Effects of Condenser Pressure Variation & Cooling Tower Performance Study of Coal Fired Thermal Power Plant

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Major Project Report

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For the Degree of

Master of Technology in Mechanical Engineering

(Energy System)

By

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Declaration

This is to certify that

1. The thesis comprises of my original work towards the degree of Master of Technology in Mechanical Engineering (Energy System Engineering) at Nirma University and has not been submitted elsewhere for a degree.

2. Due acknowledgment has been made in the text to all other material used.

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I, Kashyap N. Katesia, Roll. No. 13MMEN01, give undertaking that the Major Project entitled **"EFFECTS OF CONDENSER PRESSURE VARIATION & COOLING TOWER PERFORMANCE STUDY OF COAL FIRED THERMAL POWER PLANT"** submitted by me, towards the partial fulfillment of the requirements for the degree of Master of Technology in Mechanical Engineering (Energy System Engineering) of Nirma University, Ahmedabad, is the original work carried out by me and I give assurance that no attempt of plagiarism has been made. I understand that in the event of any similarity found subsequently with any published work or any dissertation work elsewhere; it will result in severe disciplinary action.

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This is to certify that the Major Project Report entitled **"EFFECTS OF CONDENSER PRESSURE VARIATION & COOLING TOWER PERFORMANCE STUDY OF COAL FIRED THERMAL POWER PLANT"**submitted by **Mr. Kashyap N. Katesia (13MMEN01)**, towards the partial fulfillment of the requirements for the award of Degree of Master of Technology in Mechanical Engineering (Energy System Engineering) of Institute of Technology, Nirma University, Ahmedabad is the record of work carried out by him under our supervision and guidance. In our opinion, the submitted work has reached a level required for being accepted for examination. The result embodied in this major project, to the best of our knowledge, has not been submitted to any other University or Institution for award of any degree.

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Abstract

In India thermal power plants contributes 58% of electricity generation. Out of total power produced certain energy is consumed by auxiliary systems used in power plants. It is necessary to conserve the energy usage by auxiliaries of plant by implementing energy conservation strategies. Energy audit is one of tool to know health of plant equipments. Energy audit ought to cowl evalutaion of the performance of major equipments, determine the manageable losses and recommend that parameters for improvement. In present work energy audit and exergy analysis of six cooling tower and nine circulating cooling water pumps was carried out. Based on that assesment it is evident that cooling tower number 4 and 5 have poor effectiveness 57.14% and 59.32% respectively.In natural draft cooling tower insufficient heat transfer takes place due to non-uniform distribution of water which reflects the lower effectiveness of cooling tower. Overall efficiency of circulating cooling water pump is between 51.9% to 76% for all pumps. Specific power consumption is 0.10 to 0.17 kW/m^3/hr for all pumps is higher than design value. Higher inlet temperature of cooling water at condenser leads to poor vacuum and higher heat rate of power plant. Total 48.18 crore kcal heat rate increases annualy which required approximately 137 tonnes of extra bituminous. Heat rate of all seven units, as per direct method is greater than design heat rate. Condenser effectiveness is in range of 50.67% to 73.68% . Condenser pressure increases with increament in cooling water temperature. So it is desirable to operate cooling water pumps at optimum flow rate to control cooling water temperature. Exergy analysis has been carried out for cooling tower, circulating cooling water pumps and condenser in which exergy efficiency of cooling tower is between 30.98 to 46.79%, In circulating cooling water pump exergy efficiency is between 56.53 to 68.49% and in condenser it is between 27.72 to 56.55% .

Keywords: Energy audit, Cooling water pump, Cooling tower,Condenser pressure, Exergy analysis.

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Nomenclature

d	Diameter, m
kg/m^3	Density
Q	Volume Flow Rate,m^3/s
ρ	Density of Fluid, kg/m^3
P	Pressure, Pa
L/G	Liquid/Gas Ratio
η_{motor}	Motor Efficiency in %
h_d-h_s	Total Head in meter
P_m	Motor Power
P_h	Hydraulic power in kW
P_s	Pump Shaft Power in kW
g	Acceleration Due to Gravity (m/s^2)

Abbreviations

WTPS	Wanakbori Thermal Power Station
GSECL	Gujarat State Electricity Corporation Limited
NDCT	Natural Draft Cooling Tower
CCW	Circulating Cooling Water
OECD	Organisation for Economic Co-operation and Development
ECL	Eastern Coalfields Limited
MCL	Mahanadi Coalfields Limited
MDO	Mine Developer and Operator
TWh	TeraWatt-hours
CIL	Coal India Limited
OCP	Opencast Projects
MW	Megawatt
GW	Gigawatt
ISO	International Oraganisation for Standardizataion
PAT	Perform-Achieve-Test
TPS	Thermal Power Station
APC	Auxilliary Power Consumption
NA	Not Available
MT	Metric Tonnes
RH	Realtive Humidity

WBT	Wet Bulb Temperature
DBT	dry Bulb Temperature
ERDA	Electrical Research & Developement Association
ABS	Acrylonitrile Butadiene Styrene
PVC	Polyvinyl Chloride
NPSHR	Net Positive Suction Head Requirements

Chapter 1

Energy Scenario

1.1 Global Energy Scenario

Consumption and production redoubled for all fuels, reaching record levels for each fuel sort except atomic energy. for every of the fossil fuels, international consumption rose sooner than production. the info suggests that growth in international carbonic acid gas emissions from energy use conjointly accelerated in 2013, though it remained below average. Global primary energy consumption multiplied by 2.3% in 2013, an acceleration over 2012 (+1.8%). Growth in 2013 accelerated for oil, coal, and nuclear energy. However international growth remained below the 10-year average of 2.5%. All fuels except oil, nuclear energy and renewables in power generation grew at below-average rates. Growth was below average for all regions except North America. Oil remains the world's leading fuel, with 32.9% of worldwide energy consumption.[1]

Coal reqiurement grew by third in 2013, well below the 10-year average of 3.9% but it's still The fastest-growing fuel. Coal's share of world primary energy consumption reached 30.1%, the very best since 1970. Consumption outside the Organisation for Economic Co-operation and Development (OECD) rose by a below-average 3.7%, however still accounted for eighty nine of world growth. Asian nation intimate its second largest meter increase on record and accounted for twenty first of world growth. International coal production grew by 0.8%, the weakest growth since 2002.[2]

1.2 Electricity Sector in India

The power part in Asian nation had partner degree put in capacity of 254.649 GW as completion of October 2014. Asian nation turned into the world's third biggest maker of power in year 2013 with 4.8% global experience power era. The whole yearly era of power from each sorts of sources was 1102.9 Terawatt-hours (Twh) in 2013. As of March 2013, the every capita all out power utilization in India was 917.2 kwh.[2]

The per capita normal yearly residential power utilization in India in 2009 was 96 kwh in rustic territories and 288 kwh in urban zones for those with access to power as opposed to the overall every capita yearly normal of 2,600 kwh and 6,200 kwh in the European Union.[3] Electric vitality utilization in agribusiness is most elevated (18%) in India. The per capita power utilization is lower contrasted with numerous nations not withstanding less expensive power levy in India. Gujarat has the most elevated force surplus of any Indian state, with around 1.8 GW more power accessible than its inside interest. The state was anticipating that more limit will get to be accessible.[4]

India's evaluated interest of coal in the nation would reach to 769.69 mn tons by 2013 -14 against the household accessibility of 614.55 mn tons.india foreign 137.56 mn huge amounts of coal in 2013-14 higher when contrasted with 102.85 mn tons in 2011-12 and 68.92 mn tons recorded in 2010-11, as per the information discharged by the Ministry of Coal. In the interim, Coal India Limited (CIL) has anticipated to contribute Rs. 25,400 crore amid XII-plan Period (2012-17). In request to take care of the coal demand of the nation CIL has chosen to execute some of its mine/tasks through the Mine Developer and Operator (MDO) course. Two mines in particular, Rajmahal Opencast Projects (OCP), Eastern Coalfields Limited (ECL), Mahanadi Coalfields Limited (MCL) are now working under MDO idea.[5]

1.3 Energy Audit

Energy Audit is a specialized study of a plant in which the equipmentwise example of energy utilization contemplated and endeavors to adjust the aggregate energy info connecting with total energy input for production. As a consequence of the study the regions where the energy is inefficiently utilized and the upgrades are felt, are recognized and remedial measures are prescribed so that the general plant effectiveness could be made strides. Basic understanding of the methodology is key on the off chance that we have to enhance the general proficiency of the framework.

Execution Improvement in thermal power plants is a high need inside the electric utilities in the new aggressive environment. Then again, numerous utilities have downsized and need experienced staff in the territory of execution engineering say Energy Audit. In this task is to study vitality review procedures and through which decrease of plant heat rate by enhancing adequacy of cooling tower and condenser vacuum.In this project work methodology to evaluate performance of plant heat rate , cooling tower, circulating cooling water pump have been studied to analize which factors causes the poor effectiveness of cooling tower and poor efficiency of circulating cooling water pump.

1.4 Motivation for Project

Table 1.1: Needed Change in Heat Rate

Basline Heat Rate	**Target Net heat rate**	**Reduction**
2887 kCal/kWh	2820 kCal/kWh	-67 kCal/kWh

Energy audit and exergy analysis of various components of thermal power plant is needed to analyse the performance of cooling tower, circulating cooling water pump and condenser. Plant heat rate analysis is very important parameter. To achieve better efficiency by utilization of energy is indicated in Perform-Achieve-Test scheme. Heat rate decreases if the components of plant are working efficiently. According to that Bureau of Energy Efficiency give target to GSECL-WTPS to decrease current heat rate by 67 kCal/kWh. Various parameters influencing the heat rate like cooling water temperature, flow rate of cooling water, condenser performance. So all thermal power plant have to attempt to complete neat heat rate targets so that e-scarts can be earned from energy efficiency reserve funds.

1.5 Objectives of Project

The objectives of the work is described below:

- Energy audit of cooling tower and circulating cooling water pump.
- Energy audit of condenser.
- Exergy analysis of cooling tower, circulating cooling water pump and condenser.
- Analysis of plant heat rate.

1.6 Organisation of Report

This report os organised as follows:

Chapter-1 This chapter includes energy scenario across the globe and India. Motivation of project work, objectives of project work. Basic information about energy and exergy. It will give idea about exergy in power plant to improve its performance.

Chapter-2 This chapter includes literature review of energy audit, necessity and use of energy audit, research work carried out in exergy analysis of thermal power plant.

Chapter-3 This chapter includes energy audit and exergy analysis of cooling tower. In that literature survey, measured data, calculated data, suggestions and discussions are included.

Chapter-4 This chapter includes energy audit and exergy analysis of circulating cooling water pump. In that literature survey, measured data, methodology and suggestions are inlcuded.

Chapter-5 This chapter inlcudes introduction about plant heat rate, literaure survey, methods to calculate heat rate, calulated plant heat rate for seven units of WTPS, suggestions and discussions are included.

Chapter-6 This chapter includes energy audit and exergy analysis of condenser. In that literature survey, methodology and results of energy audit and exergy analysis are discussed.

Chapter-7 This chapter includes conclusions and future work.

