 - \mathrm{h}_{12})}{En_{in}^{f}} \tag{4.26}$$

4.2.2 TURBINE SUB-SECTION

The energy balance for the Turbine cycle (Control volume 2)

The energy flow rate entering the control volume

$$\mathrm{E_{in}} = \mathrm{m_1 h_1} + \mathrm{m_3 h_3} + \mathrm{m_{13} h_{13}} \tag{4.27}$$

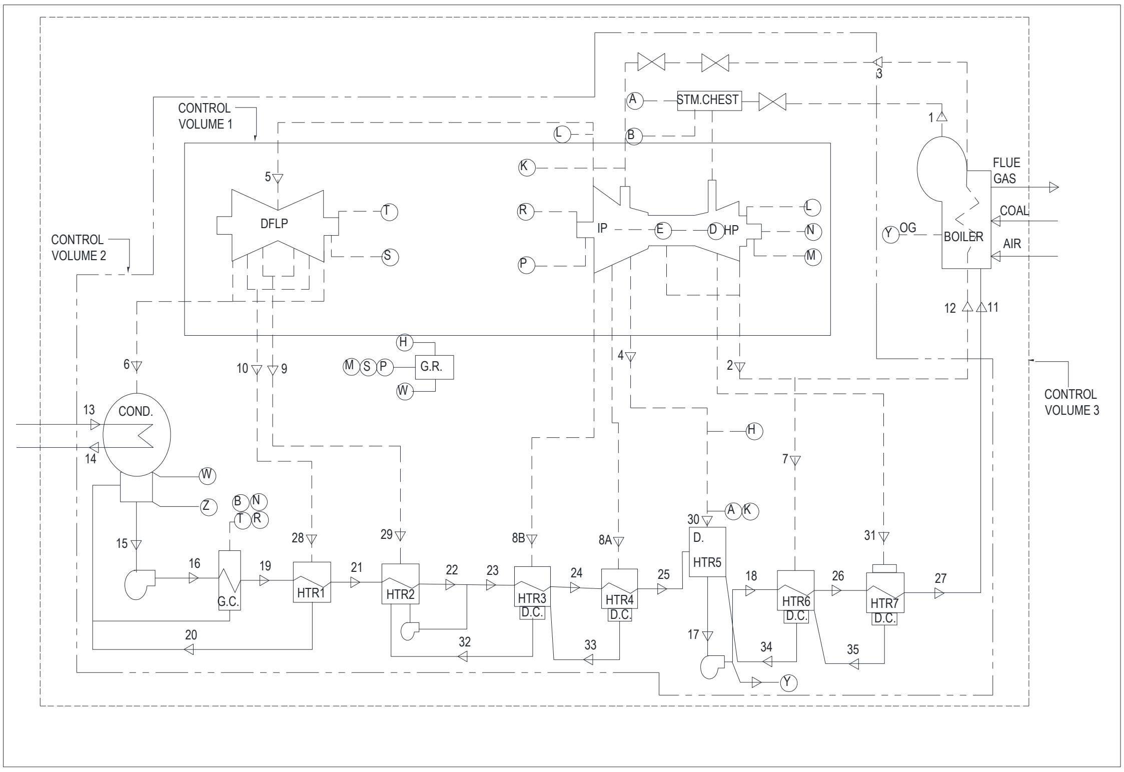

Figure 4.3 Turbine Sub-section (Control volume 2)

The energy flow rate leaving the control volume

$$\mathrm{En_{out}} = \mathrm{m_{11} h_{11}} + \mathrm{m_{12} h_{12}} + \mathrm{m_{14} h_{14}} \tag{4.28}$$

$$\mathrm{En_{14}} - \mathrm{En_{13}} = Q_k = Q_{rej} = \mathrm{m_c C_p \Delta T} \tag{4.29}$$

$$\mathrm{En_{in}} - \mathrm{En_{out}} = \mathrm{En_1} + \mathrm{En_3} + \mathrm{En_{13}} - \mathrm{En_{11}} - \mathrm{En_{12}} - \mathrm{En_{14}} \tag{4.30}$$

Pumping power inputs to BFPs and CEPs are accounted in the net power output from the control volume in this case. Neglecting the changes in kinetic energy and potential energy, the energy balance across the pumps gives

$$P_{in}\,(CEP) = m_{16}h_{16} - m_{15}h_{15} \tag{4.31}$$

$$P_{in}\,(BFP) = m_{18}h_{18} - m_{17}h_{17} \tag{4.32}$$

The net power output from the control volume is

$$P_{net} = G - P_{in}\,(CEP) - P_{in}\,(BFP) \tag{4.33}$$

$$\eta_{1,Turbine\ \ cycle} = \frac{P_{net}}{En_{in} - En_{out}} \tag{4.34}$$

4.2.3 CONDENSER SUB-SECTION

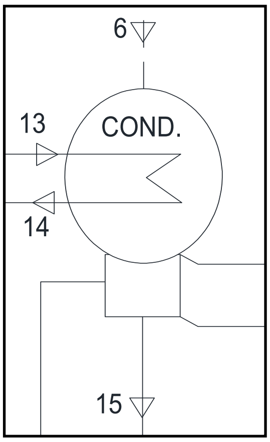

Figure 4.4 Condenser Sub-section

The energy balance for the condenser is give by:

$$0 = m_6*(h_6 - h_{15}) - Q_{rej} - \text{Energy loss}$$

$$Q_k = Q_{rej} = m_c C_p \Delta T \tag{4.35}$$

This gives:

$$\text{Energy loss} = m_6*(h_6 - h_{15}) - Q_{rej} \tag{4.36}$$

The First Law efficiency is:

$$\eta_{1,Condenser} = 1 - \frac{\text{Energy loss}}{m_6 * (h_6 - h_{15})} \tag{4.37}$$

4.2.4 PUMP SUB-SECTION

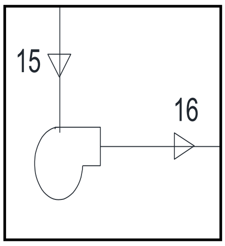

Figure 4.5 Condensate Extraction Pump, CEP Sub-section

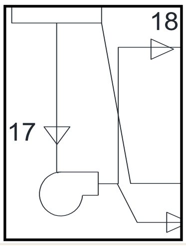

Figure 4.6 Boiler Feed Pump, BFP Sub-section

(i) The energy balance for the low pressure pump (Condensate Extraction Pump, CEP) is given by:

$$W_{CEP} = m_{15}(h_{15} - h_{16}) - \text{Energy loss}$$

$$\text{Energy loss} = m_{15}(h_{15} - h_{16}) + W_{CEP} \qquad (4.38)$$

The First Law efficiency is:

$$\eta_{1,CEP} = 1 - \frac{\text{Energy loss}}{W_{CEP}}$$

$$= \frac{m_{15}\ (h_{16} - h_{15})}{W_{CEP}} \qquad (4.39)$$

Considering, S_{16}=S_{15}

$$W_{CEP} = V\int_{15}^{16} dp \approx \text{V15(P16} - \text{P15)} \qquad (4.40)$$

(ii)The energy balance for the High pressure pump (Boiler Feed Pump, BFP) is given by:

$$-W_{BFP} = m_{17}(h_{17} - h_{18}) - \text{Energy loss}$$

$$\text{Energy loss} = m_{17}(h_{17} - h_{18}) + W_{BFP} \qquad (4.41)$$

The First Law efficiency is:

$$\eta_{1,BFP} = 1 - \frac{\text{energy loss}}{W_{BFP}}$$

$$= \frac{m_{17}\,(h_{18} - h_{17})}{W_{BFP}} \tag{4.42}$$

Considering, $S_{18}=S_{17}$

$$W_{BFP} = V\int_{17}^{18} dp \approx V17\,(P18 - P17) \tag{4.43}$$

4.2.5 FEED WATER HEATER SUB-SECTION

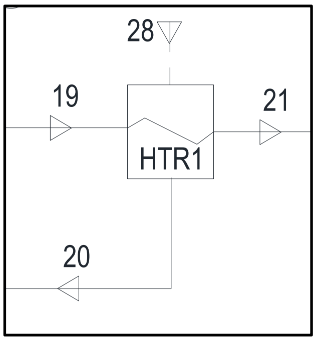

Figure 4.7 low pressure feed water heater (HTR1) Sub-section

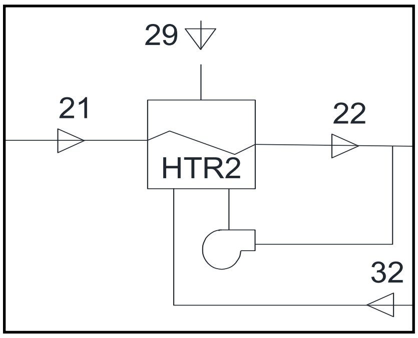

Figure 4.8 low pressure feed water heater (HTR2) Sub-section

(i) The energy flow equation for the low pressure feed water heater (HTR1) system becomes:

$$0 = m_{28}(h_{28} - h_{20}) - m_{19}(h_{21} - h_{19}) \text{ -Energy loss}$$

$$\text{Energy loss} = m_{28}(h_{28} - h_{20}) - m_{19}(h_{21} - h_{19}) \tag{4.44}$$

The First Law efficiency is:

$$\eta_{1,HTR\,1} = 1 - \frac{\text{energy loss}}{m_{28}\;(h_{28} - h_{20})}$$

$$= \frac{m_{19}\left(h_{21} - h_{19}\right)}{m_{28}\;\left(h_{28} - h_{20}\right)} \tag{4.45}$$

(ii) The energy flow equation for the low pressure feed water heater (HTR2) system becomes:

$$0 = (m_{29}h_{29} + m_{32}h_{32}) - m_{21}(h_{22} - h_{21}) \text{ -Energy loss}$$

$$\text{Energy loss} = (m_{29}h_{29} + m_{32}h_{32}) - m_{21}(h_{22} - h_{21}) \quad (4.46)$$

The First Law efficiency is:

$$\eta_{1,HTR\,2} = 1 - \frac{\text{energy loss}}{m_{29}\,h_{29} + m_{32}\,h_{32}}$$

$$= \frac{m_{21}\left(h_{22} - h_{21}\right)}{m_{29}\,h_{29} + m_{32}\,h_{32}} \quad (4.47)$$

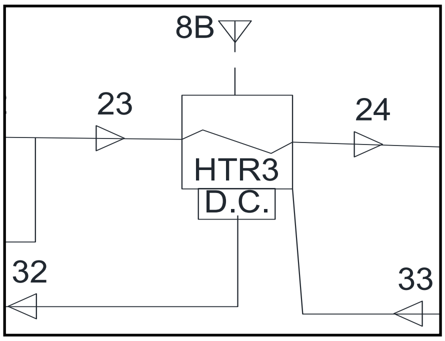

Figure 4.9 low pressure feed water heater (HTR3) Sub-section

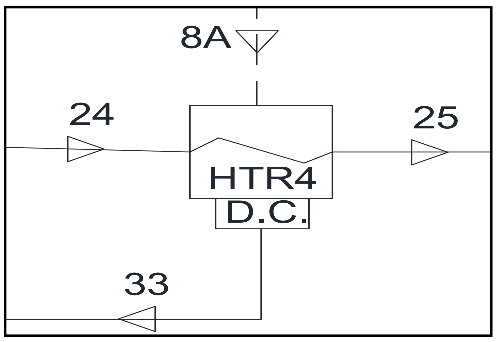

Figure 4.10 low pressure feed water heater (HTR4) Sub-section

(iii) The energy flow equation for the low pressure feed water heater (HTR3) system becomes:

$$0 = (m_8Bh_{8B} - m_{32}h_{32} + m_{33}h_{33}) - m_{23}(h_{24} - h_{23}) \text{ -Energy loss}$$

$$\text{Energy loss} = (m_{8B}h_{8B} - m_{32}h_{32} + m_{33}h_{33}) - m_{23}(h_{24} - h_{23}) \quad (4.48)$$

The First Law efficiency is:

$$\eta_{1,HTR\,3} = 1 - \frac{\text{energy loss}}{m_{8B}\,h_{8B} - m_{32}\,h_{32} + m_{33}\,h_{33}}$$

$$= \frac{m_{23}\left(h_{24} - h_{23}\right)}{m_{8B}\,h_{8B} - m_{32}\,h_{32} + m_{33}\,h_{33}} \quad (4.49)$$

(iv) The energy flow equation for the low pressure feed water heater (HTR4) system becomes:

$$0 = (m_{8A}h_{8A} - m_{33}h_{33}) - m_{24}(h_{25} - h_{24}) \text{ -Energy loss}$$

$$\text{Energy loss} = (m_{8A}h_{8A} - m_{33}h_{33}) - m_{24}(h_{25} - h_{24}) \quad (4.50)$$

The First Law efficiency is:

$$\eta_{1,HTR\,4} = 1 - \frac{\text{energy loss}}{m_{8A}\,h_{8A} - m_{33}\,h_{33}}$$

$$= \frac{m_{24}(h_{25} - h_{24})}{m_{8A}\,h_{8A} - m_{33}\,h_{33}} \quad (4.51)$$

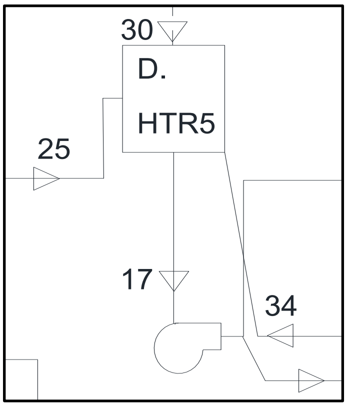

Figure 4.11 Dearetor Sub-section

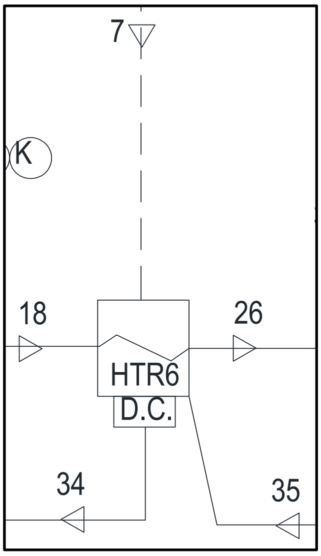

Figure 4.12 High pressure feed water heater (HTR6) Sub-section

(v) Dearetor sub system:

It is an adiabatic mixing chamber where a hot streams 30, 34 are mixed with a cold stream 25, forming a mixture 17, the energy supplied is the sum of the energies of the hot and cold streams, and the energy recovered is the energy of the mixture. The energy flow equation for the dearetor system becomes:

$$0 = m_{25}h_{25} + m_{30}h_{30} + m_{34}h_{34} - m_{17}h_{17} - \text{Energy loss}$$

where $m_{17} = m_{25} + m_{30} + m_{34}$

This gives:

$$\text{Energy loss} = m_{25}h_{25} + m_{30}h_{30} + m_{34}h_{34} - m_{17}h_{17} \quad (4.52)$$

The First Law efficiency is:

$$\eta_{1,Der} = 1 - \frac{\text{energy loss}}{m_{25}\,h_{25} + m_{30}\,h_{30} + m_{34}\,h_{34}}$$

$$= \frac{m_{17}\ h_{17}}{m_{25}\ h_{25} + m_{30}\ h_{30} + m_{34}\ h_{34}} \qquad (4.53)$$

(vi) The energy flow equation for the High pressure feed water heater (HTR6) system becomes:

$$0 = (m_7h_7 + m_{35}h_{35} - m_{34}h_{34}) - (m_{26}h_{26} - m_{18}h_{18}) \text{ -Energy loss}$$

$$\text{Energy loss} = (m_7h_7 + m_{35}h_{35} - m_{34}h_{34}) - (m_{26}h_{26} - m_{18}h_{18}) \qquad (4.54)$$

The First Law efficiency is:

$$\eta_{1,HTR\ 6} = 1 - \frac{\text{energy loss}}{m_7\ h_7 + m_{35}\ h_{35} - m_{34}\ h_{34}}$$

$$= \frac{m_{26}\ h_{26} - m_{18}\ h_{18}}{m_7\ h_7 + m_{35}\ h_{35} - m_{34}\ h_{34}} \qquad (4.56)$$

(vii) The energy flow equation for the low pressure feed water heater (HTR7) system becomes:

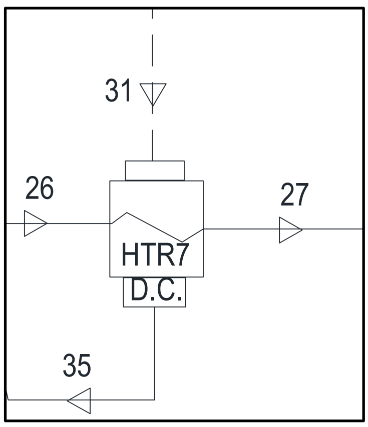

Figure 4.13 High pressure feed water heater (HTR7) Sub-section

$$0 = m_{31}(h_{31} - h_{35}) - m_{26}\ (h_{27} - h_{26}) \text{ -Energy loss}$$

$$\text{Energy loss} = m_{31}(h_{31} - h_{35}) - m_{26}\ (h_{27} - h_{26}) \qquad (4.57)$$

The First Law efficiency is:

$$\eta_{1,\mathrm{HTR}\,7} = 1 - \frac{\text{energy loss}}{m_{31}\, h_{31} - m_{35}\, h_{35}}$$

$$= \frac{m_{26}\left(h_{27} - h_{26}\right)}{m_{31}\, h_{31} - m_{35}\, h_{35}} \tag{4.58}$$

4.3 EXERGY ANALYSIS

Exergy is a generic term for a group of concepts that define the maximum possible work potential of a system, a stream of matter and/or heat interaction; the state of the (conceptual) environment being used as the datum state. In an open flow system there are three types of energy transfer across the control surface namely work transfer, heat transfer, and energy associated with mass transfer and/or flow. The work transfer is equivalent to the maximum work, which can be obtained from that form of energy.