Chapter 2

Literature Survey

2.1 Basics of Energy Audit

According to the Energy Conservation Act, 2001, Energy Audit is characterized as "The verification, checking and examination of utilization of energy including submission of specialized report containing proposals for enhancing energy efficiency with expense advantage investigation and an activity plan to diminish energy consumption". The energy audit is one of the first errands to be performed in achieving a compelling energy management project intended to enhance the energy efficiency and lessen the energy operating expenses of a facility. An energy audit comprises of an itemized examination of how a facility utilizes energy , lastly , asuggested system for changes in working practices or energy expending gear that will cost viably spare dollars of energy bills. An energy audit is procedure for recognizing energy misfortunes, measuring them, assessing conservation potential, advancing mechanical choices for for conservation and assessing techno money matters for the measures recommended e.g. Support commercial ventures in diminishing their energy utilization, To advance energy proficient innovations among industry segments, Disseminate data on energy effectiveness through preparing projects and workshops, To advance exchange of energy efficient and ecological sound advances to the industrial divisions in the connection of environmental change. The energy audits here and there called and energy surveyor an energy examination, so it is not mistaken for a monetary review. The energy audit is a positive involvement with critical profits to the facility. The expression 'audit' ought to be dodged on the off chance that it plainly delivers a negative picture in the psyche of a specific bussiness, association, or single person. An energy audit recognized where energy is expended and the amount of energy is devoured in a current facility, building or structure. Data assembled from the energy audit can be utilized to present energy preservation measures or suitable energy-sparing innovations, for example, electronic control frameworks, as retrofits. Energy audits recognized finacially legitimized, expense-sparing open doors that bring about essentially brought down electrical, natural gas, steam, water and sewer costs. An energy audit, hence, is an itemized examination of an facility's energy uses and expenses that creates suggestions to diminish those uses and expenses by executing mechanical and operational changes.[7]

Bhansali V.K. et al.[8] introduced the energy protection was practical with a short payback period and humble speculation. The results showed that the energy protection ought to be created as a mass developement like family arranging, education drive and so forth.

Babu N. Sundar et al.[9]represented the Government has given higher need for the power advancement, the Indian Power part was battling with imposing challenges of taking care of the high demands of power because of higher measure of energy losses and energy burglaries. The results demonstrated that their primary capacities for financial and emission controlled operation of the power division had been intricately broke down energy. This is accomplished by bringing the steam up in the boilers, growing it through turbine.

Bentarzi H. et al. introducing another methodology to controlling the steam turbine of thermal power plant utilizing disseminated controlled framework with fuzzy logic system. The came about showed that it was not by and large suitable for non direct, time delay, high request and complex framework.[10]

Cropper Paul A. et al. discussed about the target of the audit was make a thorough investigation of the power plant operations and projects identifying with the execution of the producing units and to recognize zones of potential change. The results demonstrated that heat rate enhancement had been attained and an effectively abnormal state of unwavering quality had been considerably further moved forward.[11]

In the energy audit book of T.E.R.I. it was discussed that the structure of the energy audit report is governed basically by the directives issued. The energy audit reports are details of energy consumption, their costs and specific energy consumption stitutional arrangement for promotion of energy efficiency.[12]

2.2 Utilization of Energy Audit in Thermal Power Plant

Role of energy audit in thermal power plant is mention below[7]:

- Clearly identify types and costs of energy use.
- Understand how energy is being used and possibly wasted.
- Identify and analyze more cost-effective ways of using energy.

- Improved operational techniques.
- New equipment, new processes or new technology.
- Perform an economic analysis on those alternatives and determine which ones are cost effective for your business or industry.
- Identify the generic design deficiencies.
- Suggest appropriate techniques to conserve energy along with economic.
- Identifies wastage areas of fuel, power and water & air utilization.
- Reduction in cost of generation by implementing findings of energy audit.
- Increases power generation by efficient utilization of steam in turbine cycle and reduction in Auxiliary Power Consumption (APC).
- Maintenance planning and availability improvement.
- Provides guidance in loading sequences of the units.
- Identification and rectification of errors in on-line instruments.
- Leads to reduction in green house gases.
- Utilizes specialized services of experienced engineers.
- Training of operating and maintenance staff for efficient control of unit operation.
- Improves competitiveness by reducing unit generation.
- Creates bench mark for all equipments and systems.
- Fulfills bureau of energy efficiency mandatory requirement of energy audit.

2.3 Classification of Energy Audit

Type of energy audit has to be performed depends on the factors like type of energy audit, desirability of cost reduction and depth of final audit method.[7]

Depending on these factors energy audit can be categorized in two types:

- Preliminary audit
- Detailed audit

Preliminary Audit identifies the immediate need of the plant, such as it.

- Establishes the energy consumption of the plant
- Estimate the saving
- Identifies area which require immediate attention
- Identifies areas where detailed study is required
- Preliminary energy audit uses existing, or easily obtained data

Detailed Audit consist of three phases

- Phase 1 : Pre-Audit phase
- Phase 2 : Audit phase
- Phase 3 : Post-Audit phase

This type of audit offers the most accurate estimate of energy savings and cost. It considers the interactive effects of all projects, accounts for the energy use of all major equipment, and includes detailed energy cost saving calculations and project cost.

2.4 Exergy Analysis

Exergy is a measure of the maximum capacity of system to perform useful work as it proceeds to a specified fianl state in equilibium with its surroundings. Exergy is generelly not conserved as energy but detructed in the system. In contrast, exergy analysis will characterize the work potential of a system. Exergy is maximum work that can be obtained from the system,when its state is brought to the reference or dead state (standard atmospheric conditions). Exergy analysis is based on the secondlaw of thermodynamics.

Exergy analysis of the systems, which analyses the processes and functioning if systems,is based on the second law of thermodynamics. In this analysis,the effectiveness of the second law which states the exact functionality of a system and depicts the irreversible factors which result in exergy loss and efficiency decrease in mentioned. Therefore, solutions to reduce exergy loss will be identifeid for optimization of engineering installations.

The great advantage of exergy calculations over energy calculations is that exergy calculates pin point exactly where the real losses in processes and components occured. Furthermore the exergy content stream is a real evaluation of energy it indicates the fraction of energy that really can be used. Keenan et al. provides derivation for the batch or no-flow availabilty (X) and the flow availability (Y) by

$$X = u - u_0 + P_0(v - v_0) - T_0(s - s_0) \tag{2.1}$$

and

$$Y = (h - T_0 s) - (h_0 - T_0 s_0) \tag{2.2}$$

The specific exergy of a stream of matter (in the absence of nuclear effects, magnesium, electricty and surface tension) is due to the contribution of kinetic, potential, physical and chemical exergy.

Li Wang et al.[10] discussed that the thermal efficiency and water to air ratio have complex effects on exergy performance of the counter flow wet cooling tower. Performance is very close to 1.0 epecially at the conditions of lower water to air ratio and higher thermal efficiency.

G.P.Verkhikar et al.[11]wrote on the exergy basis of power plant that a reduction in exergy destruction is achieved by increasing the value of thermodynamic parameter of the working fluid supplied to the turbine and by reducing the temperature differences of the net heaters.

S.C.Kaushika et al.[12] discussed in the paper "Energy and exergy analysis of thermal power plants" that exergy analysis helps to understand the performance of coal fired, gas fired combined cycle power plants. It helps to identify design possible efficiency improvements. It gives logical solution improving the power production opportunities in thermal power plants.

2.5 Exergetic Efficiency

For a control volume at steady state the exergy equation can be written as follows,

Exergy = Exergy in + Exergy loss +Exergy destruction

The Exergetic efficiency is a measure of performance in terms of optimal performance permitted by both first and second laws of thermodynamics and is devoid of the drawbacks inherent in the definition of the first law of efficiency. For a device whose output is either work or heat transfer, it is deained as a ratio of the energy transfer achieved by device or system to the maximum possible heat or work usually tranferable by any device or system using the same energy input as given system. While numerator is same for both first and second law efficiencies, the denominator in latter case brings both laws of thermodynamics directly into the definition of efficiency. First law focuses attention on reading losses, to improve efficiency. The second law efficiency point out that both losses and internal irreversibilty need to be improve performance.

From exergy balance, exergetic efficiency EX, is defined as,

$$\eta_{EX} = \frac{E_{out}}{E_{in}} \tag{2.3}$$

$$\eta_{EX} = 1 - \left[\frac{(Loss + Destruction)}{(Input)}\right] \tag{2.4}$$

Percentage loss of exergy in the component is definded as follows,

$$\eta_{EX} = \left[\frac{(Loss + Destruction)}{(Input)}\right] * 100 \tag{2.5}$$

Thus one can find out the component in which the losses are considerable, so that one can be suggest the ways of reducing losses and thus increase the exergetic efficiency.

Chapter 3

Energy Audit and Exergy Analysis of Cooling Tower

3.1 Introduction to Cooling Tower

Cooling towers are heat removal devices used to remove waste heat to the atmosphere.

A typical main cooling water system consists of the main condenser, cooling tower, coolant discharge, cooling water pump and cooling water treatment plant, as shown in Figure 3.1.

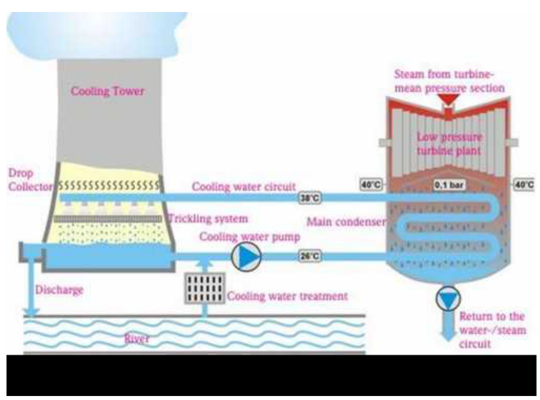

Figure 3.1: Condenser Cooling Water Circuit[14]

Table 3.1: Data of Cooling Tower in Wanakbori Thermal Power Station[6]

Location	Behind C.W. Pump House
Total Nos.	7 (One Per Unit)
Type	NDCT
Dia. of Tower at Base	94.018
Dia. of Tower at Leg Top	90.528 m
Dia. of Throat	47.01 m
Dia. of Top	50.884 m
Height of Tower above 0-0 Level	112.65 m
Height of Throat of Tower	90.12 m
Height of Top of Air Inlet	5.65 m
Depth of basin from 0-0 level	3 m
Nos. of 'A' Column	36
Individual Length of Column	6.347 m
Diameter of Column	650 mm
Height of Column	5 m
Material of Tower	RCC
Shell Thickness	175 mm
Nos. of Louvers	227568 / tower
Nos. of Splash Plates	6412 / tower
Capacity	30000 m^3/hr
Temperature Range of Cooling	42 °C to 33 °C (9 °C)
Design Atmospheric Wet Bulb Temperature	28 °C
Design atm. Relative humidity	55.00%
Dry Air flow	2880 kg/m^2/hr
Total Dry air flow per tower	16719000 kg/hr
Temp. of leaving air wet bulb / Dry bulb	38.7 °C / 38.9 °C
Evaporation loss	408457 kg/hr
Effective fill volume	32785 m^3
Effective cooling surface of fill	5805 m^2
Area of water surface in basin	6936 m^2
Total weight of water in basin	18728 MT
Total makeup per tower	4102 m^3/hr
Sensible heat gain by dry air through tower	0.72 kcal/kg
Latent heat gain by dry air	15.03 kcal/kg
Total heat exchange / hr	26300 kcal/kg

The warmed cooling water from condenser reaches the cooling tower, which is basically an air cooled heat exchanger. Cooling towers used in power plants could be natural or forced draft type. Natural draft cooling towers create air flow by the virtue of their height. Mechanical draft towers make use of a fan for this purpose. In the towers, water is sprayed down against the air flow. In this process, a small portion of water evaporates and escapes as vapour into the atmosphere. The remaining cooling water is collected in cooling tower basins. From here it can be rerouted into the extraction waters (discharge operation) or else it is again rerouted into the cooling circuit. The volume of cooling water of a coal power plant is substantial: per 100 kW power plant performance, approximately 3 - 4 m^3 water per second flow through the condenser (600MV requires approximately 75 000 m^3/h cooling water). At WTPS, all the towers are of natural draft type. The design water flow is 3.3×10^7 kg/h and the air flow is about 1.6×10^7 kg/h.

3.2 Literature Survey

Energy audit of cooling tower Performance of existing NDCT is improved either by optimization of operation method or by employing some innovative ideas. Georgia et al.[22] worked on optimization model for the operation of cooling tower system and concluded that:

- In the most economical operation of the cooling water system, the temperature of water that leaves the tower must be maintained at the highest value possible, provided the thermal requirements are achieved.
- When there is an increase of thermal demand of the process without a simultaneous requirement of a lower water outlet temperature from the cooling tower, the optimal solution prescribes increasing the flow rate of circulating water through the system, keeping the other operational conditions constant.
- In situations when cooler water is needed to fulfil the process thermal demand, and its availability is reached (i.e., increase of water flow rate), the most economical expenditure is to increase the air flow rate through the cooling tower.