Exergy of steady flow stream of matter is the sum of kinetic, potential and physical exergy. The kinetic and potential energy are almost equivalent to exergy. The physical specific exergy e_i and e_o depends on initial state of matter and environmental state. Energy analysis is based on the first law of thermodynamics, which is related to the conservation of energy. Second law analysis is a method that uses the conservation of mass and degradation of the quality of energy along with the entropy generation in the analysis design and improvement of energy systems. Exergy analysis is a useful method; to complement but not to replace energy analysis. The exergy flow for steady flow process of an open system is given by

$$\sum(1 - \frac{T_o}{T_K})Q_k + \sum_{in} m\,\dot{e_i} = e_w + \sum_{out} m\,\dot{e_{out}} + I_{destroyed} \tag{4.59}$$

Where $I_{destroyed} = T_o[S_{gen}]$, e_i and e_o are exergy associated with mass inflow and outflows respectively, e_w is useful work done on/by system, and $I_{destroyed}$ is irreversibility of process. The irreversibility may be due to heat transfer through finite temperature difference, mixing of fluids at different temperature and mechanical friction. Exergy analysis is an effective means, to pinpoint losses due to irreversibility in a real situation.

To analyze the possible realistic performance, a detailed exergy analysis of the coal fired thermal power plant has been carried out by ignoring the kinetic and potential energy change. For steady state flow the exergy balance for a thermal system is given as below

$$e_w = \sum_{k=1}^{n}(1 - \frac{T_o}{T_K})Q_k + \sum_{k=1}^{r}[(me)_i - (me)_o]_k - T_o S_{gen} \qquad (4.60)$$

where e_w represents the useful work done and/or by the system, the first term on the right hand side $(1 - \frac{T_o}{T_K})Q_k$ represents the exergy summation supplied through heat transfer, while changes in the exergy summation of the working fluid is represented by the second term $[(me)_i - (me)_o]_k$ where i and o refers the inlet and outlet states. On the other hand, the exergy destruction and/or the irreversibility in the system is given by the last term on the right hand side, $T_o S_{gen}$. The other notations such as, Q is the heat transfer rate, m is the mass flow rate of the working fluid, e is the exergy flow rate per unit mass, S_{gen} is the entropy generation rate, T_o is the ambient air temperature, T_k is the temperature of the heat source/sink at which the heat is transferred/rejected. The component wise exergy balance of the coal fire thermal power plant system is given as below.

4.3.1 BOILER SUB-SECTION

The exergy balance for the combustion/boiler is given by:

$$0 = m_p(e_i - e_o) - m_1(e_1 - e_{11}) - m_3(e_3 - e_{12}) - T_o S_{gen}$$

Where, m_p = mass flow rate of products after combustion

This gives

$$T_o S_{gen} = [\{m_p(h_i - h_o) - m_1(h_1 - h_{11}) - m_3(h_3 - h_{12})\} - T_o\{m_p(s_i - s_o) - m_1(s_1 - s_{11}) - m_3(s_3 - s_{12})\}] \qquad (4.61)$$

The Second Law efficiency is:

$$\eta_{2,\text{Boiler}} = 1 - \frac{I_{destroyed}}{m_p\,(e_i - e_o)}$$

$$= \frac{m_1(e_1 - e_{11}) + m_3(e_3 - e_{12})}{m_p\,(e_i - e_o)}$$

$$= \frac{m_1(e_1 - e_{11}) + m_3(e_3 - e_{12})}{Ex_{in}^{f}} \qquad (4.62)$$

4.3.2 TURBINE SUB-SECTION

The exergy balance for the turbine cycle is given by:

The exergy flow rate entering the control volume

$$Ex_{in} = Ex_1 + Ex_3 + Ex_{13} \qquad (4.63)$$

The exergy flow rate entering the control volume

$$Ex_{out} = Ex_{11} + Ex_{12} + Ex_{14} \qquad (4.64)$$

Because of the unavailability of required data for the calculation of exergy destruction in the condenser under all the operating conditions, the present study neglects the exergy destruction inside the condenser considering that the heat transfer in the condenser takes place with a very low temperature difference. Therefore, an exergy balance gives

$$Ex_{14} - Ex_{13} = Ex_6 - Ex_{15} \qquad (4.65)$$

The above equation is used to take care of the net exergy transfer rate with the cooling water

$$Ex_{in} - Ex_{out} = Ex_1 + Ex_3 + E_{x13} - Ex_{11} - Ex_{12} - Ex_{14} \qquad (4.66)$$

Pumping power inputs to BFPs and CEPs are accounted in the net power output from the control volume in this case. Neglecting the changes in kinetic energy and potential energy, the energy balance across the pumps gives

$$P_{in}\,(CEP) = m_{16}h_{16} - m_{15}h_{15} \qquad (4.67)$$

$$P_{in}\,(BFP) = m_{18}h_{18} - m_{17}h_{17} \qquad (4.68)$$

The net power output from the control volume is

$$P_{net} = G - P_{in}(CEP) - P_{in}(BFP) \tag{4.69}$$

$$\eta_{2,\text{Turbine cycle}} = \frac{P_{net}}{Ex_{in} - Ex_{out}} \tag{4.70}$$

4.3.3 CONDENSER SUB-SECTION

The exergy balance for the condenser is given by:

$$0 = m_6(e_{15} - e_6) - \sum_{k=1}^{n}(1 - \frac{T_o}{T_K})Q_k - T_o S_{gen}$$

$$Q_k = Q_{rej} = m_c C_p \Delta T \text{ and } T_k = T_{rej}$$

This gives:

$$T_o S_{gen} = m_6(e_{15} - e_6) - (1 - \frac{T_o}{T_{rej}})Q_{rej} \tag{4.71}$$

The Second Law efficiency is:

$$\eta_{2,\text{Condenser}} = 1 - \frac{I_{destroyed}}{m_6\,(e_{15} - e_{16})} \tag{4.72}$$

4.3.4 PUMP SUB-SECTION

(i) The exergy balance for the low pressure pump (Condensate Extraction Pump, CEP) is given by:

$$-W_{CEP} = m_{15}(e_{15} - e_{16}) - T_o S_{gen}$$

This gives,

$$I_{destroyed} = T_o S_{gen} = m_{15}(e_{15} - e_{16}) + W_{CEP} \tag{4.73}$$

Considering, $S_{16} = S_{15}$

$$W_{CEP} = V \int_{15}^{16} dp \approx V_{15}\,(P_{16} - P_{15}) \tag{4.74}$$

The Second Law efficiency is:

$$\eta_{2,CEP} = 1 - \frac{I_{destroyed}}{W_{CEP}}$$

$$= \frac{m_{15}\,(e_{16} - e_{15})}{W_{CEP}} \quad (4.75)$$

(ii) The exergy balance for the High pressure pump (Boiler Feed Pump, BFP) is given by:

$$-WBFP = m17(e17 - e18) - T_o S_{gen}$$

This gives,

$$I_{destroyed} = T_o S_{gen} = m_{17}(e_{17} - e_{18}) + W_{BFP} \quad (4.76)$$

The Second Law efficiency is:

$$\eta_{2,BFP} = 1 - \frac{I_{destroyed}}{W_{BFP}}$$

$$= \frac{m_{17}\,(e_{18} - e_{17})}{W_{BFP}} \quad (4.77)$$

Considering, S_{18}=S_{17}

$$W_{BFP} = V \int_{17}^{18} dp \approx V_{17}\,(P_{18}\text{-}P_{17}) \quad (4.78)$$

4.3.5 FEED WATER HEATER SUB-SECTION

(i) The exergy flow equation for the low pressure feed water heater (HTR1) system becomes:

$$0 = m_{28}\,(e_{28} - e_{18}) - m_{19}(e_{21} - e_{19}) - T_o S_{gen} \quad (4.79)$$

The irreversibility = exergy loss is

$$I_{destroyed} = T_o S_{gen}$$

The Second Law efficiency is:

$$\eta_{2,HTR\,1} = 1 - \frac{I_{destroyed}}{m_{28}\,(e_{28} - e_{20})}$$

$$= \frac{m_{19}\,(e_{21} - e_{19})}{m_{28}\,(e_{28} - e_{20})} \quad (4.80)$$

(ii) The exergy flow equation for the low pressure feed water heater (HTR2) system becomes:

$$0 = (\mathrm{m}_{29}\,\mathrm{e}_{29} + \mathrm{m}_{32}\,\mathrm{e}_{32}) - \mathrm{m}_{21}(\mathrm{e}_{22} - \mathrm{e}_{21}) - T_o S_{gen} \tag{4.81}$$

This gives,

The irreversibility = exergy loss is

$$I_{destroyed} \;=\; T_o S_{gen}$$

The Second Law efficiency is:

$$\eta_{2,\mathrm{HTR\,2}} = 1 - \frac{I_{destroyed}}{\mathrm{m}_{29}\mathrm{e}_{29} + \mathrm{m}_{32}\mathrm{e}_{32}}$$

$$= \frac{\mathrm{m}_{21}\ (\mathrm{e}_{22} - \mathrm{e}_{21})}{\mathrm{m}_{29}\mathrm{e}_{29} + \mathrm{m}_{32}\mathrm{e}_{32}} \tag{4.82}$$

(iii) The exergy flow equation for the low pressure feed water heater (HTR3) system becomes:

$$0 = \mathrm{m}_{8B}\mathrm{e}_{8B} - \mathrm{m}_{32}\mathrm{e}_{32} + \mathrm{m}_{33}\mathrm{e}_{33} - \mathrm{m}_{23}(\mathrm{e}_{24} - \mathrm{e}_{23}) - T_o S_{gen} \tag{4.83}$$

This gives,

The irreversibility = exergy loss is

$$I_{destroyed} \;=\; T_o S_{gen}$$

The Second Law efficiency is:

$$\eta_{2,\mathrm{HTR\,3}} = 1 - \frac{I_{destroyed}}{\mathrm{m}_{8B}\mathrm{e}_{8B} - \mathrm{m}_{32}\mathrm{e}_{32} + \mathrm{m}_{33}\mathrm{e}_{33}}$$

$$= \frac{\mathrm{m}_{23}(\mathrm{e}_{24} - \mathrm{e}_{23})}{\mathrm{m}_{8B}\mathrm{e}_{8B} - \mathrm{m}_{32}\mathrm{e}_{32} + \mathrm{m}_{33}\mathrm{e}_{33}} \tag{4.84}$$

(iv) The exergy flow equation for the low pressure feed water heater (HTR4) system becomes:

$$0 = \mathrm{m}_{8A}\mathrm{e}_{8A} - \mathrm{m}_{33}\mathrm{e}_{33} - \mathrm{m}_{24}(\mathrm{e}_{25} - \mathrm{e}_{24}) - T_o S_{gen} \tag{4.85}$$

This gives,

The irreversibility = exergy loss is

$$I_{destroyed} = T_o S_{gen}$$

The Second Law efficiency is:

$$\eta_{2,\mathrm{HTR}\,4} = 1 - \frac{I_{destroyed}}{\mathrm{m_{8A}e_{8A} - m_{33}e_{33}}}$$

$$= \frac{\mathrm{m_{24}(e_{25} - e_{24})}}{\mathrm{m_{8A}e_{8A} - m_{33}e_{33}}} \tag{4.86}$$

(v) Dearetor sub system:

It is an adiabatic mixing chamber where a hot streams 30, 34 are mixed with a cold stream 25, forming a mixture 17, the energy supplied is the sum of the exergies of the hot and cold streams, and the energy recovered is the energy of the mixture. The energy flow equation for the dearetor system becomes:

$$0 = \mathrm{m_{25}e_{25} + m_{30}e_{30} + m_{34}e_{34} - m_{17}e_{17}} - T_o S_{gen} \tag{4.87}$$

$$\text{where} \quad \mathrm{m_{17} = m_{25} + m_{30} + m_{34}}$$

This gives:

$$T_o S_{gen} = \mathrm{m_{25}e_{25} + m_{30}e_{30} + m_{34}e_{34} - m_{17}e_{17}}$$

The Second Law efficiency is:

$$\eta_{2,\mathrm{Der}} = 1 - \frac{I_{destroyed}}{\mathrm{m_{25}e_{25} + m_{30}e_{30} + m_{34}e_{34}}}$$

$$= \frac{\mathrm{m_{17}e_{17}}}{\mathrm{m_{25}e_{25} + m_{30}e_{30} + m_{34}e_{34}}} \tag{4.88}$$

(vi) The exergy flow equation for the low pressure feed water heater (HTR6) system becomes:

$$0 = \mathrm{m_7e_7 + m_{35}e_{35} - m_{34}e_{34} - (m_{26}e_{26} - m_{18}e_{18})} - T_o S_{gen} \tag{4.89}$$

The Second Law efficiency is:

$$\eta_{2,\mathrm{HTR}\,6} = 1 - \frac{I_{destroyed}}{\mathrm{(m_7e_7 + m_{35}e_{35} - m_{34}e_{34})}}$$

$$= \frac{\mathrm{(m_{26}e_{26} - m_{18}e_{18})}}{\mathrm{(m_7e_7 + m_{35}e_{35} - m_{34}e_{34})}} \tag{4.90}$$

(vii) The exergy flow equation for the low pressure feed water heater (HTR7) system becomes:

$$0 = m_{31}\,(e_{31} - e_{35}) - m_{26}\,(e_{27} - e_{26}) - T_o S_{gen} \tag{4.91}$$

The Second Law efficiency is:

$$\eta_{2,HTR\,7} = 1 - \frac{I_{destroyed}}{m_{31}(e_{31} - e_{35})}$$

$$= \frac{m_{26}(e_{27} - e_{26})}{m_{31}(e_{31} - e_{35})} \tag{4.92}$$

CHAPTER 5

RESULTS AND DISCUSSIONS

The design data of the plant components of Barapukuria 2x125 MW coal based thermal power plant have been used for the present energy analysis and exergy analysis to calculate the energy flow and exergy flow at different state points. The energy and the exergy efficiencies of these components have been determined using the equations given in the previous Chapter. Energy and exergy flow rates, for the complete power cycle are computed from the plant design data at approximately 100%, 80% and 50% of loading condition and the results are summarized in Appendix C1-C3.

5.1 T-s DIAGRAM OF THE TOTAL CYCLE

The total cycle of the power plant consists of six closed feed water and one open feed water heater (Dearetor) with the condensate trapped into the condenser. The T-s diagram is shown in Fig.5.1. For this cycle, the working fluid passes isentropically through the turbine stages and pumps, and there are no pressure drops accompanying the flow through the other components. The T-S diagram shows the principal states of the cycle. The steam does not expand to the condenser pressure in a single stage. The steam (Temp. 515^{o}C, 14.00 Mpa) enters the HP turbine at state 1 and expands to state 2, where a fraction of the total flow is extracted, or bled, into two closed feed water heater HTR6 and HTR7 (state 7 and state 31).The steam is then reheated. After reheating, the steam (Temp. 510^{o}C, 1.078 Mpa) enters the IP turbine at state 3 and expands to state 5. A fraction of the total flow is extracted, or bled, into one open feed water heater HTR5 (Dearetar, state 30) and two closed feed water heater HTR3 and HTR4 (state 8B and state 8A). The rest of the steam expands through the LP turbine to state 6.This portion of the total flow is condensed to saturated liquid at state 15.The mass flow rates of the streams entering the feed water heater are chosen so that the stream exiting the feed water heater is a saturated liquid at the extraction pressure. The liquid at state 15 is then pumped by the condensate extraction pump. Finally, after increasing the temperature by the feed water heater and increasing the pressure by the boiler feed pump to the steam generator pressure and enters the steam generator at state 27.The cycle is completed as the working fluid is heated in the steam generator at constant pressure from state 27 to 1.

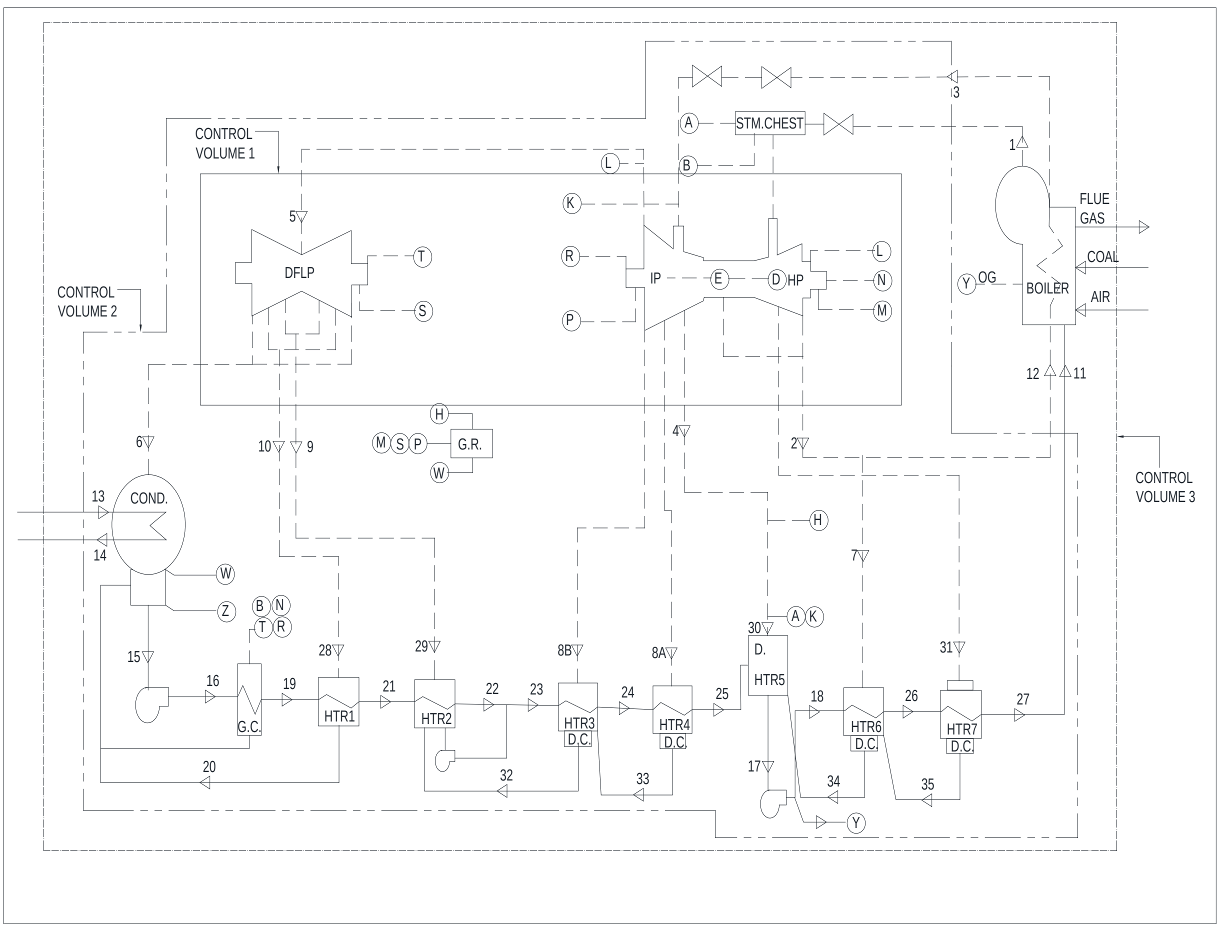

Figure: 5.1 Flow diagram of total power cycle

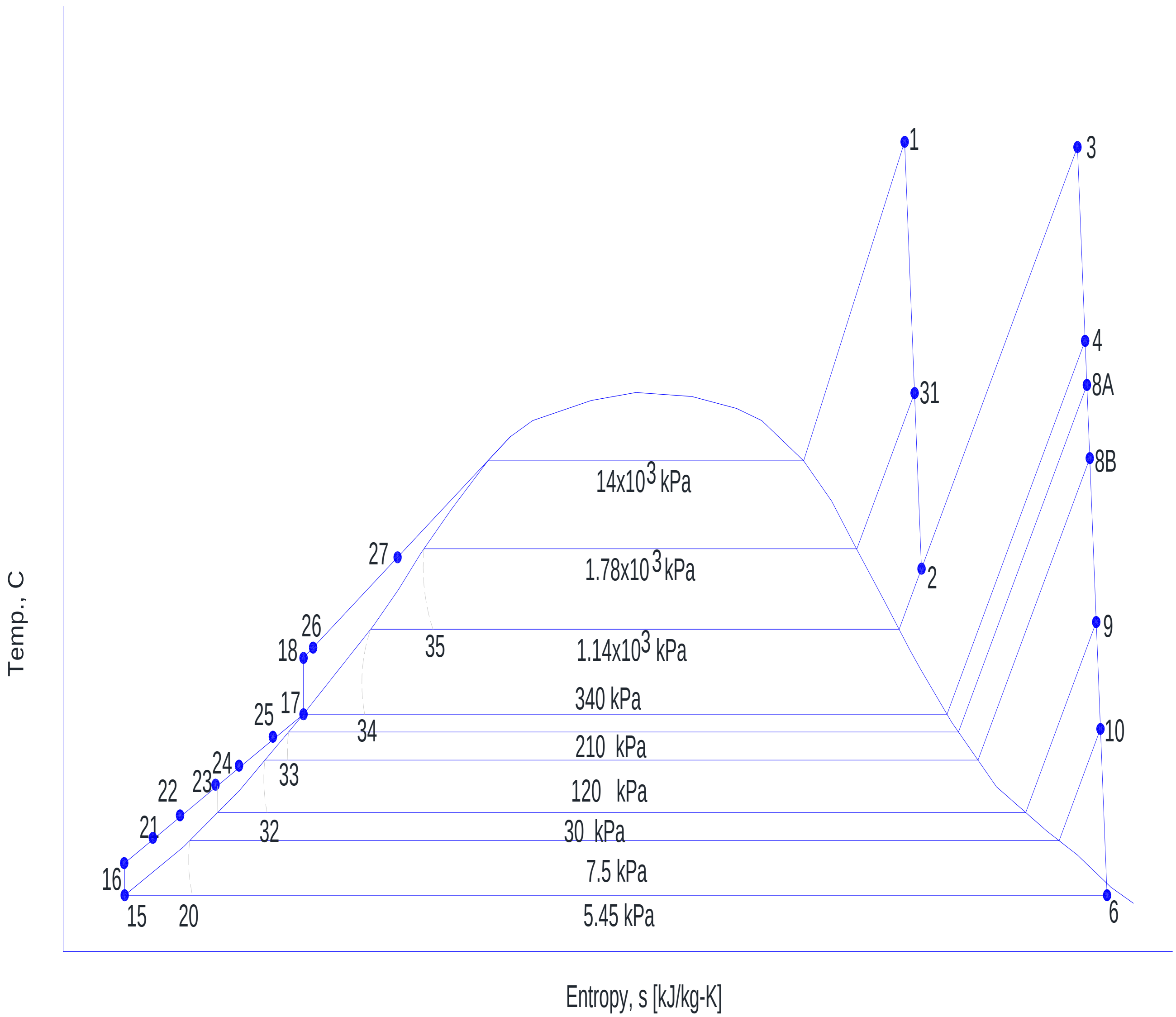

Figure: 5.2 T-s diagram of total power cycle

5.2 COMPARISON OF ENERGY AND EXERGY EFFICIENCY

5.2.1 DESIGN DATA

Energy and exergy efficiencies of the Overall power plant are derived from data summarized in Table 5.1 and shown in Fig. 5.3 as function of 100%, 80% and 50% loading condition. The analysis shows an increase in overall energy efficiency and decrease in exergy efficiency with increase in load percentage. It is evident from Fig.5.3 that operation of the plant below 56% of the designed capacity results in the significant increase of exergy efficiency and at that point energy and exergy efficiency is same. The decrease in exergy efficiency is attributed to the loss of exergy in the steam generation unit (Boiler) and turbine. There is a striking difference in the composition of the represented energy and exergy balances. It is noted that the exergy analysis has enabled the identification of the causes of process inefficiencies in detail when compared to the energy analysis.

Table 5.1: Energy and exergy efficiencies of the Overall power plant at design data

% Loading	Energy efficiency	Exergy efficiency
50	35.5	44.3
80	56.8	33.3
100	70.9	30.8
106	75.7	30.2

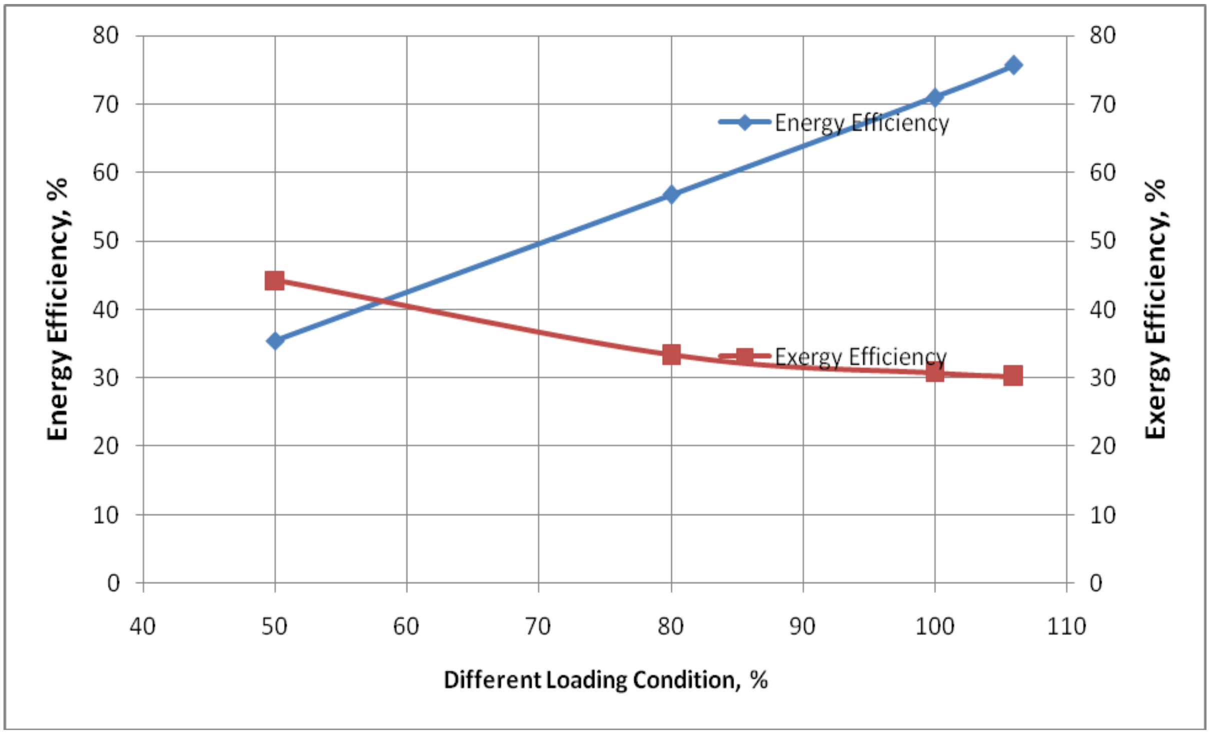

Figure: 5.3 Energy and exergy efficiencies of the Overall power plant

5.3.1 OPERATING DATA

Energy and exergy efficiencies of the Overall power plant are derived from operating data summarized in Table 5.2 and shown in Fig. 5.4 as function of 57% and 67% loading condition. From the comparison of design and operating data for the Energy and exergy efficiencies of the Overall power plant, it is found that the efficiency of the operating condition is low than the design condition. From the site visit, these are found on two different dates. Only two points data are available.

Table 5.2: Energy and exergy efficiencies of the Overall power plant at design data

% Loading	Energy efficiency	Exergy efficiency
57	39.2	27.9
67	46.6	27.2

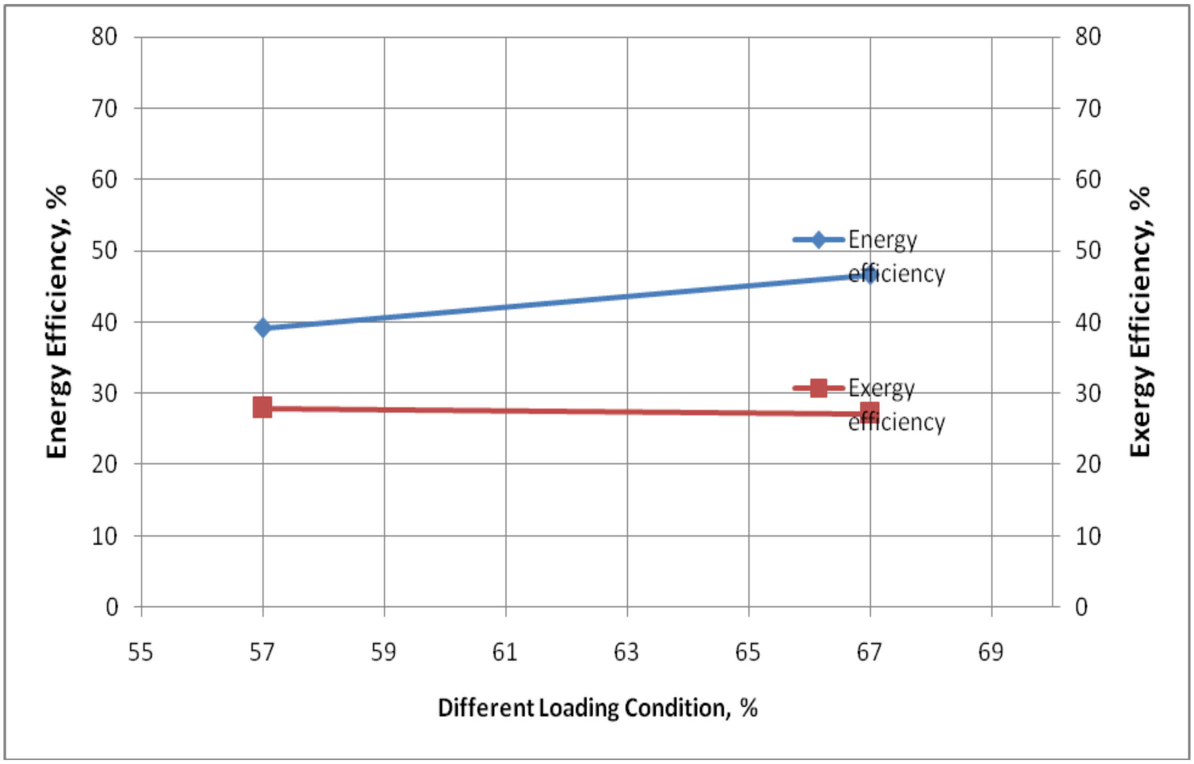

Figure: 5.4 Energy and exergy efficiencies of the Overall power plant at operating data

5.3 COMPONENTWISE ENERGY AND EXERGY

The comparison of energy efficiency and exergy efficiency between different components of the power plant is given in Tables 5.3-5.5 and Fig 5.5- 5.7 for different loading conditions.

5.3.1 MAXIMUM ENERGY LOSSES

From the Fig 5.8 it can be observed that the maximum energy loss (49.92% at 100% load) occurred in the condenser. Thus the First law analysis (energy analysis) diverts our attention towards the condenser for the plant performance improvement. Approximately half of the total plant energy losses occur in the condenser only and these losses are practically useless for the generation of electric power. Thus the analysis of the plant based only on the First law principles may mislead to the point that the chances of improving the electric power output of the plant is greater in the condenser by means of reducing its huge energy losses, which is almost impracticable. Hence the First law analysis (energy analysis) cannot be used to pinpoint prospective areas for improving the efficiency of the electric power generation. However, the Second law analysis (exergy analysis) serves to identify the true power generation inefficiencies occurring throughout the power station.

5.3.2 MAXIMUM EXERGY LOSSES

The comparison of exergy losses between the different subsystems of the plant are shown in fig 5.8 It can be noted that the maximum exergy loss occurs in the Boiler Subsystem (68.27% at 100% load). This may be due to the irreversibility of the combustion process in the combustor. while the exergy destruction rate of the condenser is only 0 .21%. The real loss is primarily back in the boiler where entropy was produced. Contrary to The First Law analysis, this demonstrates that significant improvements exist in the boiler system rather than in the condenser. The calculated exergy efficiency of the power cycle is 30.78% at 100% load. This indicates that tremendous opportunities are available for improvement. However, part of this irreversibility cannot be avoided due to physical, technological, and economic constraints.

5.3.3 ENERGY AND EXERGY EFFICIENCY OF TURBINE

Energy efficiency of the turbine cycle is low (47.25% at 100% load) due to a large quantity of energy rejection in the condenser. But the derived exergy efficiency of the turbine is high (83.14 % at 100% load), this is due to the reason that a little exergy associated with turbine exhaust steam enters condenser, part of which is rejected to CW and partly consumed due to irreversibilities.