M. Goodarzi etal[23] experimented on radiator type wind breaker for enhancement of heat rejection through Natural Draft Cooling Tower. They concluded that Regardless of increasing the initial and fabrication costs, radiator type windbreakers improve the cooling efficiency more than solid windbreakers do. In fact, the radiator type windbreakers use the cooling potential of the blowing wind in addition to the velocity deceleration characteristic. They even improve cooling efficiency under normal condition.The water particles that are carried away with saturated exit air are called as drift. This drift is a huge evaporation loss that needs to be eliminated.

Manuel Lucas et al. [24]investigated the thermal performance of cooling tower with drift eliminator. The presence of an eliminator does not necessarily worsen the performance of cooling tower as expected by the additional pressure loss incorporated into the air flow.

Further, the performance of NDCT will be improved by using the munters media of PVC as a cooling media.

Improving the efficiency of NDCT researched by Smrekar e tal.[26] measured the velocity and temperature across the periphery of Natural Draft Cooling Tower. These provide a direct means of evaluating the extent to which the falling hot water droplets and the cooling air mix in the cooling tower.

Fisenko et al.[36] formulated a mathematical model of the performance of a cooling tower and proposed its use for creating a system that controls its operation in face of a variable hydraulic load, wind velocity, temperature and humidity of the inlet air.

N. Williamson et al.[37] presented a two-dimensional axisymmetric two-phase simulation of heat and mass transfer inside the cooling tower, particularly emphasizing the effect of these radial non uniformities. The authors concluded that a high radial non-uniformity in heat transfer may occur when the cooling load in the rain zone of the tower is high.

As book of Dr. G.G.Rajan[21] "Troubleshooting, Operation and Maintenance for power plants and allied industries" that work carried out on the cooling tower. According to book cooling towers are the most significant area of the thermal power plant, for example, system pressure, temperature and so forth. Cooling water quality is the most essential parameters that influence the execution of the procedure units.

A scientific mathematical model [39] focused around heat and mass exchange standard is produced to discover the outlet state of water and air.

3.3 Cooling Tower Parameters

The important parameters, from the point of determining the performance of cooling towers, are:

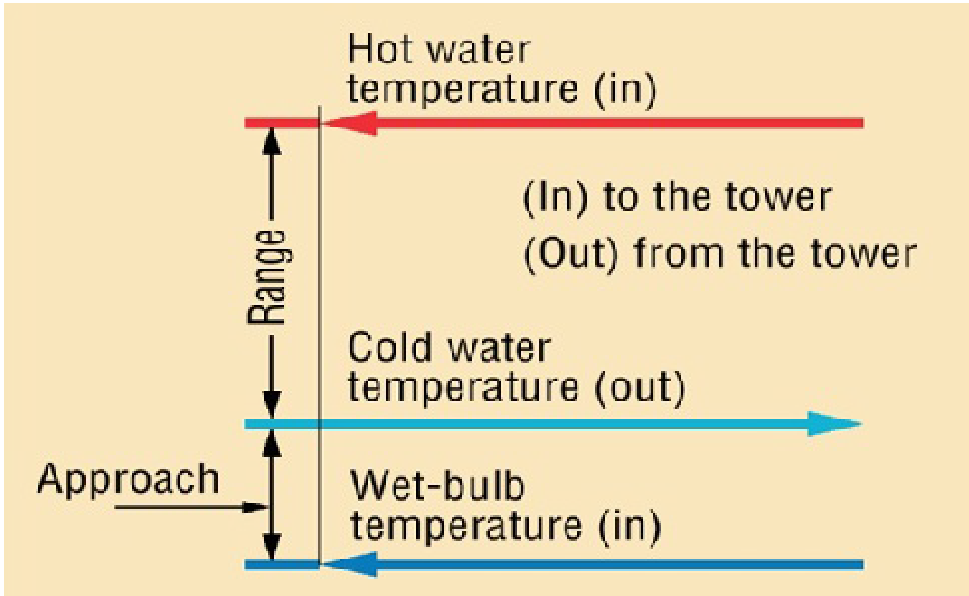

Figure 3.2: Range & Approach[14]

The important parameters, from the point of determining the performance of cooling towers, are:

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

  Effectiveness= Range / (Range + Approach).

- Cooling capacity is the heat rejected in kCal/hr or TR, given as product of mass flow rate of water, specific heat and temperature difference.
- Evaporation loss is the water quantity evaporated for cooling duty and, theoretically, for every 10,00,000 kCal heat rejected, evaporation quantity works out to 1.8 m^3. An empirical relation used often is:

  Evaporation Loss (m^3/hr) = 0.00085 x 1.8 x circulation rate (m^3/hr) x (T1-T2)

Where,

T1-T2 = Temp. difference between inlet and outlet water. [40]

- Cycles of concentration (C.O.C) is the ratio of dissolved solids in circulating water to the dissolved solids in make up water.
- Blow down losses depend upon cycles of concentration and the evaporation losses and is given by relation:

Blow Down = Evaporation Loss / (C.O.C. – 1)

- Liquid/Gas (L/G) ratio, of a cooling tower is the ratio between the water and the air mass flow rates.

$L(T1 - T2) = G(h2 - h1)$

$$\frac{L}{G} = \frac{h2 - h1}{T1 - T2} \tag{3.1}$$

Where,

$\frac{L}{G}$= liquid to gas mass flow ratio (kg/kg)
T1 = hot water temperature (°C)
T2 = cold water temperature (°C)
h2 = enthalpy of air-water vapor mixture at exhaust wet-bulb temperature.
h1 = enthalpy of air-water vapor mixture at inlet wet-bulb temperature.

3.3.1 Stability of Test Conditions

Stability of test conditions is critical to the conduct of an accurate test. For a valid test during the selected period, the limitations from maximum to minimum shall be based on the computed average of each interval of the following data. Individual readings may fluctuate during the test. Following test condition prevails during actual performance test on date 16.08.2014 Shown in Appendix A.

3.3.2 Measuring Instruments for Energy Audit

(1) Ultrasonic flow meter

Figure 3.3: Ultrasonic flow meter

Use: To measure flow of liquid.

Sensor type: from Diameter 13 mm to 6000 mm.

Temperature range : -40°C to 200°C.

Accuracy : 1 %.

(2) Relative Humidity Meter

Figure 3.4: Relative Humidity Meter

Use: To measure temperature, DBT, relative humidity.

Range : 0-100% RH and -20 to +70°C.

Accuracy : ±3 %.

(3) Digital Temperature Indicator

Figure 3.5: Digital Temperature Indicator

Type : Contact type temperature indicator.
Range : 0 to 1200 °C.
Accuracy : ±0.5%.
Resolution : 0.1°C up to 200°C.

3.4 Data Collection

3.4.1 Design Data of Cooling Tower

Design data of cooling tower which were required to evaluate cooling tower performance are mentioned in table 3.3.

Table 3.2: Design data of cooling tower[6]

Design Data for Stage 1 and 2 (Unit 1 To 6)			
S. No.	**Cooling Tower Parameters**	**Unit**	**Design Values**
1	Water Flow	kg/hr	33000000.00
2	Air Flow Calculated	kg/hr	16050000.00
3	Air inlet DBT	°C	36.00
4	Air Inlet WBT	°C	28
5	RH	%	55
6	Absolute Humidity	kg/kg of dry air	0
7	Air leaving DBT	°C	41.80
8	Air leaving WBT	°C	41.60
10	Water Outlet Temperature	°C	33.00
11	Water Inlet Temperature	°C	43.00
12	Heat Load	kJ/hr	1381380000.00
13	Range	°C	10.00
14	Approach	°C	5.00
15	Effectiveness	%	66.67
16	Liquid/Gas Ratio	Factor	2.06
17	Evaporation Loss	kg/hr	504.90
18	Windage Loss	kg/hr	66.00
19	Blow Down Loss	kg/hr	504.90
20	Make up Flow	kg/hr	1075.80

3.4.2 Measured Data

3.6.2.1 Air Side Measurement

Measured data for air side to calculate cooling tower's performance is mentioned in Appendix B.

3.6.2.2 Water Side Measurement

Measured data for water side to calculate cooling tower's performance is mentioned in Appendix C.

3.6.2.3 Calculated Parameter

Measured, theoretical and calculated data for six cooling towers is mentioned in Appendix D

3.5 Result and Discussion

3.5.1 Effectiveness

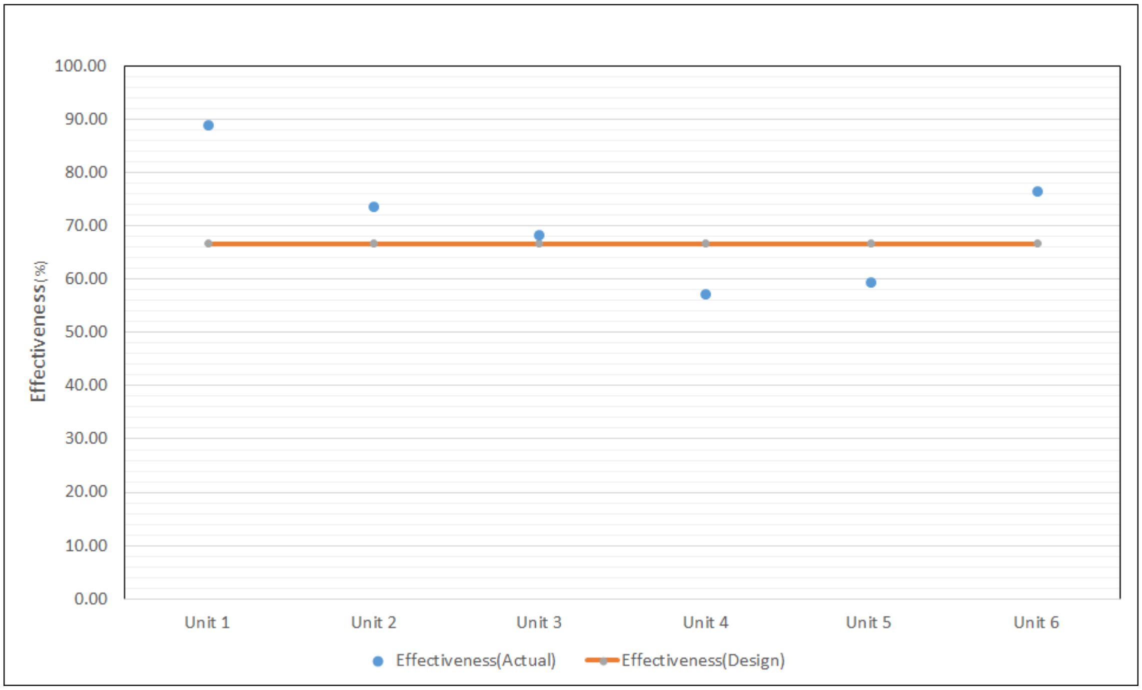

Figure 3.6: Effectiveness of Each Cooling Tower

- From the figure 3.5 it can be observed that present operating effectiveness of NDCT number 4 and 5 are 57.14% and 59.32% Respectively is lower as compare to design value of 66.67%.
- In cooling tower number 4 approach and range at present operating condition is 14.00 °C and 8.00 °C respectively. Approach is deteriorated by 2.4 °C.
- From above performance mentioned in appendix D, it is observed that Cold Water Temperature (CWT) never cross the Dry Bulb Temperature (DBT) toward approaching to Wet Bulb Temperature (WBT). This shows that heat transfer taken place inside the cooling tower mainly due to Sensible heat transfer instead of Evaporating cooling (Latent Heat Transfer). Check nozzles for appropriate functioning and distribution header for uniform distribution of hot water above the splash bars.
- L/G ratio can not possible to calculate due to outlet air side cooling tower parameters is not possible to measure.

3.5.2 Heat Load Analysis

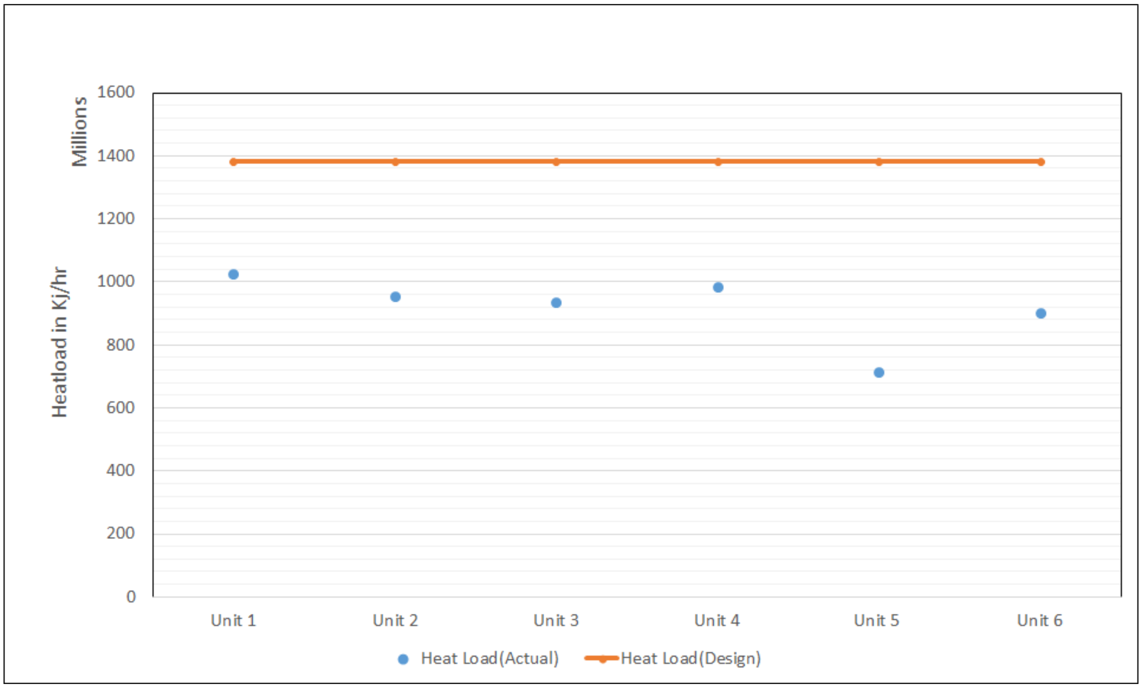

Figure 3.7: Cooling Tower Wise Heat Load

As shown in figure 3.7 we can see the graph of heat load of each cooling tower. In that heat load is too much lesser in cooling tower number 5 and 6 as compare to the design value, which is not vaible because if heat load decreases the effectiveness of cooling tower decreases.

3.5.3 Effect of Water Outlet & Inlet Temperature on Performance

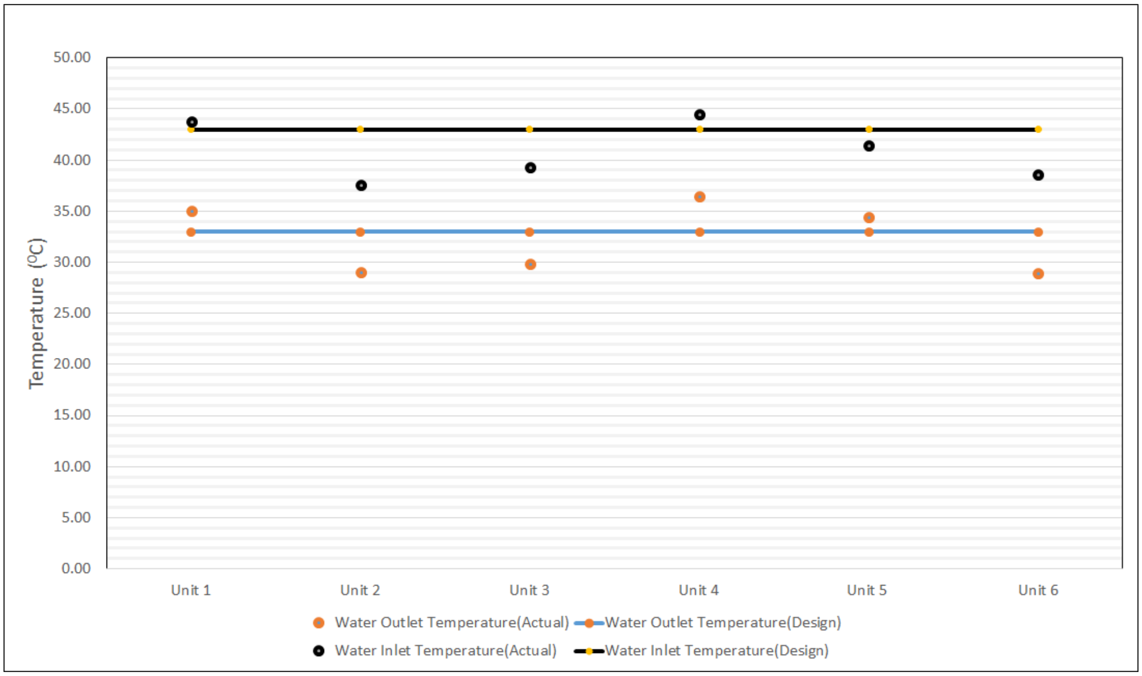

Figure 3.8: Water Inlet and Outlet Temperature

As shown in figure3.8 water inlet temperature in cooling tower number 1 and 4 are above design value which is not acceptable. Water outlet temperature from cooling tower number 1,4 and 5 is higher as compare to its design value. Condenser efficiency is affected because of that higher outlet water temperature which is working as cooling media in condenser. One degree of rise in cooling water temperature results in vacuum loss by 5 mm of Hg and increases heat rate by 12.5kCal/kWh.

3.6 Exergy Analysis of Cooling Tower

Data taken during energy audit of cooling tower has shown in table 3.4.

Table 3.3: Data for exergy analysis

Sr. No.	Parameter	Unit	CT-1	CT-2	CT-3	CT-4	CT-5	CT-6
1	T_1	°C	43.80	39.00	42.80	44.50	43.00	38.60
2	P_1	bar	1.78	1.76	1.75	1.77	1.78	1.78
3	m_1	kg/s	7265.00	7362.0	6514.0	7155.00	6737.00	6147.00
4	h_1	kJ/kg	182.80	162.74	178.62	186.10	179.45	165.00
5	S_1	kJ/kg K	0.62	0.56	0.61	0.63	0.61	0.56
6	T_2	°C	35.00	31.00	33.00	36.50	34.50	28.90
7	P_2	bar	1.22	1.21	1.22	1.21	1.22	1.21
8	m_2	kg/s	6965.00	7062.00	6264.00	6955.00	6612.00	6027.00
9	h_2	kJ/kg	146.02	129.30	137.66	149.20	143.93	121.00
10	S_2	kJ/kg K	0.50	0.45	0.48	0.51	0.50	0.42
11	P_0	bar	1.012	1.012	1.012	1.012	1.012	1.012
12	T_0	°C	27	27	27	27	27	27
13	h_0	kJ/kg	113.2	113.2	113.2	113.2	113.2	113.2
14	S_0	kJ/kg K	0.3944	0.3944	0.3944	0.3944	0.3944	0.3944

3.6.1 Second Law of Efficiency

The details of exergy analysis is discussed in section 2.5. Using equation 7 and 8 exergy available and exergy destruction is calculated using data mentioned in table 3.4. A sample calculation is mentioned under.

$$E_1 = m_1[(h_1 - h_0) - T_0(S_1 - S_0)] \tag{3.2}$$

$$\begin{aligned} &= 7265[(182.80 - 113.2) - 300(0.6200 - 0.3944)] \\ E_1 &= 16309.93\text{kJ} \end{aligned}$$

$$E_2 = m_2[(h_2 - h_0) - T_0(S_2 - S_0)] \tag{3.3}$$

$$\begin{aligned} &= 6965[(146.02 - 113.2) - 300(0.5002 - 0.3944)] \\ E_2 &= 6306.02\text{kJ} \end{aligned}$$

$$Exergy\,Destruction = m_0 * T_0 * (Sgen) \tag{3.4}$$

$$\begin{aligned} &= 6965 * 300 * (0.6200 - 0.5002) \\ &= 250322 \text{ W} \\ &= 250.32 \text{ kW} \end{aligned}$$

$$Second\,law\,Efficiency = 1 - \frac{Exergy\,Destruction}{Exergy\,in}$$

$$= 1 - \frac{250.32}{16309.93}$$

$$\eta \text{II} = 38.66\%$$

Table 3.4: Second Law Efficiency

Parameters	Unit	CT-1	CT-2	CT-3	CT-4	CT-5	CT-6
Exergy Efficiency	%	38.66	46.79	30.98	39.13	39.87	38.18
Exergy In	MW	163.09	101.96	128.06	281.34	148.34	169.04
Exergy Destruction	MW	100.03	54.25	88.39	171.25	89.19	104.49

The following Table 3.5 shows second law efficiency and exergy destruction occured in cooling towers. figure 3.10 shows the Exergy in and Exergy destruction of cooling towers.

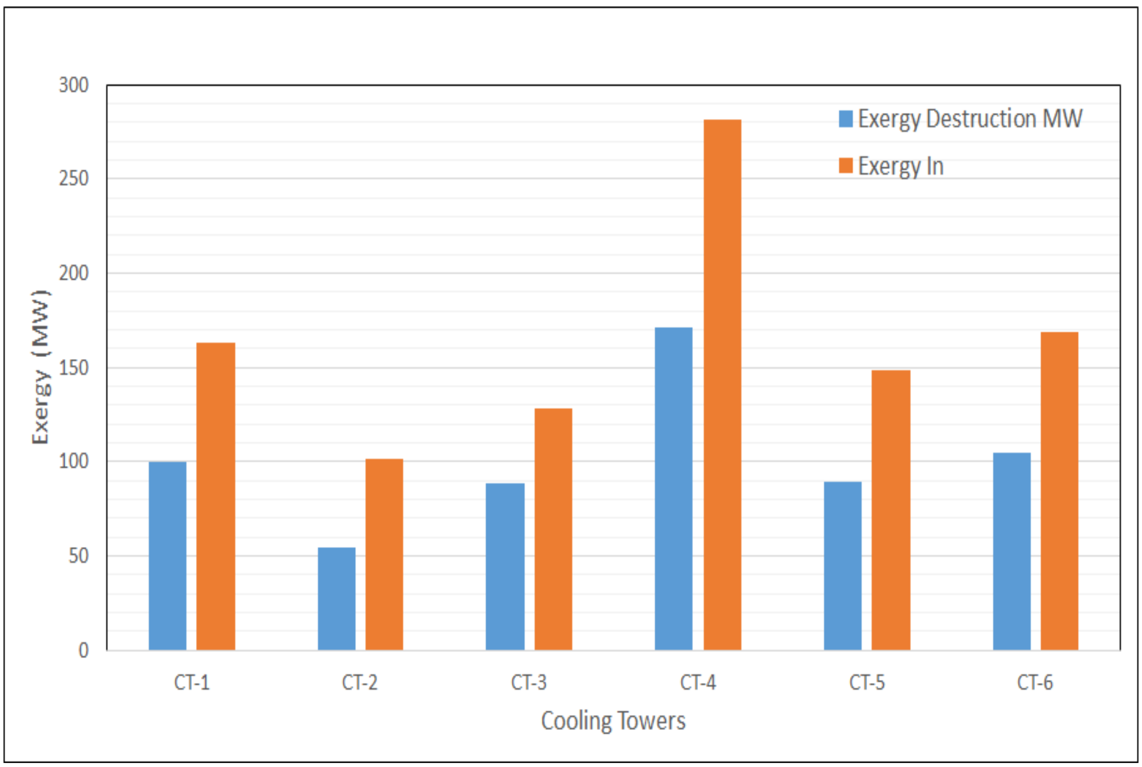

Figure 3.9: Exergy In and Exergy Destruction in Cooling Towers

Exergy in and exergy destruction in cooling towers are affecting parameters for exergy efficiency.From figure exergy destruction in cooling tower is very large as compared to exergy in. Due to exergy destruction exergy efficiency is very less in cooling tower.