Table 5.3: Different Component Energy and Exergy efficiency for 50% Load

Component	Energy Efficiency, $\bullet_1$	Exergy Efficiency, $\bullet_2$	Energy loss,%	Exergy loss,%
Boiler	97.3	32.9	2.7	67.0
Turbine cycle	54.4	81.4	45.6	18.6
Condenser	45.2	99.0	54.8	0.9
Condensate extraction pump,CEP	96.0	96.0	3.9	3.9
HTR1	93.4	0.0	6.6	0.0
HTR2	74.8	50.5	25.2	49.5
HTR3	100.0	78.6	0.0	21.5
HTR4	100.0	41.5	0.0	58.6
HTR5 (Deaerator)	100.0	91.3	0.0	8.7
HTR6	100.0	92.8	0.0	7.3
HTR7	100.0	87.3	0.0	12.7
Overall Plant	35.5	44.3	64.5	55.8

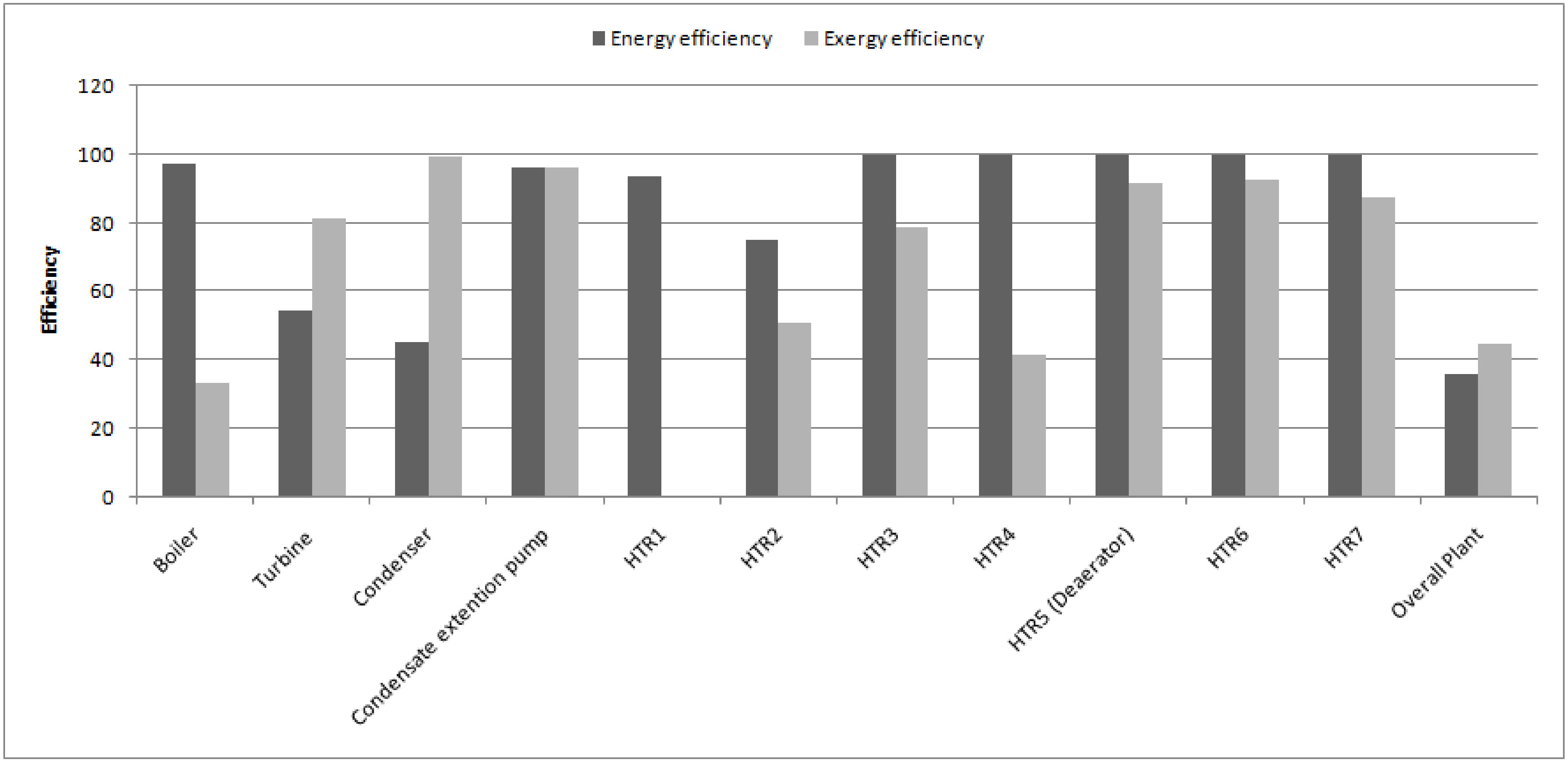

Figure: 5.5 Comparison of efficiency in the plant and components at 50% load

Table 5.4: Different Component Energy and Exergy efficiency for 80% Load

Component	Energy Efficiency, η_1	Exergy Efficiency, η_2	Energy loss,%	Exergy loss,%
Boiler	92.8	32.2	7.2	67.8
Turbine cycle	58.4	82.4	41.6	17.6
Condenser	49.1	99.6	50.9	0.4
Condensate extraction pump,CEP	95.4	95.4	4.6	4.6
HTR1	99.7	0.0	0.3	0.0
HTR2	72.2	63.6	27.8	36.5
HTR3	100.0	49.8	0.0	50.2
HTR4	100.0	66.4	0.0	33.6
HTR5 (Deaerator)	100.0	95.4	0.0	4.6
HTR6	100.0	79.9	0.0	20.1
HTR7	99.9	90.2	0.1	9.8
Overall Plant	56.8	33.3	43.2	66.7

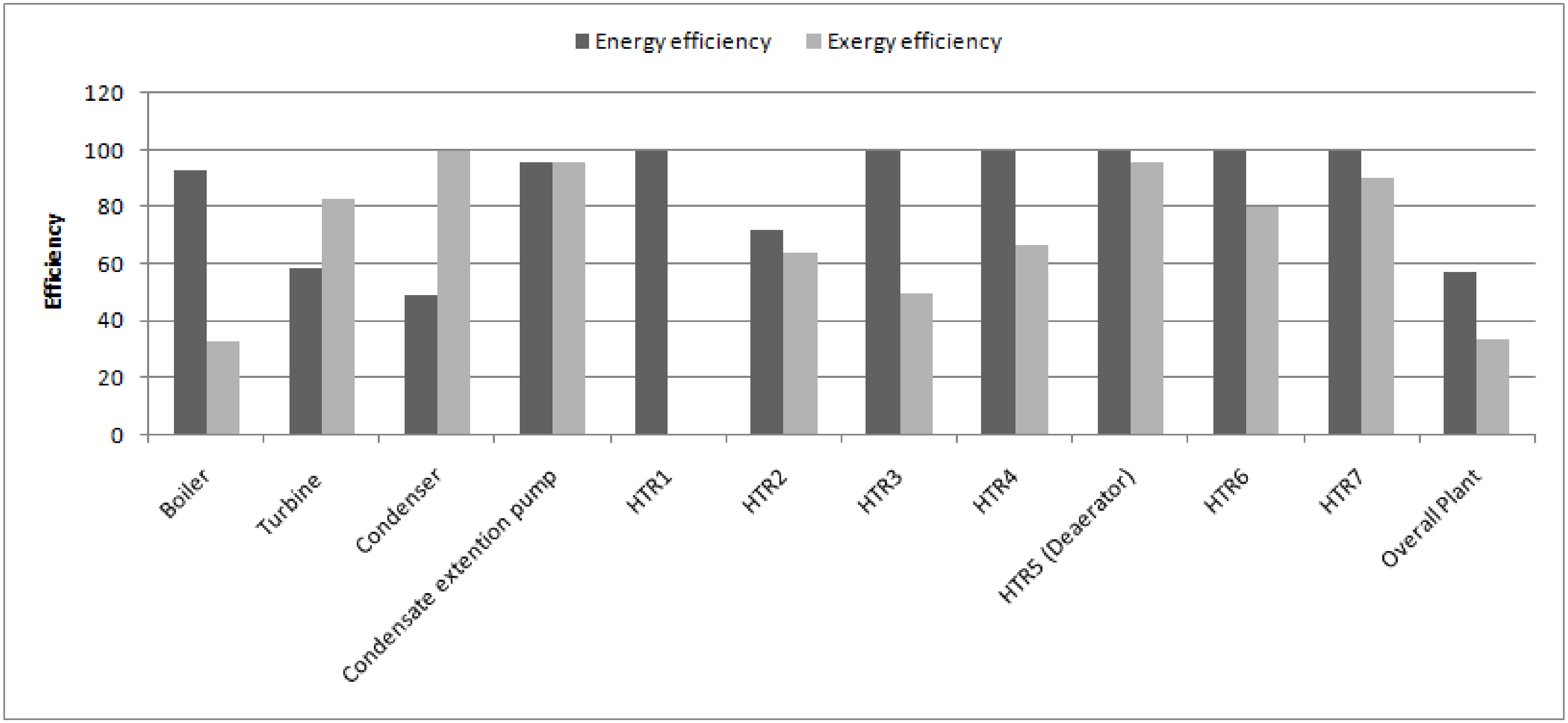

Figure: 5.6 Comparison of efficiency in the plant and components at 80% load

Table 5.5: Different Component Energy and Exergy efficiency for 100% Load

Component	Energy Efficiency, η_1	Exergy Efficiency, η_2	Energy loss	Exergy loss
Boiler	91.6	31.7	8.4	68.3
Turbine cycle	59.3	82.7	40.7	17.3
Condenser	50.1	99.8	49.9	0.2
Condensate extention pump	91.4	91.4	8.6	8.6
HTR1	100.0	79.5	0.0	20.5
HTR2	70.3	79.8	29.7	20.2
HTR3	100.0	69.4	0.0	30.6
HTR4	100.0	97.3	0.0	2.7
HTR5 (Deaerator)	100.0	98.2	0.0	1.8
HTR6	100.0	80.8	0.0	19.2
HTR7	99.9	99.9	0.1	0.1
Overall Plant	71.0	30.8	29.0	69.2

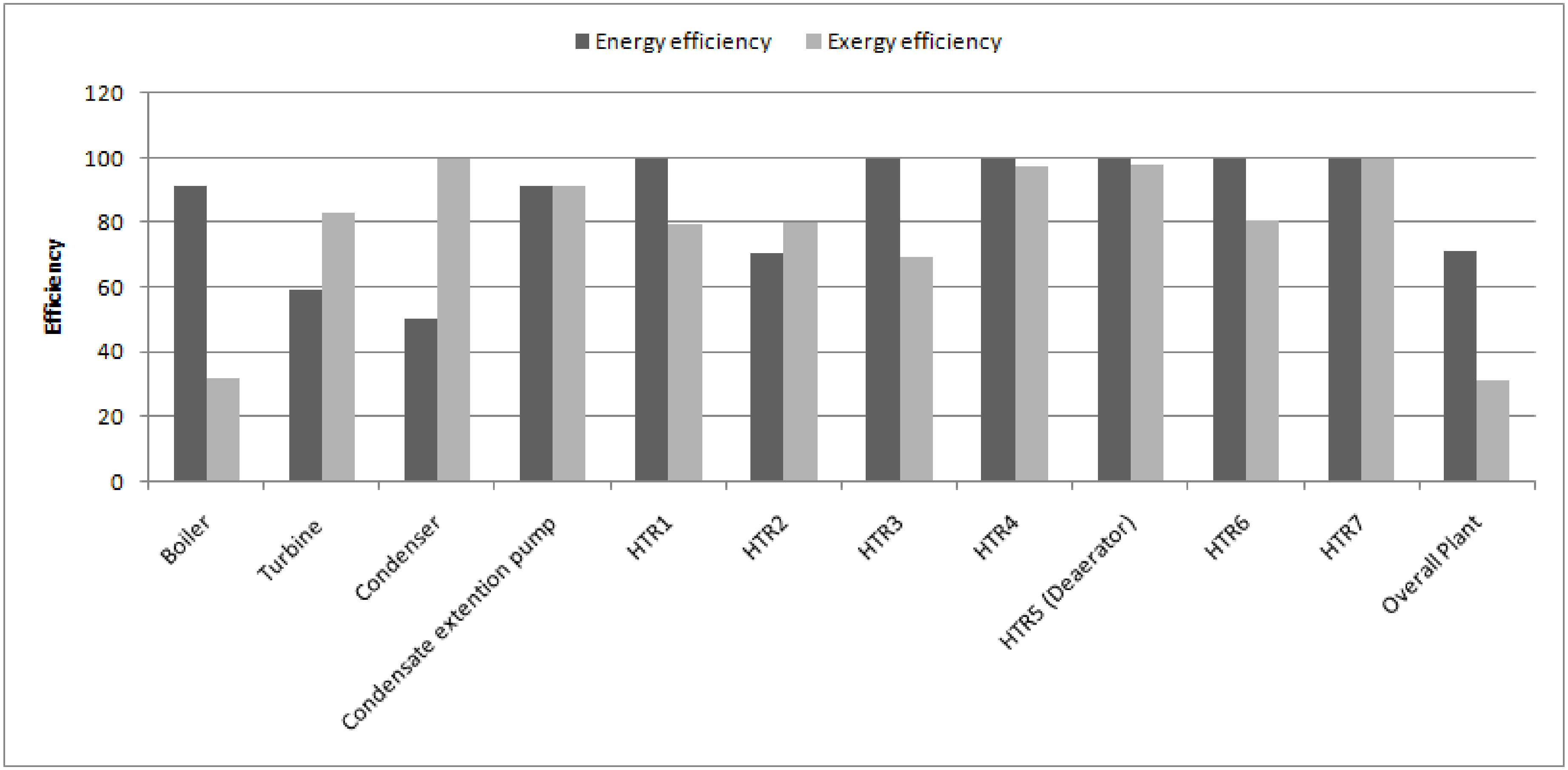

Figure: 5.7 Comparison of efficiency in the plant and components at 100% load

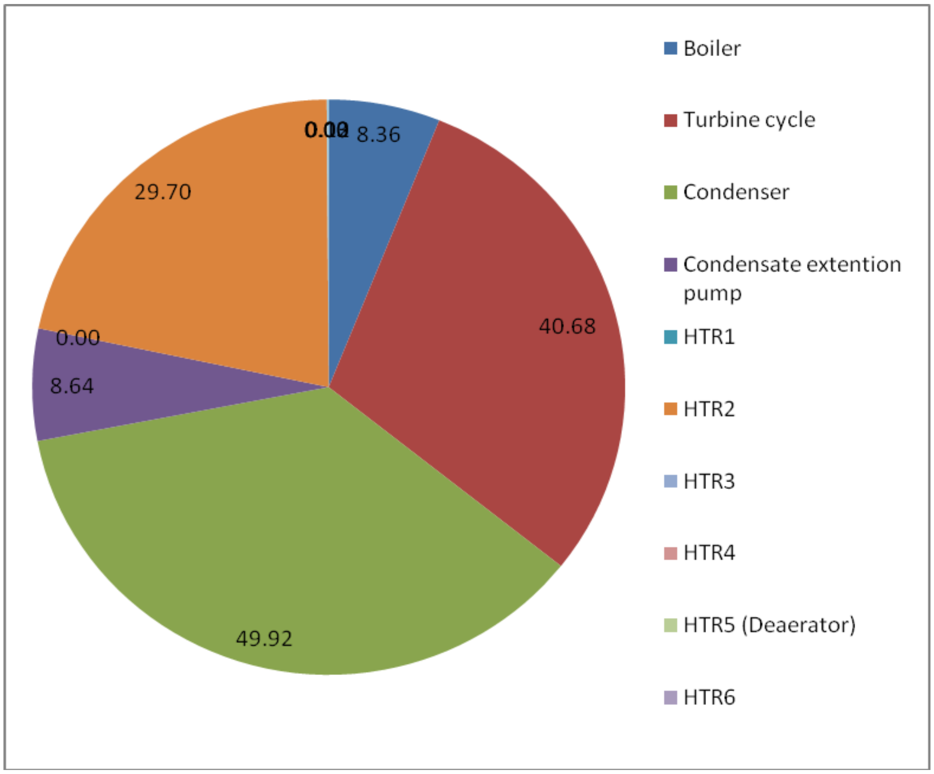

Figure: 5.8 Energy loss of different components of the plant at 100% loading condition

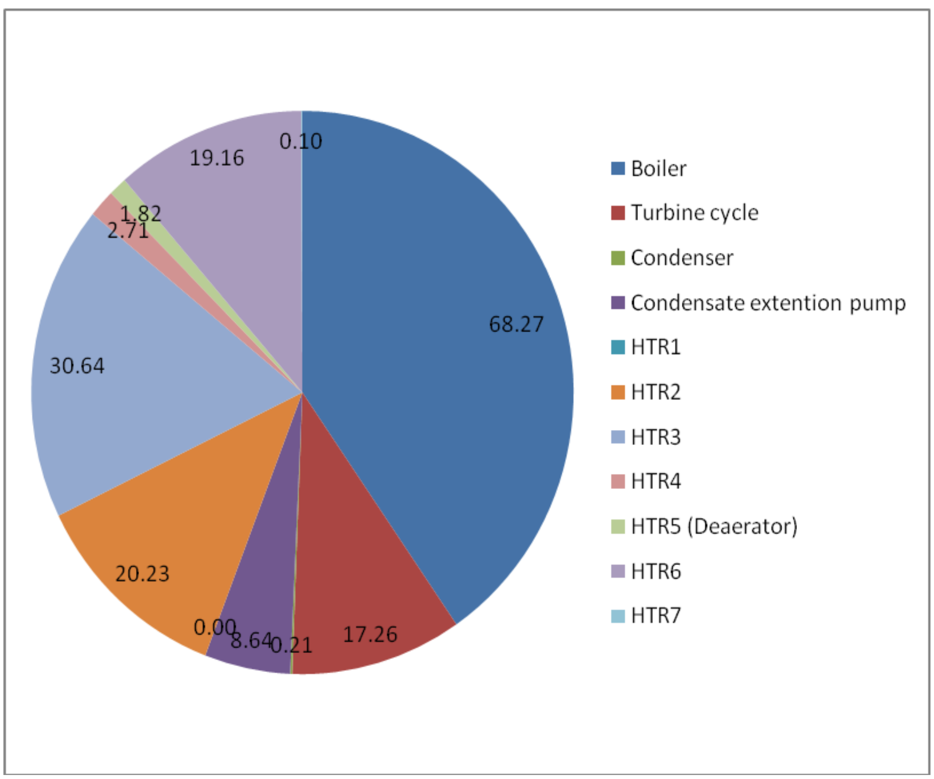

Figure: 5.9 Exergy loss of different components of the plant at 100% loading condition

5.3.4 ENERGY AND EXERGY EFFICIENCY OF HEATER

Extraction steam to LP heaters (HTR1 to HTR4) are at low pressure and temperature and have low exergy. Deaerator has high exergy efficiency, since exergy flow to it is high in comparison to all other heaters, and large surface area provided for heat transfer and deaeration also reduces irreversibilities (exergy consumption) which are shown in the table 5.6. Higher exergy consumption rate in HTR6 is due to increased irreversibilities caused by higher temperature difference between hot 217 °C and cold streams 164.9 °C in HTR6.