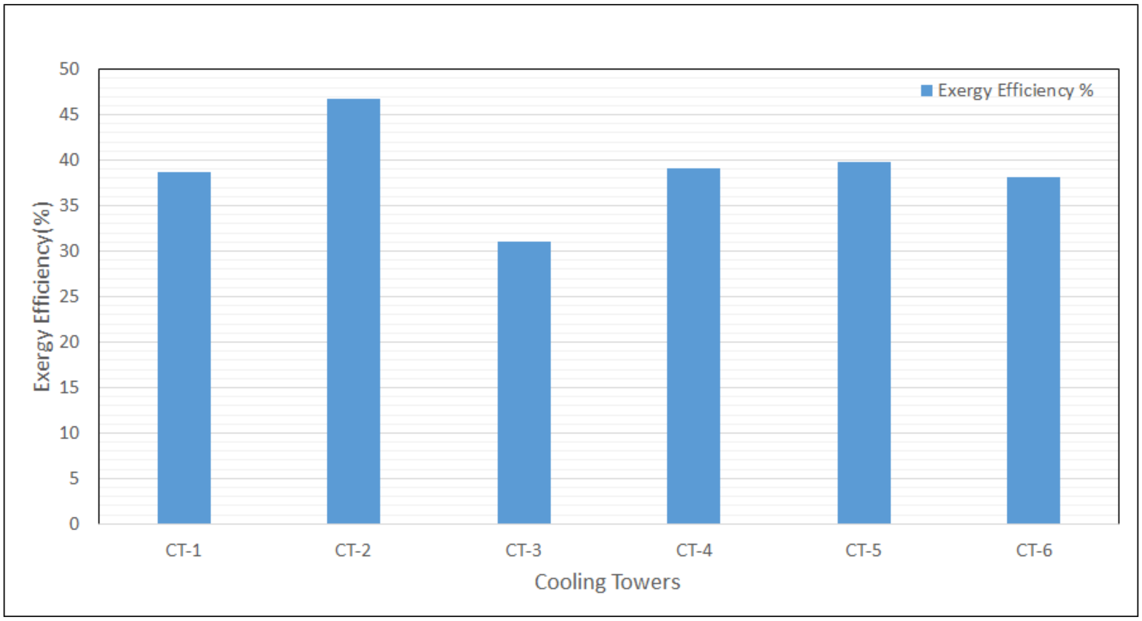

Figure 3.10: Exergy Efficiency of Cooling Towers

Figure 3.11 shows the exergy efficiency in cooling towers.

Exergy efficiency in cooling towers are very low in range of 30 to 47 %. That means exergy destruction is large because of low performance of cooling tower.

3.7 Result and Energy Saving Opportunities in Cooling Tower

- As on present performance of the cooling tower, there is scope to decrease cooling water temperature by 4.0 to 5.0 °C by taking proper corrective action to imporovise the condenser vacuum by 20 -25 mm of Hg.
- Heat rate can be reduced by 2.25 kCal/kWh if condenser vacuum gain by 1.0 mm of Hg.
- Extra dosing of chemicals on processed water may prevent the growth of algae on splash bars and other area of cooling tower.
- Conduct cooling tower health study periodically to observe any deterioration due to time and weather, this includes calculation of effectiveness, Range and approach and correspondingly vacuum in condenser.

- Cooling tower channel and basin should be covered with fish net to prevent from any foreign particle/leaf/material to enter on the cooling system.
- Moniter approach, effectiveness and cooling capacity for continuous optimization efforts, as per seasonal variations as well as load side variations.
- Nozzle falling in the cooling tower.
- Provide vent hole at the end of the header to release pipe air during initial startup of the cooling water pump after Annual Overhauling(AOH) and Comprehensive Overhauling(COH). Because this air is responsible for majority of nozzle falls.
- It is better idea to go for lawn around cooling tower area to maintain cleanliness and aesthetic view. This also reduce dust particle to enter cooling tower with incoming cooling air.
- Cooling tower effectiveness due decrease of no equal flow distribution in cooling tower system. More than 5mm thick scale in the pipe of cooling tower so because of that hot water and cold water can not contact directly.
- Consider possible improvements on CW pumps w.r.t. effieciency improvement.
- Consider COC omprovement measures for water savings.
- Replace slat type drift eliminators with low pressure drop, self extinguishing, PVC cellular units.

3.7.1 Suggestion to use Ultra-ever Dry Top Coat

Ultratech 4001 ultra-ever dry top coat, 1 quart, works with the ultra-ever dry bottom coat to form the two-part ultra-ever dry coating which helps repel water and oil from a coated surface, protecting against corrosion caused by moisture and keeping coated surfaces dry, clean, and free of ice. The top coat forms a thin coating that fills its own interstitial spaces with trapped air molecules to prevent water and oil from wetting the coated surface. This product must be applied over ultra-ever dry bottom coat and according to the instructions to be effective. The two-part coating adds a translucent white color and matte finish to coated surfaces after curing. The ultra-ever dry top coat uses a solvent called acetone to allow the active ingredients to be sprayed on for a thin, uniform coating. The top coat can help protect steel, aluminum, and other metals, as well as plastic, leather, fabric, wood, concrete, and many other materials. The two-part coating is non-flammable after application and suitable for indoor and outdoor use.[44]

Benefits:

- Top coat of two-part Ultra-ever dry coating which helps repel water and oil from coated surfaces,
- Protects against corrosion caused by moisture
- Keeps coated surfaces clean and free of ice.
- Forms a thin coating that fills its own interstitial spaces with trapped air molecules to prevent water and oil from wetting the coated surface.

Table 3.5: Specifications for Ultra-Ever Dry two-part coating

Percent of solids	>10%
Surface application temperature	10 °C to 40°C
Storage temperature	-40 to 115 degrees °C
Shelf life	1 year at 25 °C
Coverage per gallon*	240 sq. ft./22. sq. m
Working temperature	-30 to 300 °C

*1 US gallons = 3.7854 liters

3.7.2 Estimation of Cost

- Price of Ultra-ever dry coat = Rs. 5759.18 for 1 litre
- Total effectiveness cooling surface : 5805 m^2= 1393200 sq.ft
- Total required amount of Ultra-ever dry coat : 263.86 gallons = 997.40 litres
- Material Cost = 997.40 liters * 62.38 $ = Rs. 62217.81 approx. (1 $ = Rs. 62.38)
- Labour Cost = as per GSCEL's contract price for labour work.
- For Seven cooling tower total cost = 62217.81 * 7 = Rs. 435524 approx.

As per Annual overhauling cost for other work in whole power plant this amount is vaible for implementation.

3.7.3 Suggestion to Replace Old Tpye Nozzles With Spray ABS Nozzles

Figure 3.11: Cooling Tower Spray Nozzles

In Natural Draft Cooling Tower hot water is distributed through nozzle via common header. That nozzles spray hot water in uniform direction towards fill surfaces. Old type nozzle are not working properly and so many nozzles were drop off from its distribution point. Due to this problems hot water can not distributed properly. Thus improvement can done by replacing old nozzle by new nozzles shows in figure 3.12.**[45]**

Cooling tower nozzle/sprayer characteristics:

- Condition: New.
- Price per unit : 0.5 to 3 USD per piece.
- Cooling type: Counter Flow.
- Material: ABS,PVC.
- Place of origin: Hebei China (Mainland).
- Brand name: Sinta.

- Model number: ST-0421.
- Application: cooling tower.
- Colour: white, balck.

Benifits :

- Maximum operating temperature: 120°C to 300°C.
- Improves circulation of the flow and optimizes distribution of water.
- Constructed of carbon fiber-glass-reinforced polypropylene or stainless steel

Chapter 4

Energy Audit and Exergy Analysis of Circulating Water Pump

4.1 Introduction

Pumping is a process of moving liquid from one point to another by addition of kinetic and potential energy to it. The energy will help the liquid to do work such as flow through pipe or rise to a higher head level,while Circulating Water(CW) pumps in thermal power plant primarily used to circulate water through the condenser tubes of a thermal power plant, in order to maintain the adequate vacuum in the condenser.This pumps also find application as pumping units for irrigation and metro water supply projects.This are vertical, single stage, mixed flow type centrifugal pumps.

Different types of CW pumps used in thermal power plant:[41]

- Vertical mounting
- Centrifugal type
- Single-stage, Multi-shaft
- Mixed Flow Impeller
- Non pull-out
- Single/Double foundation

Circulating Cooling Water Pump Used in GSECL thermal power plant is :[6]

- 1000VMA10 For Unit-1.
- BHQ-70 for Unit 2 & 3.

4.2 Literature Survey

Pumps are often substantially larger than they need to be for an industrial plant's process requirements. Centrifugal pumps can often be oversized because of misestimation, trying to change stepwise increases in pipe surface roughness and flow resistance over time, or anticipating future plant capacity expansions. In addition, the plant's pumping requirements might not have been clearly defined during the design phase.Because of this conservative approach, pumps can have operating points completely different from their design points. The pump head is often less than expected, while the flow rate is greater. This can cause cavitation and waste energy as the flow rate typically must be regulated with bypass or throttle control. Oversized and throttled pumps that produce excess pressure are excellent candidates for impeller replacement or "trimming," to save energy and reduce costs. Trimming involves machining the impeller to reduce its diameter. Trimming should be limited to about 75% of a pump's maximum impeller diameter, because excessive trimming can result in a mismatched impeller and casing.[42]

Pumps with an optimised flow passage and a speed control system make system operation efficient. Apart from using a speed control system with the pump, offering impeller trimming as a standard is another cost-effective option of saving customers energy. Reducing the outside diameter by 10 percent reduces the head by about 20 percent and the power input by a total of 35 percent. This small adjustment saves up to 10 percent in energy — continuously and without any additional operating costs.

4.3 Purpose of Performance Test

The purpose of the performance test of pump is described below:

- To determine the effciency of pump under various operating conditions.
- To determine Specific Power Consumption.

After this we can compare the actual data with design data and access the pump performance and take necessary remedial actions.

4.4 Methodolgy Adopted in Audit

The pump efficiency can be determined by three key parameters: Flow, Head and Power. Of which flow measurement is most important as online flow meters are not available. The field testing of the pump can be done by four ways:

1. **Tracer method**
2. **Ultrasonic flow measurement**
3. **Tank filling method**
4. **Online Flowmeters**

4.5 Design Data

The following table provides the information for the Design Data of Circulating Cooling Water Pump.

Table 4.1: Design Data of Circulating Cooling Water Pump and Motor[6]

Particuars	**Units**	**Design**
Unit Load	MW	210
Feed Water Temperature	C	34
Power Consumption	kW	820
Head Developed	mWC	21
Shut Off Head	mtrs	35.8
Density	kg/m^3	0.99
Flow Rate	m^3/hr	10000
Speed	R.P.M	594
Voltage	KV	6.6
Full Load Current	Amp.	97.87

4.6 Performance Test Measured Data and Calculation Procdure

The data of performance test was taken for nine Cooling water pumps.

4.6.1 Calculation Procedure

(A) Hydraulic Power (Liquid Horse Power)

Hydraulic power of a pump is gives by:

$$P_h(kW) = Q \times (h_d - h_s) \times \rho \times \frac{g}{1000}$$

Where,

Q= Volume Flow rate (m^3/s)

ρ=Density of fluid(kg/m^3)

g= Acceleration due to gravity (m/s^2)

(h_d-h_s)= Total Head in meters

(B) Motor Input Power

Input Power can be calculated using Power Analyzer.

(C) Pump Shaft Power

It is calculated by multiplying the motor input power by motor efficiency.

P_s= P_m $\times$ η_{motor}

(D)Pump Efficiency

It is calculated by dividing the hydraulic power by pump shaft power.

η_{motor}= P_h/P_s

4.7 Exergy Analysis of Pump

The following part shows the methodology adopted for exergy analysis of pump.

The second law efficiency of a process is defined as the ration of the minmum available energy which must be consumed to do a task divided by the actual amount of exergy consumed in performing the same task.

$$\eta = \frac{Minimum\,energy\,consumed\,to\,perform\,given\,task}{Actual\,work\,supllied}$$

$$\eta = \frac{\dot{m}A}{W}$$

$$A = C_p \cdot (T_0 - T_i) - T_r(C_p{\cdot}ln\frac{T_O}{T_i} - R{\cdot}ln\frac{P_O}{P_i})$$

Where,
$\dot{m}$ =Mass flow rate(kg/s)
A =Availabilty or Exergy(kJ/kg)

C_p =Specific heat is taken 4.186 kJ/kgK

R =Gas constant is taken 0.287 kJ/kgK

T_o =Outlet temperature(K)

T_i =Inlet temperature(K)

T_r =Reference temperature is taken 300 K

P_o =Outlet temperature(K)

P_i =Inlet temperature(K)

P_r =Reference pressure(bar)

W =Input power(kW)

4.7.1 Measured Data

Measured, theoretical and calculated data for nine Circulating Cooling Water Pumps is mentioned in Appendix E.

4.7.2 Summery Sheets of Circulating Cooling Water Pump

Overall efficiency of nine cooling water pumps along with base line corresponding to design data is shown in figure 4.1

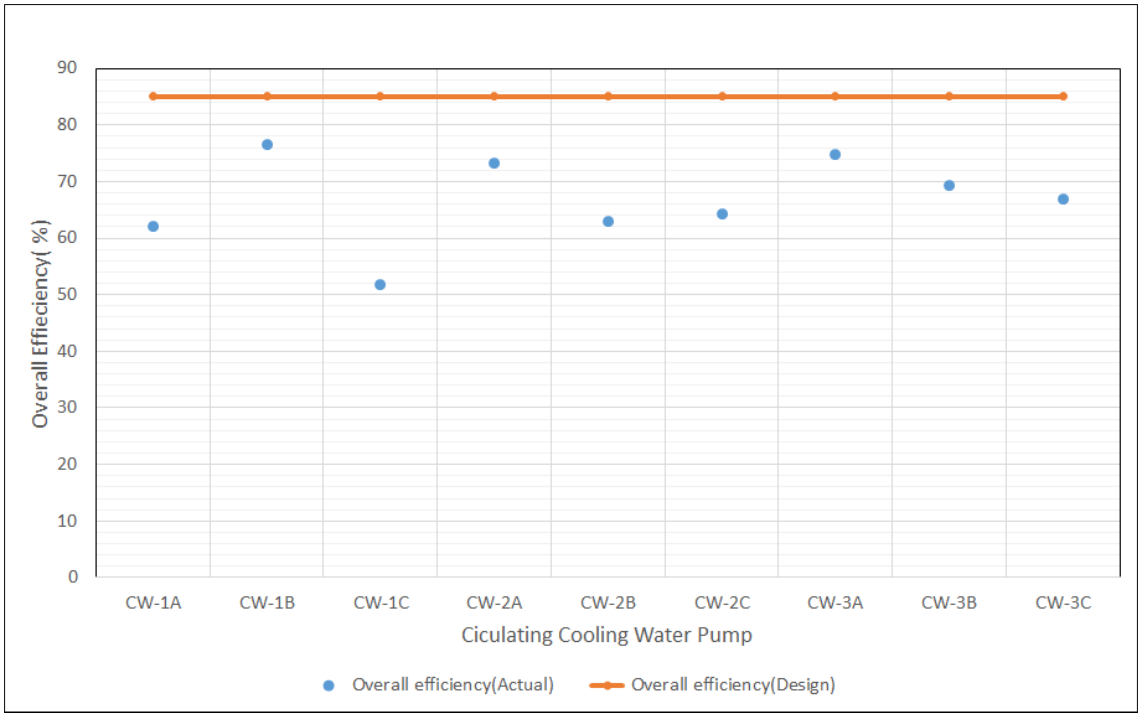

Figure 4.1: Overall Effieciency of Circulating Cooling Water Pumps

Table 4.2: Categorization of Pump Based on Perfromance

Pump Performance Category	**Efficiency Range(%)**	**Pump**
Normal	75 to 85	1B
OK	65 to 75	2A,3A,3B,3C
Poor	50 to 65	1A,1C,2B,2C

Pump performance is catagorized as normal, ok or poor based on efficiency. A table 4.2 gives details about this.

Discussion:

As we can see from figure 4.1 that overall efficiency is less for all pumps with compare to design efficiency.

The efficiency of Pump 1C is on lower side. This is due to lower flow delivered.

The reasons for lower flow delivery and correspondingly, high power consumed by the motor are mentioned below.

- Impeller or bowl partially plugged
- Leaking joints
- Strainer partially clogged
- Worn pump bearings
- Heavy lubricating oil
- Cavitation

Specific energy consumption of nine cooling water pumps along with base line corresponding to design data is shown in figure 4.2.

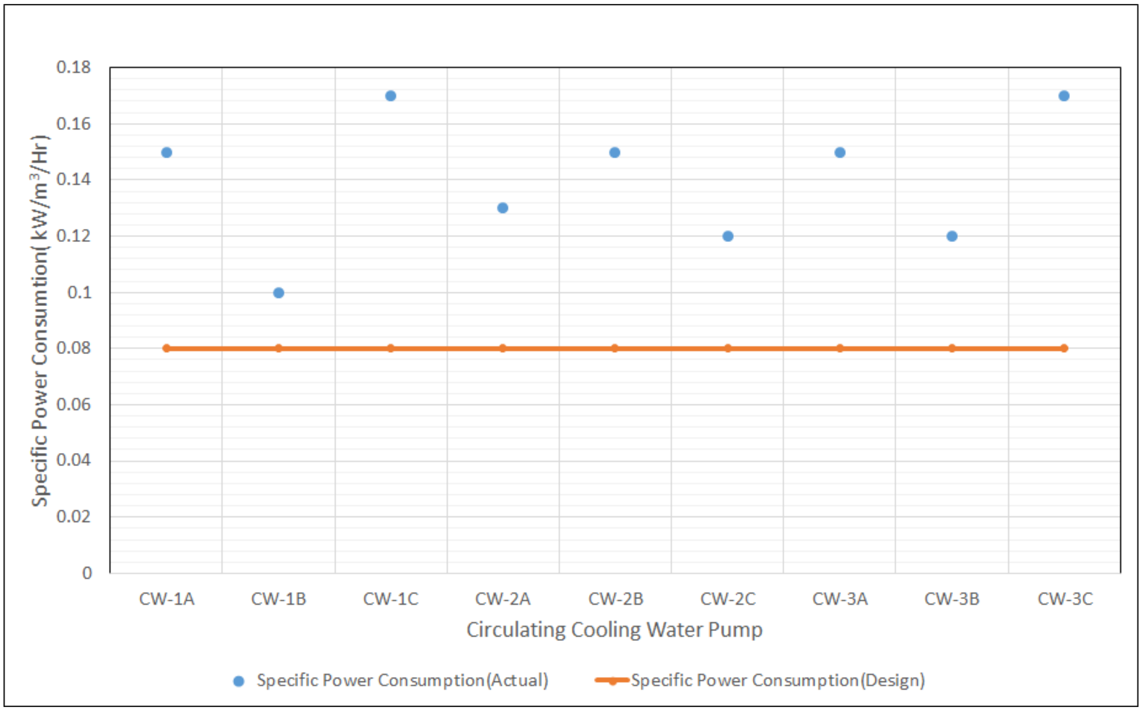

Figure 4.2: Specific Power Consumption of Circulating Cooling Water Pump

Table 4.3: Specific Power Consumption of Pumps

Pump	**Specific Power Consumption**
1A,1B,1C,2A,2B,2C,3A,3B,3C	Higher

Suggestion

The pump is having low loading on flow, which may be due to the following reasons:

- Impeller or Bowl partially plugged.
- Leaking joint
- Worn pump bearings
- Heavy lubricating oil
- Cavitation

Hence it is recommended to check impeller of the pump – 1C, 1A ,2B and 2C from wear and tear and other mechanical problem and rectify the same by doing the maintenance if possible to rectify.

4.7.3 Exergy Efficiency in Circulating Cooling Water Pumps

The calculated parameters for exergy analysis of nine circulating cooling water pumps is shown in table 4.4.

Table 4.4: Exergy Efficiency of Circulating Cooling Water Pumps

Parameters	Unit	CW-1A	CW-1B	CW-1C	CW-2A	CW-2B	CW-2C	CW-3A	CW-3B	CW-3C
Mass Flow Rate	kg/s	2034	3145	1939	2328	2024	2984	2292	2556	2095
Suction	bar	0.6372	0.6372	0.6372	0.6372	0.6372	0.6372	0.6372	0.6372	0.6372
Discharge	bar	1.618	1.569	2.180	1.480	1.430	1.667	1.147	1.569	1.667
Temperature Inlet	K	306	306	306	306	306	306	306	306	306
Temperature Outlet	K	309	310	308	309	309	309	309	309	309
Input Power	kW	1160	1170	1190	1133	1144	1290	1273	1144	1282
Exergy In	MW	771.94	801.24	752.88	737.99	745.15	812.94	719.63	766.11	777.60
Exergy Destruction	MW	388.05	368.76	437.12	395.00	398.85	477.06	553.37	377.88	504.40
Exergy Effieciency	%	66.55	68.49	63.27	65.13	65.13	63.10	56.53	66.96	60.65

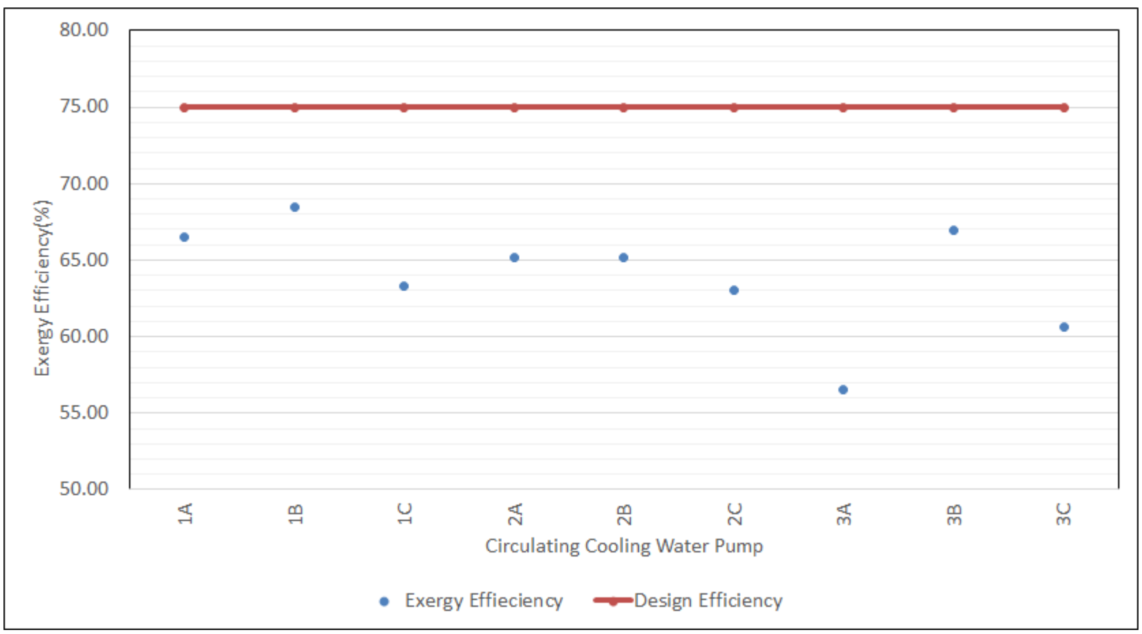

Figure 4.3: Exergy Efficiency

Exergy efficiency of circulating cooling water pumps are less than design value. Pumps 1C,2C,3A and 3C have very low exergy efficiency. The factors that affects the efficiency are high specific power consumption, low flow rate, different suction and discharge head, load fluctuation on pump etc.

4.7.4 Exergy Available in Circulating Cooling Water Pumps

Exergy available in circulating cooling water pumps are shown in following figure 4.4 .