Table 5.6: Exergy flows and exergy efficiencies at 100% loading condition of Heater

Components	Net exergy input rate, (MW)	Useful exergy output rate, (MW)	Exergy Cons. Rate (MW)	Exergy efficiency,%
HTR1	0.3	0.2	0.1	79.52
HTR2	2.2	1.8	0.5	79.77
HTR3	3.6	2.5	1.1	69.36
HTR4	2.3	2.3	0.1	97.29
HTR5 (Deaerator)	9.8	9.7	0.2	98.18
HTR6	10.2	8.2	2.0	80.84
HTR7	5.33	5.32	0.01	99.90

Exergy analysis can also be effectively used for taking important decisions pertaining to operation and maintenance. There are cases, when conventional performance parameters do not indicate, whether an off-design operating condition is beneficial or detrimental to the overall cycle performance. For example, traditional performance analysis of HTR6 shows that with increase in reheater attemperation spray (flow 11 as shown in Fig.5. 1), there is no significant change in terminal temperature difference (the difference of saturation temperature corresponding to bled steam pressure and heater outlet feed water temperature) and drain cooling approach (the difference of heater drip outlet temperature and inlet feed water temperature). Rather feed water temperature rise across HTR6 increases due to increased rate of extraction steam flow to HTR6 and lower rate of feed water flow through HTR6. An energy analysis would thus apparently show an improvement in HTR6 performance over its original design value. However, this is contradictory to the fact that cycle efficiency is severely affected due to reheater spray. Thus any operation decisions based on the energy analysis will be incorrect for the overall plant performance. Exergy analysis across HTR6 under this condition shows larger consumption of exergy than the design figure, which corroborates to the overall plant performance degradation. First law analysis alone also often does not reflect properly the performance deterioration level of a single component. For example, if the plant runs always with reheat spray (due to some unavoidable limitation at the boiler side), any degradation of the HTR6 performance over time (e.g., due to scale formation on the heat transfer surfaces) would be difficult to trace by energy analysis alone, since the heater would always show better-than-design performance. If an exergy analysis performed across HTR6, it immediately shows that exergy consumption in HTR6 has markedly increased over its design value, warranting remedial actions. The first level of corrective actions is taken in operation level by proper adjustment of drip level, proper venting of air. The next tier of remedial actions can be taken in maintenance level. For instance, after a number of years of service, heater performance deteriorates through film build up (scaling) on heat transfer surfaces. Re-tubing or replacement of the heater can rectify this problem. Exergy analysis can benefit by pinpointing the sources of irreversibility in different components of a power cycle. Performance tests of a cycle, if conducted on the basis of exergy, can quantify the contribution of individual equipment towards the total deviation of cycle efficiency from the design values. Increase in exergy consumption by one component can be interpreted directly as the "lost power", and hence, the "lost revenue". The cost of maintenance can be weighed against the "lost revenue", making maintenance decision easier. For example, the "lost revenue" due to an aberrant equipment can be compared against the remedial maintenance cost, and decisions can be made if the concerned equipment needs minor repairing, retrofitting or full replacement.

5.4 CASE STUDY

The total cycle of this power plant consists of six closed feed water heaters (HTR1, HTR2, HTR3, HTR4, HTR6 and HTR7) and one open feed water heater (Dearetor, HTR5). The thermal efficiency of the cycle can be increased by incorporating several feed water heaters at suitably chosen pressures. But the number of feed water heaters used is based on economic considerations, since incremental increases in thermal efficiency achieved with each additional heater must justify the added capital costs (heater, piping, pumps etc). The analyses have been performed by inactivated feed water heater one by one cumulatively. When inactive feed water heater one by one, then to maintain the steam temperature fixed at the outlet condition of the boiler, the fuel flow rate must be increased. From the energy balance equation of boiler, the required fuel flow rate is calculated. By using this fuel flow rate, energy and exergy efficiencies of the overall cycle have been calculated.

5.4.1 MODEL CASE -1

If HTR3 is inactive, then the temperature rise of 34^{o}C of feed water is not obtained. So, the temperature of feed water decreases. Finally, the temperature of feed water becomes to 174^{o}C from 208.1^{o}C at 50% loading condition which is passed through the boiler. At this condition energy efficiency and exergy efficiency of the overall cycle have been calculated which are shown in table 5.7.

5.4.2 MODEL CASE -2

If HTR3 and HTR4 both are inactive, then the temperature rise of 53^{o}C of feed water is not found. So, the temperature of feed water decreases more than the model case-1. Finally, the temperature of feed water becomes to 155^{o}C from 208.1^{o}C at 50% loading condition which is passed through the boiler. At this condition energy efficiency and exergy efficiency of the overall cycle have been calculated which are shown in table 5.7. Both the heaters, HTR3 and HTR4 are low pressure heater.

5.4.3 MODEL CASE -3

If HTR3, HTR4 and HTR6 are inactive, then the temperature rise of 97.5°C of feed water is not obtained . So, the temperature of feed water decreases more than the model case-2. Then, the temperature of feed water becomes to 110.6°C from 208.1°C at 50% loading condition which is passed through the boiler. At this condition energy efficiency and exergy efficiency of the overall cycle have been calculated which are shown in table 5.7.

5.4.4 MODEL CASE -4

If HTR3, HTR4, HTR6 and HTR7 are inactive, then the temperature rise of 120.1°C of feed water is not found . So, the temperature of feed water decreases more than the model case-3. Then, the temperature of feed water becomes to 88°C from 208.1°C at 50% loading condition which is passed through the boiler. At this condition energy efficiency and exergy efficiency of the overall cycle have been calculated which are shown in table 5.7.

Table 5.7: Energy and exergy efficiencies of the Overall power plant by inactivating heater

50 % Loading	Energy efficiency	Exergy efficiency
All heaters are active	35.48	44.25
HTR3 inactive	31.49	41.97
HTR3 ,4 inactive	29.75	40.71
HTR3,4,6 inactive	26.66	38.42
HTR3,4,6,7 inactive	25.34	37.45

From the analysis of the model case, it is found that energy efficiency is decreased by 10% and exergy efficiency is decreased by 7% from the all heaters active condition to four heaters inactive condition. So, energy efficiency is decreased more than exergy efficiency.

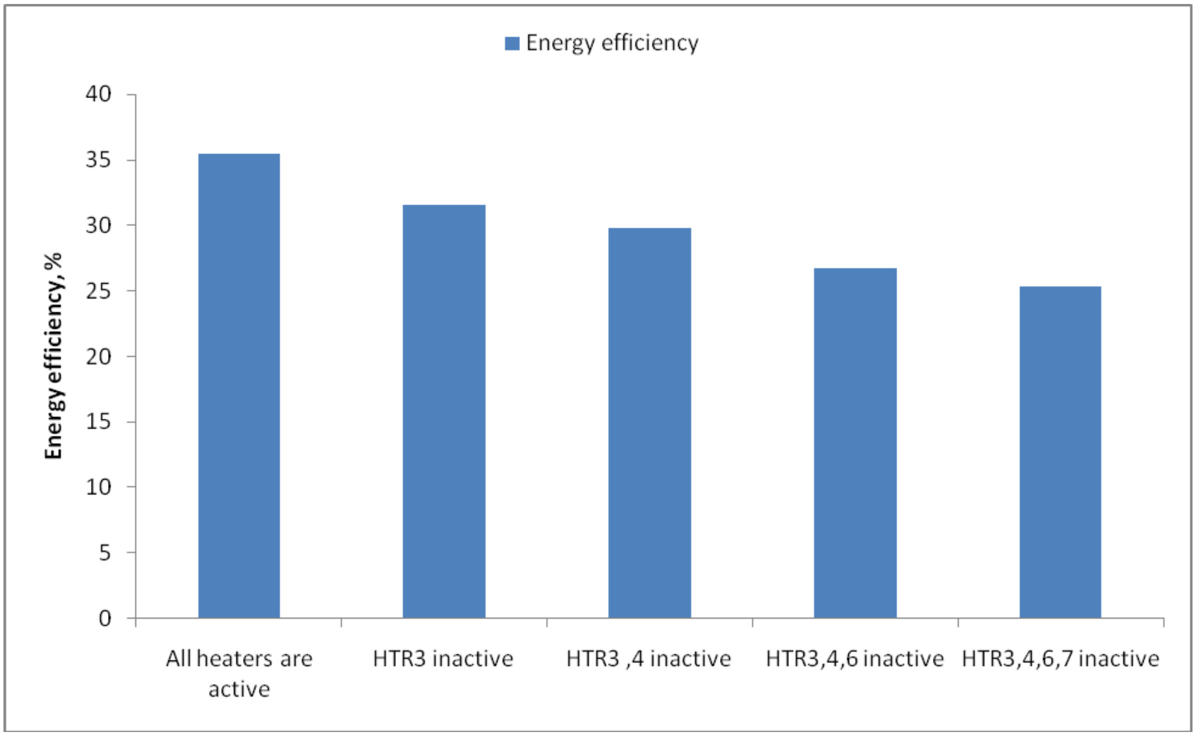

Figure: 5.10 Energy efficiency of the Overall power plant by inactivating feed water heater

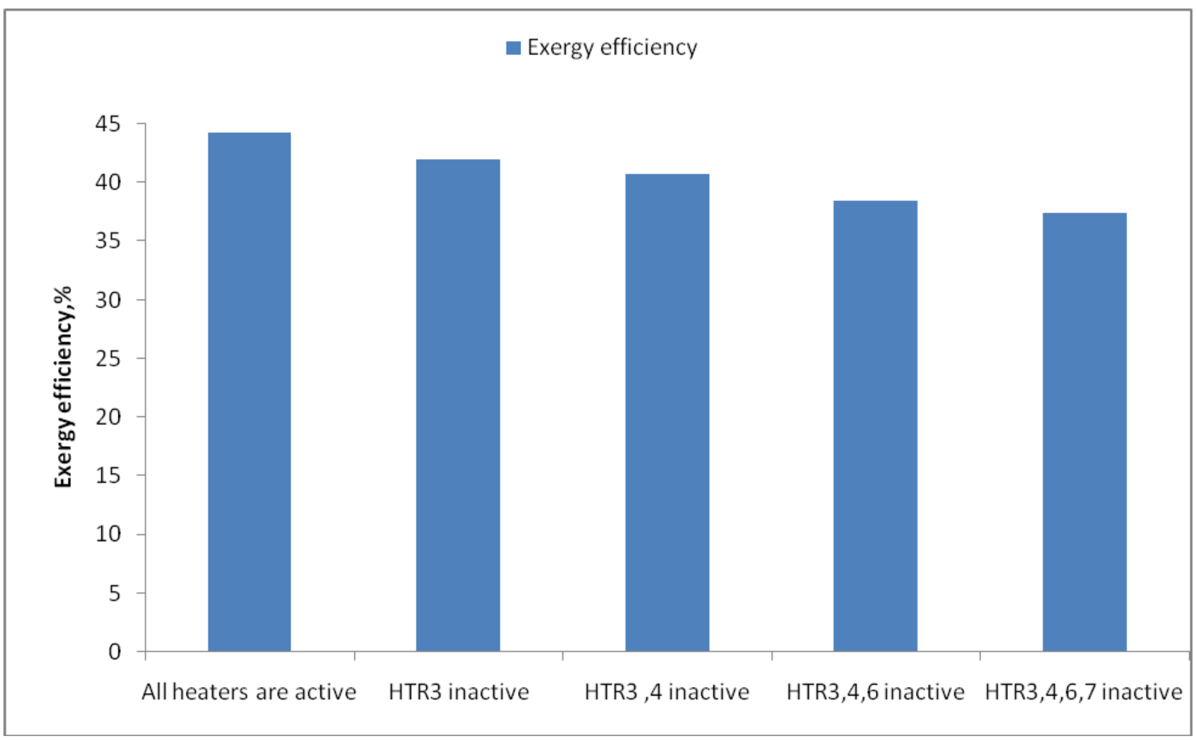

Figure: 5.11 Exergy efficiency of the Overall power plant by inactivating feed water heater

Table 5.8: Difference between Energy efficiencies by inactivating heater

50 % Loading	Energy efficiency	Difference between Energy Efficiencies
All heaters are active	35.48	
HTR3 inactive	31.49	3.99
HTR3 ,4 inactive	29.75	1.74
HTR3,4,6 inactive	26.66	3.09
HTR3,4,6,7 inactive	25.34	1.32

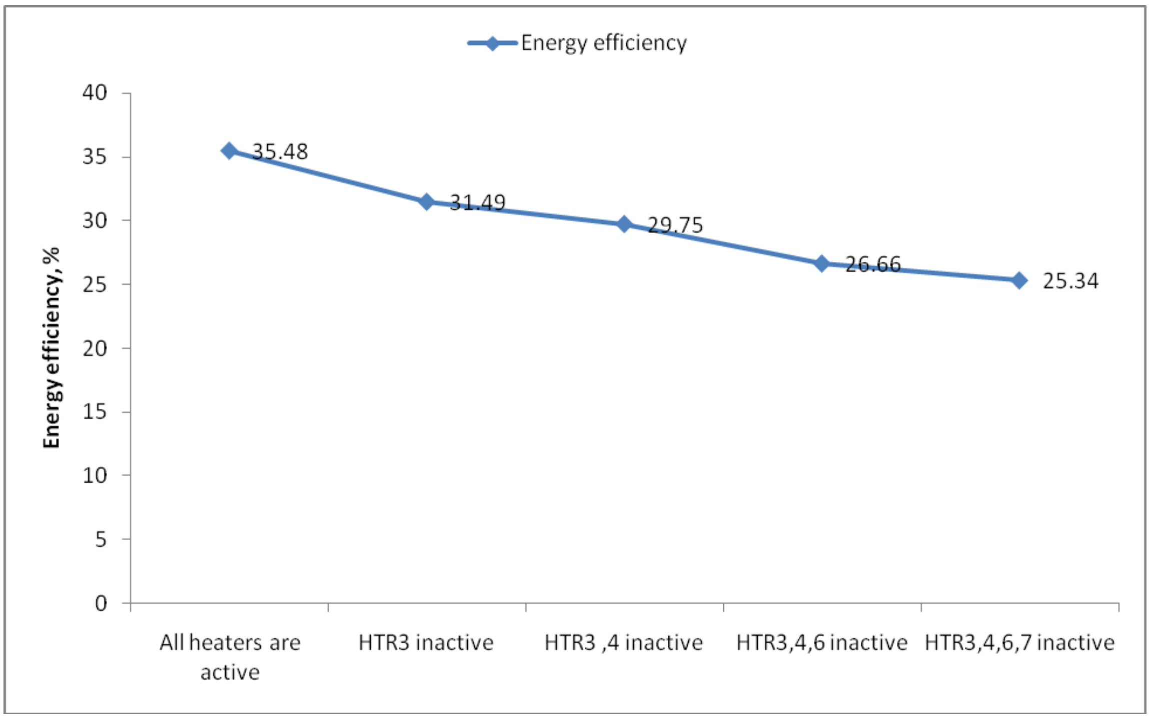

Figure: 5.12 Difference between Energy efficiencies by inactivating heater

From Table 5.8 and fig 5.11, it is found that when low pressure heater HTR3 has been inactivated Energy efficiency is decreased by 3.99%. But when both low pressure heater HTR3 and HTR4 have been inactivated Energy efficiency is decreased only by 1.74%. Later when HTR3, HTR4 and high pressure heater HTR6 are inactivated then Energy efficiency is decreased by 3.09%.But when four heaters (two low pressure heaters HTR3, HTR4 and two high pressure heaters HTR6, HTR7) are inactivated, then Energy efficiency is decreased only by 1.32%.

Table 5.9: Difference between Exergy efficiencies by inactivating heater

50 % Loading	Exergy efficiency	Difference between Exergy Efficiencies
All heaters are active	44.25	
HTR3 inactive	41.97	2.28
HTR3 ,4 inactive	40.71	1.26
HTR3,4,6 inactive	38.42	2.29
HTR3,4,6,7 inactive	37.45	0.97

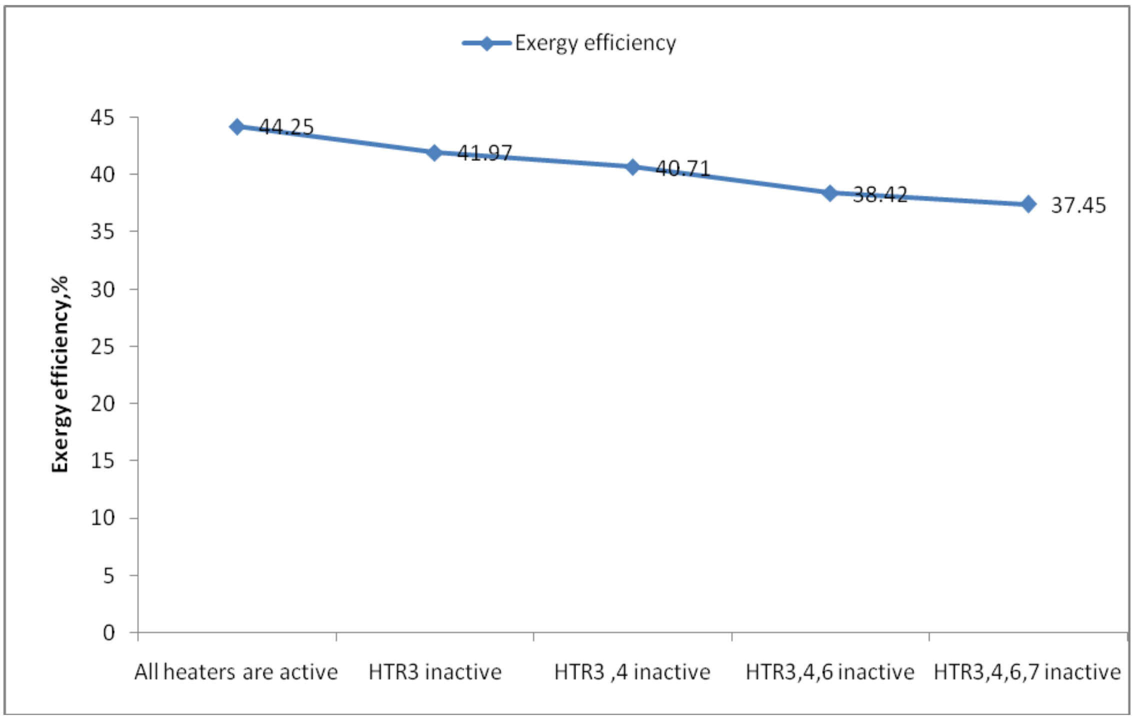

Figure: 5.13 Difference between Exergy efficiencies by inactivating heater

From Table 5.9 and fig 5.11, it is found that when low pressure heater HTR3 has been inactivated Exergy efficiency is decreased by 2.28%. But when both low pressure heater HTR3 and HTR4 have been inactivated Exergy efficiency is decreased only by 1.26%. Later when HTR3, HTR4 and high pressure heater HTR6 are inactivated then Exergy efficiency is decreased by 2.29%.But when four heaters (two low pressure heaters HTR3, HTR4 and two high pressure heaters HTR6, HTR7) are inactivated, then Exergy efficiency is decreased only by 0.97%.