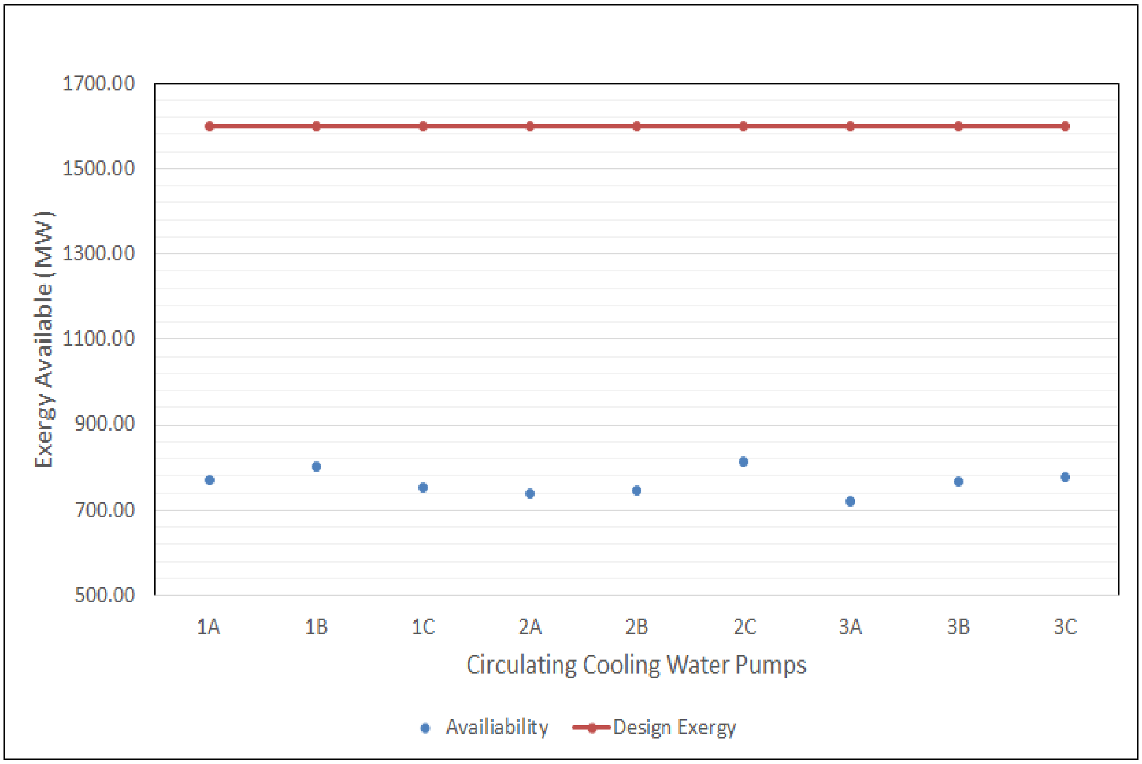

Figure 4.4: Exergy Available in Circulating Cooling Water Pumps

As shown in figure 4.4 exergy available in all pumps is very less than design value which directly affects on efficiency of pump.

4.7.5 Exergy Destruction in Circulating Cooling Water Pumps

Exergy destruction in circulating cooling water pumps are shown in following figure 4.5 .

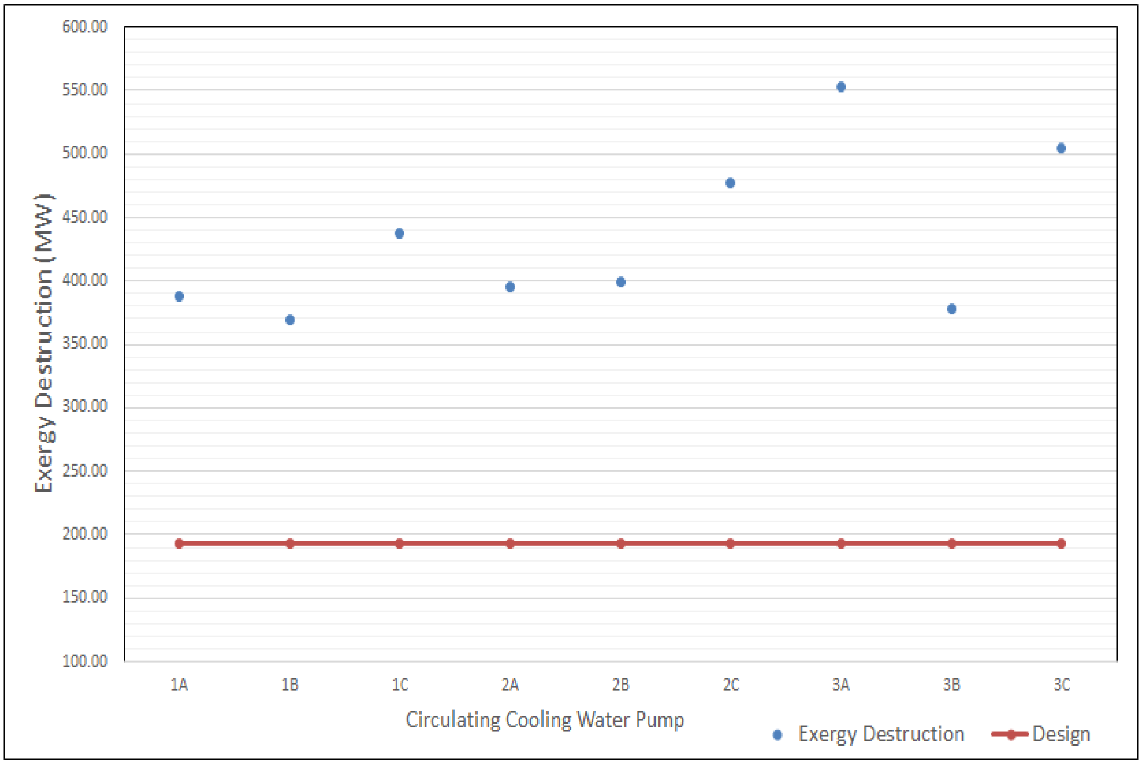

Figure 4.5: Exergy Destruction in Circulating Cooling Water Pumps

As shown in figure 4.5 exergy destruction in pumps are very high than the design value of exergy destruction.

Chapter 5

Energy Audit and Exergy Analysis of Condenser

5.1 Introduction

The purpose of the condenser is to condense the exhaust steam from the turbine so that it can be returned to the system for reuse. In the rankine cycle the condenser is complementary to the boiler in that it condenses the steam while the boiler boils the water. Like the boiler it has a free water surface that interfaces with the steam and some form of level control is required. Steam leaving the turbine enters at the top of the condenser and circulates around the outside of tubes where it is condensed by cooling water passing through the tubes. The resulting condensate rains down to collect in a hot-well at the bottom of the condenser.

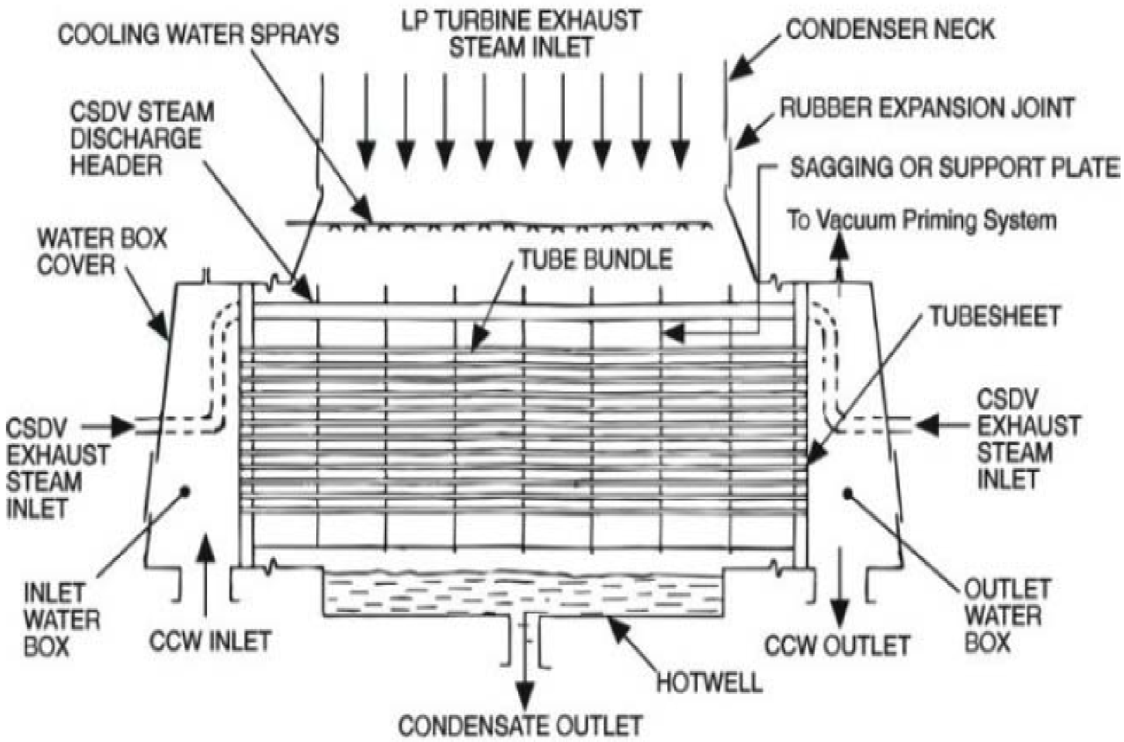

Figure 5.1: Condenser Longitudinal Section

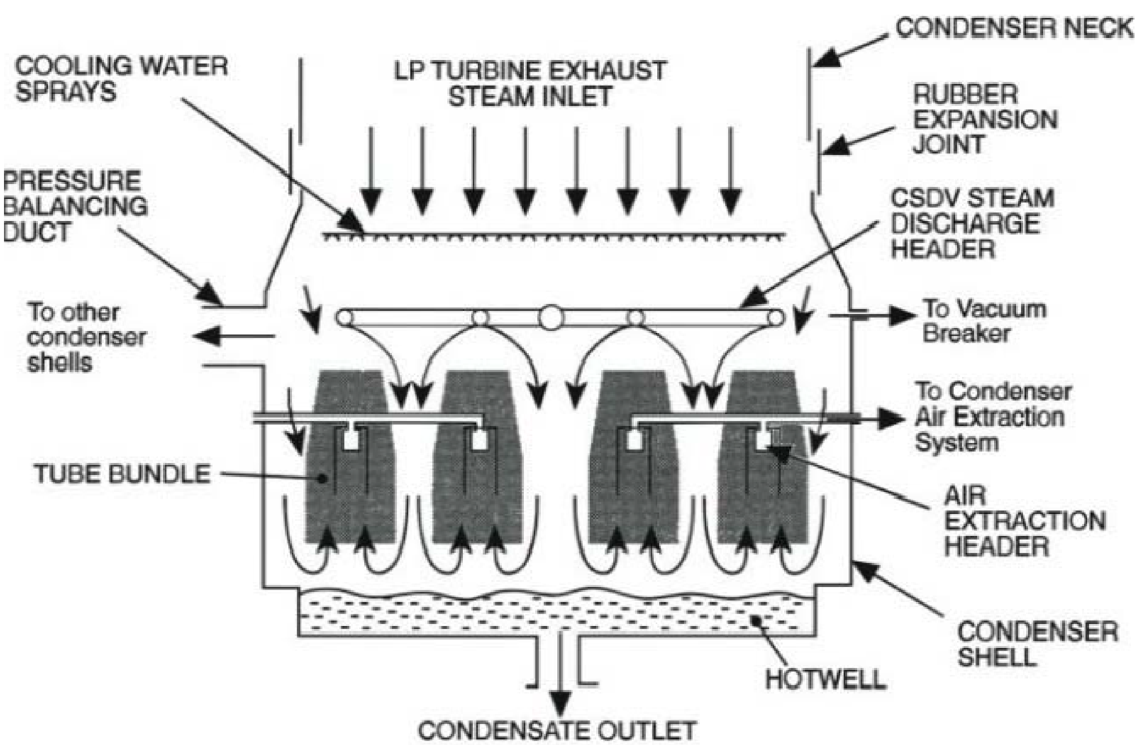

Figure 5.2: Condenser Cross Section

Figure 5.1 and 5.2 shows the typical arrangement of large condenser.