CHAPTER 6

CONCLUSIONS AND RECOMMENDATIONS

6.1 CONCLUSIONS

Energy and Exergy analyses are shown in this study to be able to help understand the performance of coal fired thermal power plants and identify design possible efficiency improvements. It gives logical solution improving the power production opportunities in thermal power plants. From the data presented and the subsequent analysis, following conclusions can be drawn:

1. From the analysis it is found that exergy efficiency is lower than energy efficiency. Boiler is the main part that contributed to lower the exergy efficiency.

2. It has been observed that 68.27% exergy loss occur in combustor (Boiler) which shows combustor is not fully adiabatic and combustion may not be complete. This large exergy loss is mainly due to the combustion reaction and to the large temperature difference during heat transfer between the combustion gas and steam. Comparing with the exergy input to the plant, this actually reduces the overall plant output . Other factors that may contribute to the high amount of exergy loss are tubes fouling, defective burners, fuel quality, inefficient soot blowers, valves steam traps and air heaters fouling. Inspections of this equipment need to be carried out during the boiler outage. This study pin points that the combustor requires necessary modification to reduce exergy destructions thereby plant performance can be improved.

3. The major energy destruction occurs in the Condenser which leads to inefficient heat transfer between hot stream (flue gas) and cold stream (water and air).It indicates heat exchanger system need to be carefully inspected.

4. Poor part-load energy efficiency is attributed to higher relative energy rejection. On the contrary, poor part-load exergy efficiency is not due to higher relative exergy rejection but caused by higher relative exergy consumption.

5. Results show that feed heaters with higher temperature difference between feed water and extraction steam consume more exergy and HP heaters deal with larger quantity of exergy. Thus, a little deviation in performance of HP heaters will have greater impact on cycle efficiency and therefore, need more attention.

6. Considering the emission of flue gas, stack height is calculated. From the site visit stack height is found to be about 100 m which meets the requirement of Environmental energy conservation rules.

7. When traditional First Law analysis does not indicate performance deterioration, exergy analysis pinpoints inefficiencies and shows avenues for improvement.

8. Thus, energy analysis results lead to erroneous conclusion that major loss is associated with the heat rejection at the condenser, while exergy analysis quantitatively demonstrates that only a very small amount of work potential is lost in the condenser (since the heat is rejected nearly at the ambient temperature).

9. Operation and maintenance decisions based on exergy analysis of the power plants proved more effective. Power station equipment involves high density of exergy transfer and therefore, it is important that exergy destruction is minimized in such devices. Exergy-based approach of performance monitoring in operating power plants helps in better management of energy resources and environment.

6.2 RECOMMENDATIONS

Our country is in great crisis of power. So improvement of the efficiency of power plant is necessary. To establishment of new power plant are costly. So proper operation and maintenance decisions have been taken for increasing the efficiency of the existing power plant. The main problem of our country is that after a period of operation most of the sensors do not work properly. For this reason sometimes the correct information of the plant can not be found. However further studies are required to improve the performance of our plant. The recommendations for future work are listed below:

- Study can be performed by the effect of ambient temperature on the performance of the plant.

- For calculation of energy and exergy efficiency EXCEL spreadsheet are used. In this calculation different types of simulation software (such as ASPEN Plus) can be used.

- The analysis can also be performed by changing governor settings of the turbine control valves (i.e. at constant pressure operation and sliding pressure operation).

- Further analysis can be done with and without feed water heaters and increasing the number of feed water heater to evaluate the influence of feed water heater efficiency to the total performance of the power plant.

REVIEWER
APPROVER
RATIFIER

Checked and approved with regard to the system requirements and/or main dimensions and/or the overall arrangement. The Contractor is responsible for design, calculation, detail dimensions and operability, as well as compliance with the conditions of the Contract.

FICHTNER

Date: 2/12/02

Signature:

STC

Barapukuria 2x125 MW (Rev.0)

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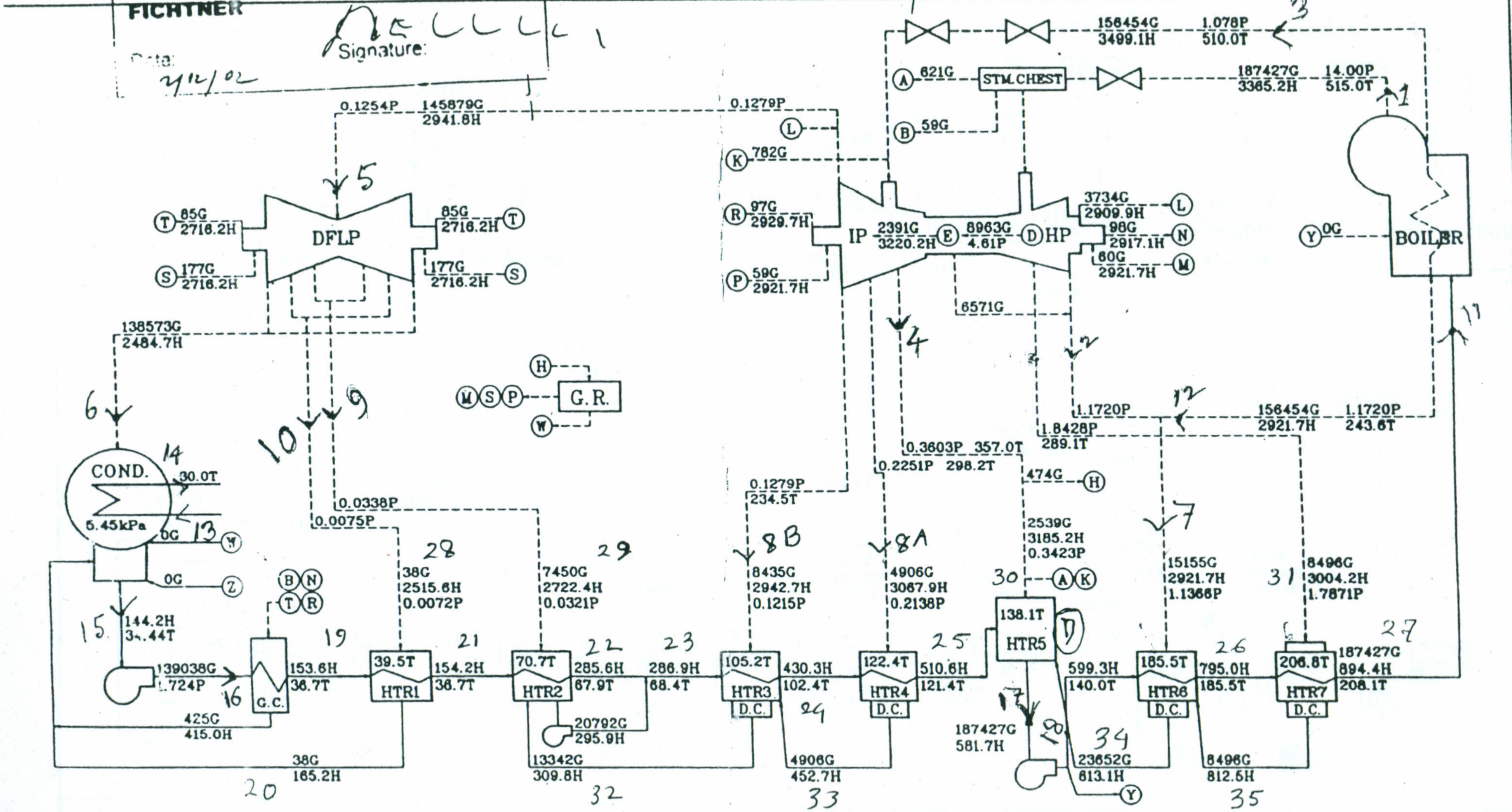

HEAT RATE 8853.1 kJ/(kW.h) (2114.5 kcal/(kW.h))	G kg/h	No : stc20012156	14.00 MPa(a) 515.0 °C/510.0 °C 5.45 kPa(a)
1. CALCULATIONS ARE BASED ON NO RADIATION LOSSES TO HEATERS OR EXTRACTION PIPING LOCATED IN THE CONDENSER NECK.	P MPa	PROJECT : BENGAL	0.0% MAKE UP
2. HEAT RATES ARE BASED ON LOCUS OF FULL VALVE POINTS.	T °C	DESIGNER : YY	62512kW
	H kJ/kg	DATE : 2001-11-9	50%TMCR
			SHANGHAI TURBINE CO. LTD.

REVIEWER		01. 11
APPROVER		01. 11
RATIFIER		

STC Barapukuria 2X 125MW (Rev. 0)

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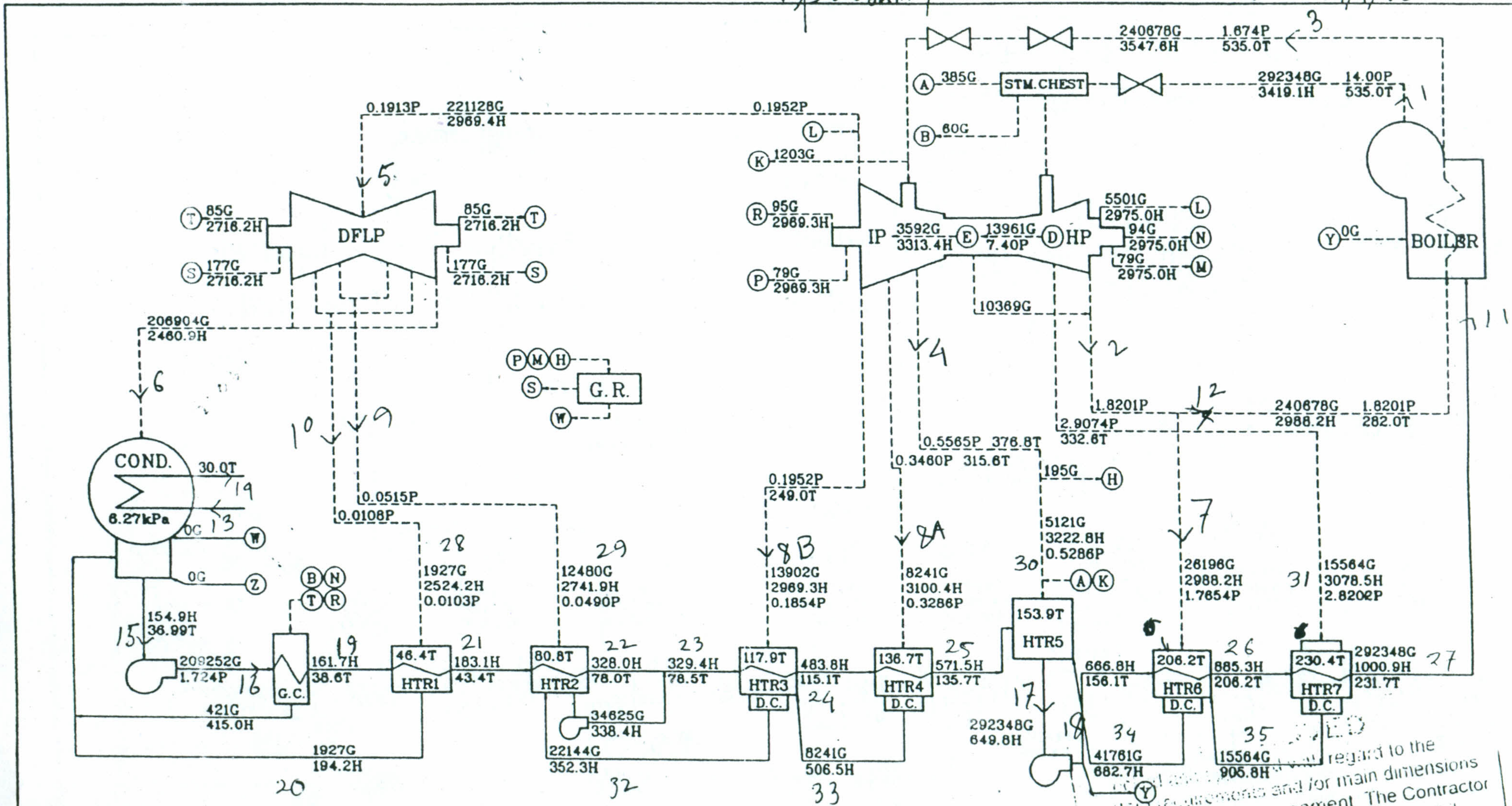

HEAT RATE 8411.5 kJ/(kW.h) (2009.1 kcal/(kW.h))	G kg/h	No : stc20012301	14.00 MPa(a) 535.0°C/535.0°C 6.27 kPa(a)
1. CALCULATIONS ARE BASED ON NO RADIATION LOSSES TO HEATERS OR EXTRACTION PIPING LOCATED IN THE CONDENSER NECK.	P MPa	PROJECT : BENGAL	0.0% MAKE UP
2. HEAT RATES ARE BASED ON LOCUS OF FULL VALVE POINTS.	T °C	DESIGNER : YY	100054kW
	H kJ/kg	DATE : 2001-11-9	80%TMCR
			SHANGHAI TURBINE CO., LTD.

... with regard to the ... and /or main dimensions ... requirements ... overall arrangement. The Contractor ... responsible for ... calculation, detail dimensions and operability, as well as compliance with the conditions of the Contract.

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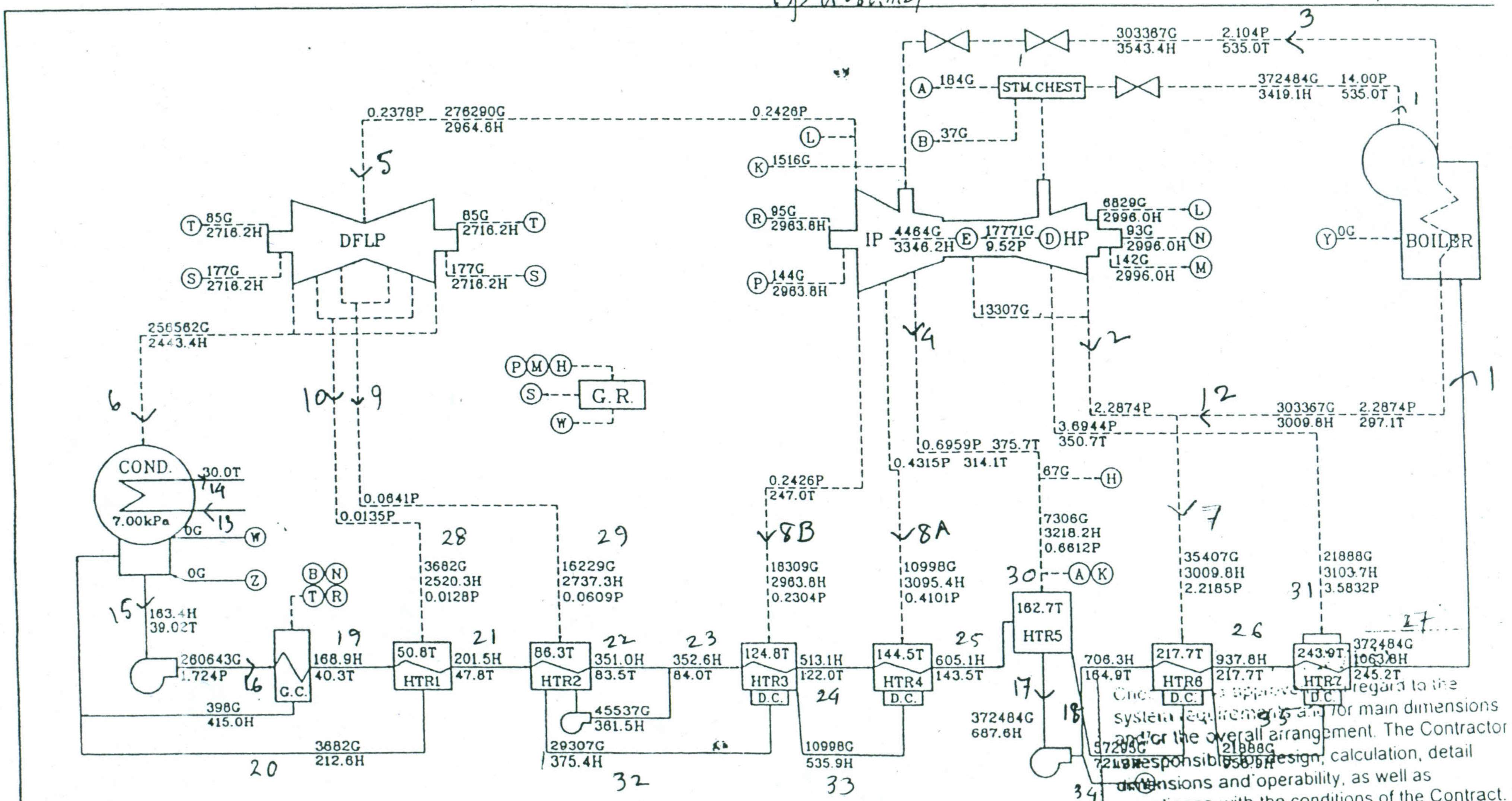

Checked with regard to the system requirements and/or main dimensions and/or the overall arrangement. The Contractor is responsible for design, calculation, detail dimensions and operability, as well as compliance with the conditions of the Contract.

FICHTNER

HEAT RATE 8308.7 kJ/(kW.h) (1984.5 kcal/(kW.h))	G kg/h	No : stc20012151	14.00 MPa(a) 535.0 °C/535.0 °C 7.00 kPa(a)
1. CALCULATIONS ARE BASED ON NO RADIATION LOSSES TO HEATERS OR EXTRACTION PIPING LOCATED IN THE CONDENSER NECK.	P MPa	PROJECT : BENGAL	0.0% MAKE UP
2. HEAT RATES ARE BASED ON LOCUS OF FULL VALVE POINTS.	T °C	DESIGNER : YY	125073kW
	H kJ/kg	DATE : 2001-11-9	TMCR
			SHANGHAI TURBINE CO., LTD.

7/11/02

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APPENDIX-A

SAMPLE CALCULATION

A1. Calculation of Overall energy efficiency (For 50% loading condition)

From the collected data

Mass flow rate of coal, m_{in}^{f}=6.94 kg/sec
Mass flow rate of air, m_{in}^{a}=65 kg/sec

The considered coal for this paper represents the typical coal of Bangladesh (Baropukuria coal mine) with lower heating value 22.7 MJ/Kg .

Therefore, the rate of energy entering the control volume with fuel:

$En_{in}=En_{in}^{f}+En_{13}$ =1.58E+05 kW

The following composition (by mass) of coal has been considered based on the design:

83% C, 5.1% H_2, 9.4% o2, 1.7% N_2, Ash 12.4%, H_2O=10%

$C+0.37H_2+1.18\ O_2=CO_2+0.37H_2O$

Neglecting the ash in flue gas, the mass flow rate of flue gas can be evaluated as

$m_g=(0.876m_{in}^{f}+m_{in}^{a})$=71.07 kg/sec

m_{in}^{a} is obtained from designed value of the plant corresponding to different boiler loading.

Mass flow rate of moisture in flue gas:

m_g^{H2O}=moisture from coal+moisture from combustion of hydrogen in coal+moisture from air = (moisture % in coal)x m_{in}^{f} +(%H_2 in coal x m_{in}^{f})+moisture from air

The mass flow rate of moisture from air is obtained from the psychrometric chart considering the air temperature and the relative humidity.

From the psychrometric chart, Humidity ratio=Kg water/kg dry air=0.016

Moisture from air=Humidity ratio x kg dry air

m_g^{H2O}= .1x6.94+(.051x6.94)+1.04=4.92kg/sec

The mass flow rate of dry flue gas:

$m_g^{dry}=m_g-m_g^{H2O}$= 66.16 kg/sec

The oxygen and carbon dioxide percentages at the air-heater inlet are used for calculating the mass flow rate of oxygen and CO_2 in the flue gas as

$m_g^{O2}=n_g^{O2}$x m_g^{dry} =13.86

$m_g^{CO2}=n_g^{CO2}$x m_g^{dry}=0.02

$m_g^{N2}=n_g^{N2}$x m_g^{dry}=51.7

From experimental analysis $n_g^{O}{}_2$=20.95%,n_g^{CO2}=0.03%,n_g^{N2}=78.09%,n_g^{H2}=.00005%

The energy flow rate leaving the control volume with flue gas is

$E^g_{out}=\Sigma m_j^g e_j^g$= -1.59E+05 kW

The total exergy flow rate leaving the control volume including that with circulating water is

$E_{out}=E^g_{out}+E_{14}$=-1.59E+05 kW

Flue gas Temp,T=409 k, ambient Temp=300 k

	Cp at 409 K	m	• T=T-To	ho	h=mcp• T	En=m(h-ho)
CO_2	0.9	0.02	109	2.2	1.8	-0.3
O_2	0.9	13.9	109	2083.6	1394.4	-689.1
N_2	1.0	51.7	109	8911.8	5856.2	-157853.0

The net power output from the control volume is= P_{net}=G-P_{aux}=5.63E+05 kW		
The energy efficiency of the control volume 3 is		
	η= $P_{net}/(E_{in}-E_{out})$	

	=35.48%	

A-2 Exergy efficiency calculation of the Total Power Cycle (Exergy Analysis of the control volume 3) (For 50% loading condition)

The exergy of coal comprises of both chemical and thermo-mechanical components and is defined on the basis of a single step irreversible reaction in consideration of the ultimate analysis of coal. The following composition (by mass) of coal has been considered based on the design:

83% C, 5.1% H_2, 9.4% O_2, 1.7% N_2, Ash 12.4%, H_2O 10%

The Stoichiometric equation representing the complete combustion of the coal in oxygen and considering that the hydrogen in coal (as H2) is fully oxidized during combustion is

$C + 0.37H_2 + 1.18\ O_2 = CO_2 + 0.37H_2O$

The fuel is assumed to enter the system at room temperature. Therefore, the specific exergy of coal per mole of carbon (neglecting thermo-mechanical component) can be written as [17]

$$e_{in}^{-f} = \left[\bar{g}C + 0.37\bar{g}H_2 + 1.18\bar{g}O_2 - \bar{g}CO_2 - 0.37H_2 0\right](T_o, P_o)$$
$$+ \bar{R}T_o \ln\left[\frac{\left(y_{O_2}^e\right)^{1.18} x \left(y_{H_2}^e\right)^{0.37}}{\left(y_{CO_2}^e\right)\ x\left(y_{H_2 0(g)}^e\right)^{0.37}}\right]$$

The variables with bar (e.g. $\bar{e}$) designate the molar values of the respective quantities. For the specific molar entropy calculations in the above equation, the reference mole fractions of O_2, CO_2 and H_2O are taken from [17]. From the above equation, it is found

$e_{in}^{-f} = 4.96 x 10^5$ kJ/kmol

n_c=	0.83
M_c=	12
m_{in}^{f} =	6.94
m_{in}^{a} =	65

Therefore, the rate of exergy entering the control volume with fuel:

$$Ex_{in}^{f} = m_{in}^{f} n_c \frac{e_{in}^{-f}}{M_c}$$

where m_{in}^{f} is the mass flow rate of coal at the inlet, n_C is the mass fraction of carbon in coal and M_C is the molecular weight of carbon.

Exergy entering the control volume with air at reference temperature is zero. Considering the exergy input with fuel and circulating water, the total exergy flow rate entering the control volume is

$$Ex_{in}=Ex_{in}^{f}+Ex_{13}$$

$$Ex_{in}^{f}=2.38E+05$$

$$Ex_{in}=Ex_{in}^{f}+Ex_{13}=2.38E+05 \text{ kW}$$

Neglecting the ash in flue gas, the mass flow rate of flue gas can be evaluated as

$m_g=(0.876m_{in}^{f}+m_{in}^{a})=71.07$ kg/sec

m_{in}^{a} is obtained from designed value of the plant corresponding to different boiler loading.

Mass flow rate of moisture in flue gas:

m_g^{H2O}=moisture from coal+moisture from combustion of hydrogen in coal+moisture from air = (moisture % in coal)x m_{in}^{f} +(%H_2 in coal x m_{in}^{f})+moisture from air

The mass flow rate of moisture from air is obtained from the psychrometric chart considering the air temperature and the relative humidity.

From the psychrometric chart, Humidity ratio=Kg water/kg dry air=0.016

Moisture from air=Humidity ratio x kg dry air
m_g^{H2O}= .1x6.94+(.051x6.94)+1.04
=4.92 kg/sec
The mass flow rate of dry flue gas:

$m_g^{dry}=m_g-m_g^{H2O}$= 66.16 kg/sec

The oxygen and carbon dioxide percentages at the air-heater inlet are used for calculating the mass flow rate of oxygen and CO_2 in the flue gas as

$m_g^{O2}=n_g^{O2}x\ m_g^{dry}$ =13.86

$m_g^{CO2}=n_g^{CO2}x\ m_g^{dry}=0.02$

$m_g^{N2}=n_g^{N2}x\ m_g^{dry}=51.7$

From experimental analysis $n_g^{O2}=20.95\%, n_g^{CO2}=0.03\%, n_g^{N2}=78.09\%, n_g^{H2}=.00005\%$

$$Ex_{out}^{g} = \sum m_j^g e_j^g$$

Where, the summation applies over all the components of the flue gas. The total exergy flow rate leaving the control volume including that with circulating water

$$Ex_{out}=Ex^{g}_{out}+Ex_{14}$$

The total exergy flow rate leaving the control volume including that with circulating water is

$Ex_{out}=Ex^{g}_{out}+Ex_{14}$= 1.11E+05 kW

Taking, T=409, T=300 Calculating the specific exergy the components of the flue gas.

	Cp at 409K	m	• T=T-To	ho	h=mcp• T	So	$S=mc_p ln(T/To)$	$e=(h-h_o)-T_o(S-S_o)$
CO_2	0.87	0.02	109	3.03	1.8	0.08	0.005	23.4
O_2	0.92	13.86	109	2083.5	1386.9	81.3	3.9	22521.9
N_2	1.04	51.66	109	8911.8	5856.2	321.6	16.6	88415.7

The net power output from the control volume is

Pnet=G-Paux

=5.63E+04 kW

The exergy efficiency of the control volume 3 is evaluated accordingly using the following Equation

$$\eta_2 = \frac{P_{net}}{Ex_{in} - Ex_{out}}$$

=44.25%

A-3 Energy efficiency calculation of the Components of the Equipment (Energy Analysis of the control volume 3) (For 50% loading condition)

Energy efficiency of Boiler:

The First Law efficiency is defined as

Energy efficiency of Boiler=(m3(h3-h12)+m1(h1-h11))/Q_k

$Q_{k=}$ 1.58E+05 kW

ηBoiler= 97.30%

Energy Analysis of the control volume 2 (Turbine cycle)

The energy flow rate entering the control volume

$$E_{in} = m_1 h_1 + m_3 h_3 + m_{13} h_{13}$$

The energy flow rate leaving the control volume

$$E_{in} = m_1 h_1 + m_3 h_3 + m_{13} h_{13}$$

E_{14}-E_{13}= Qk=Qrej=$m_c C_p$• T= 40638 kW

E_{in}-E_{out}= E_1+E_3+E_{13}-E_{11}-E_{12}-E_{14}=1.13E+05 kW

$P_{in}^{(CEP)}$= $m_{16}h_{16}$-$m_{15}h_{15}$= 6.37E+01 kW

$P_{in}^{(BFP)}$= $m_{18}h_{18}$-$m_{17}h_{17}$= 9.16E+02 kW

The net power output from the control volume is

$P_{net} = G - P_{in}^{(CEP)} - P_{in}^{(BFP)}$
=6.15E+04 kW

ηTurbine=54.4%

Energy efficiency of condenser

The energy balance for the condenser is given by:

0 = m6*(h6 − h15) − Qrej − Energy loss $Qrej = m_c C_p \bullet T = 40697$ kW

T_{out}=30 K

This gives: T_{in}=25 K

Q loss = m6*(h6 − h15) – Qrej= 4.94E+04 kW

The First Law efficiency is:

η Condenser = 1 − (Q loss/m6(h6-h15)= 45.17%

Energy efficiency of Heater

Efficiency of HTR1=(m19*(h21-h19))/(m20*(h28-h20))= 93%

Efficiency of HTR2=(m21*(h22-h21))/(m29*h29+m32*h32))= 75%

Efficiency of HTR3=(m23*(h24-h23))/(m8B*h8B+m33*h33-m32*h32)= 100%

Efficiency of HTR4=(m24*(h25-h24))/(m8A*h8A-m33*h33)= 100%

Efficiency of HTR5(Deaerator)=(m17*h17)/(m25*h25+m30*h30+m34*h34)= 100%

Efficiency of HTR6=(m18*(h26-h18))/(m7*h7-m34*h34+m35*h35)= 100%

Efficiency of HTR7=(m26*(h27-h26))/(-m35*h35+m31*h31)= 100%

Energy efficiency of Condensate Pump

P_{15}=.00554 Mpa P_{16}=1.724 Mpa
$Wp=h_{16}-h_{15}$

$s_{16}=s_{15}$
$h_{16}-h_{15}=v(P_{16}-P_{15})$

$Wp=v(P_{16}-P_{15})$=66.37kW

Efficiency=$m_{15}(h_{16}-h_{15})/Wp$=96%

A-4 Exergy efficiency calculation of the Components of the Equipment (Energy Analysis of the control volume 3) (For 50% loading condition)

Exergy efficiency of Boiler:

The Second law efficiency is defined as

Exergy efficiency of Boiler=(m3(e3-e12)+m1(e1-e11))/Q_k

$Q_{k=}$	2.37E+05	kW
η Boiler=	32.9%	

Exergy Analysis of the control volume 2 (Turbine cycle)

The exergy flow rate entering the control volume

$Ex_{in}= Ex_1+Ex_3+Ex_{13}$

The exergy flow rate leaving the control volume

$Ex_{out}= Ex_{11}+Ex_{12}+Ex_{14}$

$Ex_{14}-Ex_{13}= Ex_6-Ex_{15}$

Ex_6-Ex_{15}= 2.88E+03 kW

Ex_{in}-Ex_{out}= Ex1+Ex3+Ex13-Ex11-Ex12-Ex14=7.56E+04kW

$P_{in}^{(CEP)}$= $m_{16}h_{16}$-$m_{15}h_{15}$= 6.37E+01 kW
$P_{in}^{(BFP)}$= $m_{18}h_{18}$-$m_{17}h_{17}$= 9.16E+02 kW

The net power output from the control volume is

P_{net} = G - $P_{in}^{(CEP)}$ - $P_{in}^{(BFP)}$=

• ,turbine=81.3%

<u>Exergy efficiency of condenser</u>

The exergy balance for the condenser is given by: To=298 K
Tk=303 K

0 =$m_6(e_6$-$e_{15})$-• $(1$-$T_0/T_k)Q_k$-T_oS_{gen} Qk=Qrej=m_cC_p• T=40640 kW

The irreversibility=exergy loss is:

$I_{destroyed}$=ToSgen=(m6(h6-h15)-To(m6(s6-s15)-• (1-T0/Tk)Qk)= 2.21E+03 kW

The exergy efficiency is:

1-$I_{destroyed}/m_6(e_6$-$e_{15})$= 99.07%

<u>Exergy efficiency of Heater</u>

Efficiency of HTR1=(m19*(e21-e19))/(m20*(e28-e20))= 50.5%

Efficiency of HTR2=(m21*(e22-e21))/(m29*e29+m32*e32))= 50.5%

Efficiency of HTR3=(m23*(e24-e23))/(m8B*e8B+m33*e33-m32*e32)= 78.5%

Efficiency of HTR4=(m24*(e25-e24))/(m8A*e8A-m33*e33)= 34.9%

Efficiency of HTR5(Deaerator)=(m17*e17)/(m25*e25+m34*e34+m30*e30)= 91.3%

Efficiency of HTR6=(m18*(e26-e18))/(m7*e7-m34*e34+m35*e35)= 92.75%

Efficiency of HTR7=(m26*(e27-e26))/(-m35*e35+m31*e31)= 87.8%

Exergy efficiency of Pump

$Wp=h_{16}-h_{15}$

$s_{16}=s_{15}$

$h_{16}-h_{15}=v(P16-P15)$

$Wp=v(P_{16}-p_{15})$= 66.36 kW

Efficiency=$m_{16}(e_{16}-e_{15})/Wp$=96.02%

Table B1: Design data on pressure, temperature and mass flow rate at different points

SL No	50%			80%			100%		
j	Pr (Mpa)	Temp(°C)	m (kg/h)	Pr (Mpa)	Temp(°C)	m (kg/h)	Pr (Mpa)	Temp(°C)	m (kg/h)
1	14	515	187427	14	535	292348	14	535	372484
2	1.172	243.6	180583	1.8201	282	282637	2.2874		360735
3	1.078	510	156454	1.674	535	240678	2.104	535	303367
4	0.3603	357	3013	0.5565	376.8	5316	0.6959	375.7	7373
5	0.1254		145879	0.1913		221128	0.2378		276290
6	0.00554	X=.97	138573	0.00627		206904			256562
7	1.1366		15155	1.7654		26196	2.2185		35407
8A	0.2251	298.2	4906	0.3286		8241	0.4101		10998
8B	0.1279	234.5	8435	0.1854		13902	0.2426	247	18309
9	0.0321	119.46	7450	0.049		12480	0.0609		16229
10	0.0072	X=.98	38	0.0103		1927	0.0128		3682
11	15.7	208.1	187427		231.7	292348		245.2	372484
12	1.172	243.6	156454	1.8201	282	240678	2.2874	297.1	303367
13			7000200			11197440			13996800
14			7000200			11197440			13996800
15	0.00554	34.44	139038		36.99	209252		39.02	260643
16	1.724	34.44	139038	1.724	36.99	209252	1.724	39.02	260643
17		138.1	187427		153.9	292348		162.7	372484
18		140	187427		156.1	292348		164.9	372484
19		36.7	139038		38.6	209252		40.3	260643
20			38			1927			3682
21		36.7	139038		43.4	209252		47.8	260643
22		67.9	139038		78	209252		83.5	260643
23		68.4	159830		78.5	243877		84	306180
24		102.4	159830		115.1	243877		122	306180
25		121.4	159830		135.7	243877		143.5	306180
26		185.5	187427		206.2	292348		217.7	372484
27		208.1	187427		231.7	292348		245.2	372484
28	0.0072		38	0.0103		1927	0.0128		3682
29	0.0321		7450	0.049		12480	0.0609		16229
30	0.3423		3942	0.5286		5121	0.6612		7306
31	1.7671		8496	2.8202		15564	3.5832		21888
32			13342			22144			29307
33			4906			8241			10998
34			23652			41761			57295
35			8496			15564			21888
Generation (MW)		62.5		100			125		
Coal flow (T/h)		25		40			50		
Total air flow (Kg/sec		65		104			130		
Auxiliary power (%)		10		16			20		

N.B Data is collected by BPDB Officials

Table B2: Operating data on pressure, temperature and mass flow rate at different points

SL No.	67% (Dated on 14.04.2013)			57% (Dated on 13.04.2013)		
j	Pr (Mpa)	Temp(oC)	m (kg/h)	Pr (Mpa)	Temp(oC)	m (kg/h)
1	11.1	533	284000	8.69	522	232000
2	1.75	313		1.5	529	
3	1.65	525		1.43	520	230000
4	0.51	362		0.37	377	
5					131	
6	1.76	311				220000
7	1.54	240		1.54	316	
8A	0.24	300		0.24	298	
8B	0.09	225		0.04	225	
9				0.02	131	
10				0.01	60	
11		226	274000	9.8	218	202000
12	1.75	313		1.54		
13	0.19	32.3		0.2	32.7	
14	0.1	37.1		0.1	37.8	
15	0.086	51		0.085	134	220000
16	1.91	55.9	274000	1.8	57	220000
17	0.43	152	257000	0.34		202000
18	12.4	154	257000	9.85	148	202000
19	inactive			inactive		
20						
21					63.5	220000
22					78.9	220000
23						220000
24					112	220000
25					131	220000
26						202000
27				9.8	218	202000
28						
29					131	
30				0.37	361	
31				2.2	351	
32						
33						
Generation (MW)		83.9		71.3		
Coal flow (T/h)		40		35		
Total air flow (Nm^3/hr)		321218		290120		
Auxiliary power (MW)		10		9.18		

Table C1: Calculated Flow data corresponding to 50% load

Flow, j	To	mj,kg/h	Mass flow rate, mj,kg/s	ho,kj/kg	Specific enthalpy,hj, kj/kg	en,kj/kg	Energy flow rate En,kj/s	Specific entropySj,KJ/Kg K	So,KJ/KgK	Specific Exergy,e,kj/ kg	Exergy flow rate,Ex,kj/s
						en=(hj-ho)	En=mj*en			ej=(hj-ho)-To(sj-	Ex=ej*mj
1	298	187427	52.06	104.6	3365.2	3260.6	169756.799	6.447	0.367	1448.76	7.54E+04
2	298	180583	50.16	104.6	2921.7	2817.1	141311.214	6.82	0.367	894.106	4.49E+04
3	298	156454	43.46	104.6	3499.1	3394.5	147523.084	7.756	0.367	1192.578	5.18E+04
4	298	3013	0.84	104.6	3185.24	3080.64	2578.32453	7.812	0.367	862.03	7.21E+02
5	298	145879	40.52	104.6	2941.8	2837.2	114968.861	7.867	0.367	602.2	2.44E+04
6	298	138573	38.49	104.6	2484.7	2380.1	91615.9993	8.1	0.367	75.666	2.91E+03
7	298	15155	4.21	104.6	2921.7	2817.1	11859.2085	6.98	0.367	846.426	3.56E+03
8A	298	4906	1.36	104.6	3067.9	2963.3	4038.31939	7.83	0.367	739.326	1.01E+03
8B	298	8435	2.34	104.6	2942.7	2838.1	6649.82597	7.84	0.367	611.146	1.43E+03
9	298	7450	2.07	104.6	2722.4	2617.8	5417.39167	6.99	0.367	644.146	1.33E+03
10	298	38	0.01	104.6	2515.6	2411	25.4494444	8.19	0.367	79.746	8.42E-01
11	298	187427	52.06	104.6	894.4	789.8	41119.4013	2.38	0.367	189.926	9.89E+03
12	298	156454	43.46	104.6	2921.7	2817.1	122429.601	6.82	0.367	894.106	3.89E+04
13	298	7000200	1944.50	104.6		-104.6	-203394.7		0.367	4.766	9.27E+03
14	298	7000200	1944.50	104.6		-104.6	-203394.7		0.367	4.766	9.27E+03
15	298	139038	38.62	104.6	144.2	39.6	1529.418	0.497	0.367	0.86	3.32E+01
16	298	139038	38.62	104.6	145.85	41.25	1593.14375	0.497	0.367	2.51	9.69E+01
17	298	187427	52.06	104.6	581.7	477.1	24839.2838	1.76	0.367	61.986	3.23E+03
18	298	187427	52.06	104.6	599.3	494.7	25755.5936	1.76	0.367	79.586	4.14E+03
19	298	139038	38.62	104.6	153.6	49	1892.46167	0.52	0.367	3.406	1.32E+02
20	298	38	0.01	104.6	165.2	60.6	0.63966667	0.58	0.367	-2.874	-3.03E-02
21	298	139038	38.62	104.6	154.2	49.6	1915.63467	0.49	0.367	12.946	5.00E+02
22	298	139038	38.62	104.6	285.6	181	6990.52167	0.94	0.367	10.246	3.96E+02
23	298	159830	44.40	104.6	286.9	182.3	8093.61361	0.98	0.367	-0.374	-1.66E+01
24	298	159830	44.40	104.6	430.3	325.7	14460.1753	1.36	0.367	29.786	1.32E+03
25	298	159830	44.40	104.6	510.6	406	18025.2722	1.6	0.367	38.566	1.71E+03
26	298	187427	52.06	104.6	795	690.4	35944.3336	2.2	0.367	144.166	7.51E+03
27	298	187427	52.06	104.6	894.4	789.8	41119.4013	2.4	0.367	183.966	9.58E+03
28	298	38	0.01	104.6	2515.6	2411	25.4494444	8.19	0.367	79.746	8.42E-01
29	298	7450	2.07	104.6	2722.4	2617.8	5417.39167	8.2	0.367	283.566	5.87E+02
30	298	3942	1.10	104.6	3185.2	3080.6	3373.257	6.1	0.367	1372.166	1.50E+03
31	298	8496	2.36	104.6	3004.2	2899.6	6843.056	6.2	0.367	1161.366	2.74E+03
32	298	13342	3.71	104.6	309.8	205.2	760.494	1.4	0.367	-102.634	-3.80E+02
33	298	4906	1.36	104.6	452.7	348.1	474.382944	1.8	0.367	-78.934	-1.08E+02
34	298	23652	6.57	104.6	613.1	508.5	3340.845	1.91	0.367	48.686	3.20E+02
35	298	8496	2.36	104.6	812.5	707.9	1670.644	2.2	0.367	161.666	3.82E+02

Table C2: Calculated Flow data corresponding to 80% load

Flow, j	To	mj,kg/h	Mass flow rate, mj,kg/s	ho,kj/kg	Specific enthalpy,hj,	en,kj/kg	Energy flow rate En,kj/s	Specific entropySj,KJ/	So,KJ/KgK	Specific Exergy,e,kj/k	Exergy flow rate,Ex,kj/s
						en=(hj-ho)	En=mj*en			ej=(hj-ho)-To(sj-	Ex=ej*mj
1	298	292348	81.21	104.6	3419.1	999567047	8.1173E+10	6.52	0.367	1.48E+03	1.20E+05
2	298	282637	78.51	104.6	2988.2	844575883	6.6308E+10	6.82	0.367	9.61E+02	7.54E+04
3	298	240678	66.86	104.6	3547.6	853829273	5.7083E+10	7.63	0.367	1.28E+03	8.55E+04
4	298	5316	1.48	104.6	3222.8	17132404.8	25298851.1	6.78	0.367	1.21E+03	1.78E+03
5	298	221128	61.42	104.6	2969.4	656617483	4.0332E+10	7.8	0.367	6.50E+02	3.99E+04
6	298	206904	57.47	104.6	2460.9	509170054	2.9264E+10	8.1	0.367	5.19E+01	2.98E+03
7	298	26196	7.28	104.6	2988.2	78278887.2	569609369	6.59	0.367	1.03E+03	7.49E+03
8A	298	8241	2.29	104.6	3100.4	25550396.4	58489115.8	6.53	0.367	1.16E+03	2.65E+03
8B	298	13902	3.86	104.6	2969.3	41279208.6	159406544	7.81	0.367	6.47E+02	2.50E+03
9	298	12480	3.47	104.6	2741.9	34218912	118625562	7.98	0.367	3.69E+02	1.28E+03
10	298	1927	0.54	104.6	2524.2	4864133.4	2603662.52	7.52	0.367	2.88E+02	1.54E+02
11	298	292348	81.21	104.6	1000.9	292611113	2.3762E+10	2.6	0.367	2.31E+02	1.87E+04
12	298	240678	66.86	104.6	2988.2	719194000	4.8082E+10	6.82	0.367	9.61E+02	6.42E+04
13	298	11197440	3110.40	104.6		0	0		0.367	4.77E+00	1.48E+04
14	298	11197440	3110.40	104.6		0	0		0.367	4.77E+00	1.48E+04
15	298	209252	58.13	104.6	154.9	32413134.8	1884031468	0.52	0.367	4.71E+00	2.74E+02
16	298	209252	58.13	104.6	156.54	32756308.1	1903978605	0.52	0.367	6.35E+00	3.69E+02
17	298	292348	81.21	104.6	649.8	189967730	1.5427E+10	1.86	0.367	1.00E+02	8.14E+03
18	298	292348	81.21	104.6	666.8	194937646	1.583E+10	1.86	0.367	1.17E+02	9.52E+03
19	298	209252	58.13	104.6	161.7	57.1	3318.96922	0.56	0.367	-4.14E-01	-2.41E+01
20	298	1927	0.54	104.6	194.2	89.6	47.9608889	0.8	0.367	-3.94E+01	-2.11E+01
21	298	209252	58.13	104.6	183.1	78.5	4562.85611	0.63	0.367	1.26E-01	7.32E+00
22	298	209252	58.13	104.6	328	223.4	12985.2491	1.2	0.367	-2.48E+01	-1.44E+03
23	298	243877	67.74	104.6	329.4	224.8	15228.7638	1.011	0.367	3.29E+01	2.23E+03
24	298	243877	67.74	104.6	483.8	379.2	25688.3773	1.49	0.367	4.45E+01	3.02E+03
25	298	243877	67.74	104.6	571.5	466.9	31629.492	1.7	0.367	6.97E+01	4.72E+03
26	298	292348	81.21	104.6	885.3	780.7	63398.9121	2.36	0.367	1.87E+02	1.52E+04
27	298	292348	81.21	104.6	1000.9	896.3	72786.5312	2.6	0.367	2.31E+02	1.87E+04
28	298	1927	0.54	104.6	2524.2	2419.6	1295.15811	7.5	0.367	2.94E+02	1.57E+02
29	298	12480	3.47	104.6	2741.9	2637.3	9142.64	7.98	0.367	3.69E+02	1.28E+03
30	298	5121	1.42	104.6	3222.8	3118.2	4435.6395	6.78	0.367	1.21E+03	1.72E+03
31	298	15564	4.32	104.6	3078.5	2973.9	12857.161	6.54	0.367	1.13E+03	4.90E+03
32	298	22144	6.15	104.6	352.3	247.7	1523.63022	0.65	0.367	1.63E+02	1.00E+03
33	298	8241	2.29	104.6	506.5	401.9	920.016083	1.58	0.367	4.04E+01	9.25E+01
34	298	41761	11.60	104.6	682.7	578.1	6706.12058	1.7	0.367	1.81E+02	2.10E+03
35	298	15564	4.32	104.6	905.8	801.2	3463.85467	2.43	0.367	1.86E+02	8.06E+02

Table C3: Calculated Flow data corresponding to 100% load

Flow, j	To	mj,kg/h	Mass flow rate, mj,kg/s	ho,kj/kg	Specific enthalpy,hj, kj/kg	en,kj/kg	Energy flow rate En,kj/s	Specific entropySj,KJ/KgK	So,KJ/KgK	Specific Exergy,e,kj/kg	Exergy flow rate,Ex,kj/s
						en=(hj-ho)	En=mj*en			ej=(hj-ho)-To(sj-s	Ex=ej*mj
1	298	372484	103.47	104.6	3419.1	1273560044	1.3177E+11	6.56	0.367	1.47E+03	1.52E+05
2	298	360735	100.20	104.6	2975	1073186625	1.0754E+11	6.73	0.367	9.74E+02	9.76E+04
3	298	303367	84.27	104.6	3543.4	1074950628	9.0585E+10	7.52	0.367	1.31E+03	1.10E+05
4	298	7373	2.05	104.6	3218.2	23727788.6	48595829.3	7.58	0.367	9.64E+02	1.97E+03
5	298	276290	76.75	104.6	2964.6	819089334	6.2863E+10	7.51	0.367	7.31E+02	5.61E+04
6	298	256562	71.27	104.6	2443.4	626883590.8	4.4676E+10	8.1	0.367	3.44E+01	2.45E+03
7	298	35407	9.84	104.6	3009.8	106567988.6	1048125770	6.72	0.367	1.01E+03	9.95E+03
8A	298	10998	3.06	104.6	3095.4	34043209.2	104002004	7.65	0.367	8.20E+02	2.51E+03
8B	298	18309	5.09	104.6	2963.8	54264214.2	275978749	7.56	0.367	7.16E+02	3.64E+03
9	298	16229	4.51	104.6	2737.3	44423641.7	200264245	7.73	0.367	4.39E+02	1.98E+03
10	298	3682	1.02	104.6	2520.3	9279744.6	9491116.56	7.52	0.367	2.84E+02	2.91E+02
11	298	372484	103.47	104.6	1063.8	396248479.2	4.0999E+10	2.74	0.367	2.52E+02	2.61E+04
12	298	303367	84.27	104.6	3009.8	913073996.6	7.6943E+10	6.73	0.367	1.01E+03	8.50E+04
13	298	13996800	3888.00	104.6		0	0		0.367	4.77E+00	1.85E+04
14	298	13996800	3888.00	104.6		0	0		0.367	4.77E+00	1.85E+04
15	298	260643	72.40	104.6	163.4	42589066.2	3083483884	0.56	0.367	1.29E+00	9.31E+01
16	298	260643	72.40	104.6	164.97	42998275.71	3113110993	0.56	0.367	2.86E+00	2.07E+02
17	298	372484	103.47	104.6	687.6	256119998.4	2.65E+10	2.01	0.367	9.34E+01	9.66E+03
18	298	372484	103.47	104.6	706.3	263085449.2	2.7221E+10	1.98	0.367	1.21E+02	1.25E+04
19	298	260643	72.40	104.6	168.9	64.3	4655.37358	0.56	0.367	6.79E+00	4.91E+02
20	298	3682	1.02	104.6	212.6	108	110.46	0.72	0.367	2.81E+00	2.87E+00
21	298	260643	72.40	104.6	201.5	96.9	7015.64075	0.68	0.367	3.63E+00	2.63E+02
22	298	260643	72.40	104.6	351	246.4	17839.5653	1.1	0.367	2.80E+01	2.02E+03
23	298	306180	85.05	104.6	352.6	248	21092.4	1.12	0.367	2.36E+01	2.01E+03
24	298	306180	85.05	104.6	513.1	408.5	34742.925	1.56	0.367	5.30E+01	4.51E+03
25	298	306180	85.05	104.6	605.1	500.5	42567.525	1.78	0.367	7.94E+01	6.76E+03
26	298	372484	103.47	104.6	937.8	833.2	86209.3524	2.49	0.367	2.01E+02	2.08E+04
27	298	372484	103.47	104.6	1063.8	959.2	99246.2924	2.74	0.367	2.52E+02	2.61E+04
28	298	3682	1.02	104.6	2520.3	2415.7	2470.72428	7.52	0.367	2.84E+02	2.91E+02
29	298	16229	4.51	104.6	2737.3	2632.7	11868.3579	7.73	0.367	4.39E+02	1.98E+03
30	298	7306	2.03	104.6	3218.2	3113.6	6318.87822	7.58	0.367	9.64E+02	1.96E+03
31	298	21888	6.08	104.6	3103.7	2999.1	18234.528	6.74	0.367	1.10E+03	6.69E+03
32	298	29307	8.14	104.6	375.4	270.8	2204.53767	1.18	0.367	2.85E+01	2.32E+02
33	298	10998	3.06	104.6	535.9	431.3	1317.6215	1.6	0.367	6.39E+01	1.95E+02
34	298	57295	15.92	104.6	721.8	617.2	9822.90944	2.2	0.367	7.10E+01	1.13E+03
35	298	21888	6.08	104.6	956.9	852.3	5181.984	2.48	0.367	2.23E+02	1.35E+03