5.2 Literature Survey

The conventional steam power plant working under the Rankine Cycle and the steam condenser as a heat sink and the steam boiler as a heat source have the same importance for the power plant operating process. Energy efficiency of the coal fired power plant strongly depends on its turbine-condenser system operation mode. Operating the condenser at optimum circulation water flow rate is essentially important to ensure maximum efficiency and minimum operating cost of the plant. To control the condenser variables like cooling water flow rate, condenser pressure, condenser temperature having vital importance on entire plant performance. For the given thermal power plant configuration, cooling water temperature or/and flow rate change generate alterations in the condenser pressure. Those changes have great influence on the energy efficiency of the plant. The effect of the cooling water temperature and flow rate on the condenser performance, and thus on the specific heat rate of the coal fired plant and its energy efficiency.

The need for electrical energy will certainly continue to grow, and it has become imperative to lower the cost of electricity and enhance the operational economy of the turbine unit. Heat losses from the steam power plant cycle are mostly due to heat rejection through the condenser. Operating condenser at optimum variables is essentially important to ensure maximum efficiency and minimum input of the plant. The condenser operating conditions are of the great influence on the maximum generated power and the heat rate value. In the same time, the operating conditions of the cooling water system determine the operating conditions of the condenser. For cooling its condenser, steam power plants use basically two types of cooling systems: open-cycle and closed cycle. A closed-cycle cooling system transfers waste heat from circulating water to air drawn through cooling towers. Conventional wet cooling towers depend on evaporating heat exchange and require a continuous source of fresh water to replace evaporation losses. The ability of cooling towers to provide cold water to steam condensers of a thermoelectric unit decreases with increasing air temperatures and, in case of wet cooling tower, increasing humidity.

A.N. Anozie et al.[33] had worked upon surface condenser cooling water flow rate and by optimizing it, decreased surface area of condenser for same heat duty. The computer program codes had been developed for simulation of power plant by varying cooling water mass flow rate. Decreased flow by optimization, he proved that condensate sub-cooling can be prevented and the enthalpy of feed water can be increased. Higher enthalpy FW means less fuel consumption in steam generator and finally fuel cost reduces.

Mirjana, S. lakovic.et al.[16] focused on the influence of the cooling water temperature and flow rate on the condenser performance, and thus on the specific heat rate of the coal fired plant and its energy efficiency. As cooling water flow decreases the vacuum inside the condenser increases and heat rate reduces. This increased condenser pressure will decrease the power output of LP turbine which is not advisable. From mathematical model and analysis author proved that energy efficiency of condenser reduces when cooling water inlet temperature increases.

5.3 Purpose of Performance Test

The purpose of the performance test of condenser is described below:

- To determine the performance of condenser under various operating conditions.
- To determine effect of cooling water temperature and flow rate on performance of condenser.
- To analyze performance of condenser under different pressure condition.

After this we can compare the actual data with design data and access the condenser performance and take necessary remedial actions.

5.4 Methodology Adopted in Audit

The steps involved in conducting energy audit of condenser are:

- Data collection
- Observations and analysis
- Findings for energy conservation measures

5.4.1 Performance parameters

1) Condenser heat load $= Q * T * C_p$

where,

Q=Water flow rate(kg/hr)

T=Average CW temperature rise (^{0}C)

Cp= Specific heat (kcal/kg ^{0}C)

Expected condenser vacuum can be calculated from the performance curves of condenser as given by the manufacturer.

2) Calculated condenser vacuum

Calculated condenser vacuum = Atmospheric pressure - Condenser back pressure

3) Deviation in condenser vacuum

Deviation in vacuum = Expected condenser vacuum - Measured condenser vacuum

4) Condeser TTD

Condenser TTD = Saturation temperature - cooling water outlet temperature

5) Condenser Effectiveness

Condenser Effectiveness $= \frac{Rise\ in\ cooling\ water\ temperature}{Saturation\ temperature - cooling\ water\ inlet\ temperature}$

5.5 Design Data

The performance of condenser can be evaluated by measuring parameters and comparing the same with design. The able 5.1 lists the design parameters of condenser.

Table 5.1: Design Parameters of Condenser

Description	Units	Value
Type	Twin cell design	
Number of Passes	No	2
Heat Load	M kcal/hr	285.88
Cooling Water Flow Rate	m^3/hr	25750
Pressure at Turbine Exhaust	mm of Hg(a)	86
Tube Lenght between Tube Plates	m	10
Total Number of Tubes	No	15620
Cooling Surface Area	m^2	17350
Cleanliness Factor	%	85
Pressure at Tube and Water Box	kg/cm^2(a)	6
Cooling Water Temperature Raise	^{0}C	9
TTD	^{0}C	3.4
Saturation Temperature	^{0}C	45.4
Saturation and CW Inlet Temperature Difference	^{0}C	12.4
Condenser Effectiveness Factor		0.72
Log Mean Temperature Difference	^{0}C	6.95
DP across condenser	mmWC	3250

5.6 Calculated Parameters

The following table show the calculated parameters of condenser performance.

Table 5.2: Calculated Parameters of Condenser Performance

Parameters	**Unit**	**Unit 1**	**Unit 2**	**Unit 3**	**Unit 4**	**Unit 5**	**Unit 6**
Saturation Temp of Steam.	^{0}C	45.5	45.0	44.0	48.0	47.8	45.6
Temperature Diff. in Cooling Water	^{0}C	7.0	7.6	7.8	8.0	7.0	7.9
CW Outlet Temperature	^{0}C	43.0	37.6	39.3	44.5	41.4	39.0
CW Inlet Temperature	^{0}C	36.0	30.0	31.5	36.5	34.0	31.1
Atmospheric Pressure	mmHg	762.0	762.0	762.0	762.0	762.0	762.0
Condenser Back Pressure	mmHg	86.0	86.0	86.0	86.0	86.0	86.0
Expected Condenser Vacuum	mmHg	676.0	676.0	676.0	676.0	676.0	676.0
Measured Condenser Vacuum	mmHg	698.0	695.0	700.0	695.0	690.0	698.0
Deviation in Vacuum	mmHg	22.0	19.0	24.0	19.0	14.0	22.0
TTD	^{0}C	2.5	7.4	4.7	3.5	6.4	6.6
Effectiveness	%	73.68	50.67	62.40	69.57	52.24	54.48

5.7 Calculation of 2^{nd}Law Effieciency

Basically 2nd law efficiency is related with entropy term. This will give better idea about the performance of components than 1st law efficiency. 2nd law efficiency depends on exergy term. According to this law:

$$\eta_{II} = \frac{Exergy\,recovered\,(Available\;Exergy\;after\;process)}{Exergy\;supplied(Available\;Exergy\;at\;beginning)}$$

$$= 1 - \frac{Exergy\;destroyed}{Exergy\;supplied}$$

To find 2nd law efficiency exergy destruction is to be calculated. $Exergy\;destruction = m * T_0 * S_{gen}$

5.7.1 Exergy Analysis

Total Exergy is given by $Exergy = \dot{m}[(h_1 - h_0) - T_0(S_1 - S_0)]$

Where ṁ is the mass flow rate.

h_0,$S_0$and T_0represents the reference state point (standard environmental condition)

Exergy at the inlet = $EX_{in} = \dot{m}[(h_1 - h_0) - T_0(S_1 - S_0)]$

Exergy at outlet = $EX_{out} = \dot{m}[(h_2 - h_0) - T_0(S_2 - S_0)]$

Exergy Destruction (I) = $I = EX_{in} - EX_{out} - W$

Table 5.3: Exergy Values for Condenser

Parameters	Units	Unit 1	Unit 2	Unit 3	Unit 4	Unit 5	Unit 6
Mass Flow Rate of Steam	kg/s	148.6	147.2	147.7	148.3	149.4	150.0
Latent Heat of Evaporation	kJ/kg	2393.5	2394.8	2397.6	2386.4	2387.0	2393.8
Steam Temperature Inlet	^{0}C	55.0	54.5	53.5	54.5	54.5	54.7
Steam Temperature Outlet	^{0}C	51.2	50.5	50.0	50.2	50.8	50.5
Mass Flow Rate of Cooling Water	kg/s	7408.0	7362.0	6514.0	7100.0	6737.0	6147.0
CW Inlet Temperature	^{0}C	36.0	30.0	31.5	36.5	34.4	31.1
CW Outlet Temperature	^{0}C	43.0	37.6	39.3	44.5	41.4	39.0
Q Steam	MW	355.70	352.56	354.31	353.98	356.72	359.07
Q Water	MW	217.06	234.21	212.68	237.76	197.40	203.27
Loss Factor		1.63	1.50	1.67	1.49	1.81	1.77
Exergy In	MW	10.00	6.40	5.21	9.44	7.07	5.65
Exergy Destruction	MW	9.82	6.24	5.04	9.18	6.90	5.45
Exergy Efficiency	%	56.55	38.84	31.20	36.17	41.88	27.72

5.8 Results and Discussion

5.8.1 First Law Efficiency of Condenser

Following figure 5.3 shows the first law efficiency of condenser.

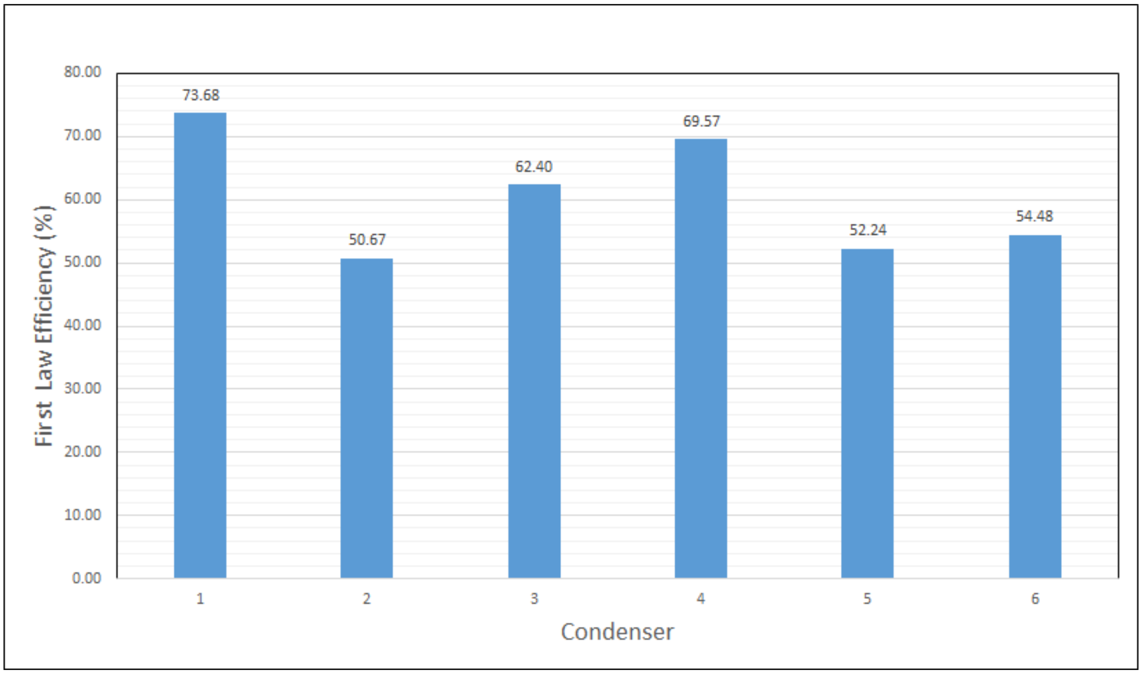

Figure 5.3: First Law Efficiency of Condenser

Condenser performance by first law efficiency are categorized in table 5.4.

Table 5.4: Categorized Condenser Efficiency

Condenser Performance	**Efficiency (%)**	**Condenser**
Good	> 65	1,4
OK	55 to 65	2
Normal	50 to 55	3,5,6

5.8.2 Exergy Destruction in Condenser

Following figure 5.4 shows the exergy destruction of condenser.

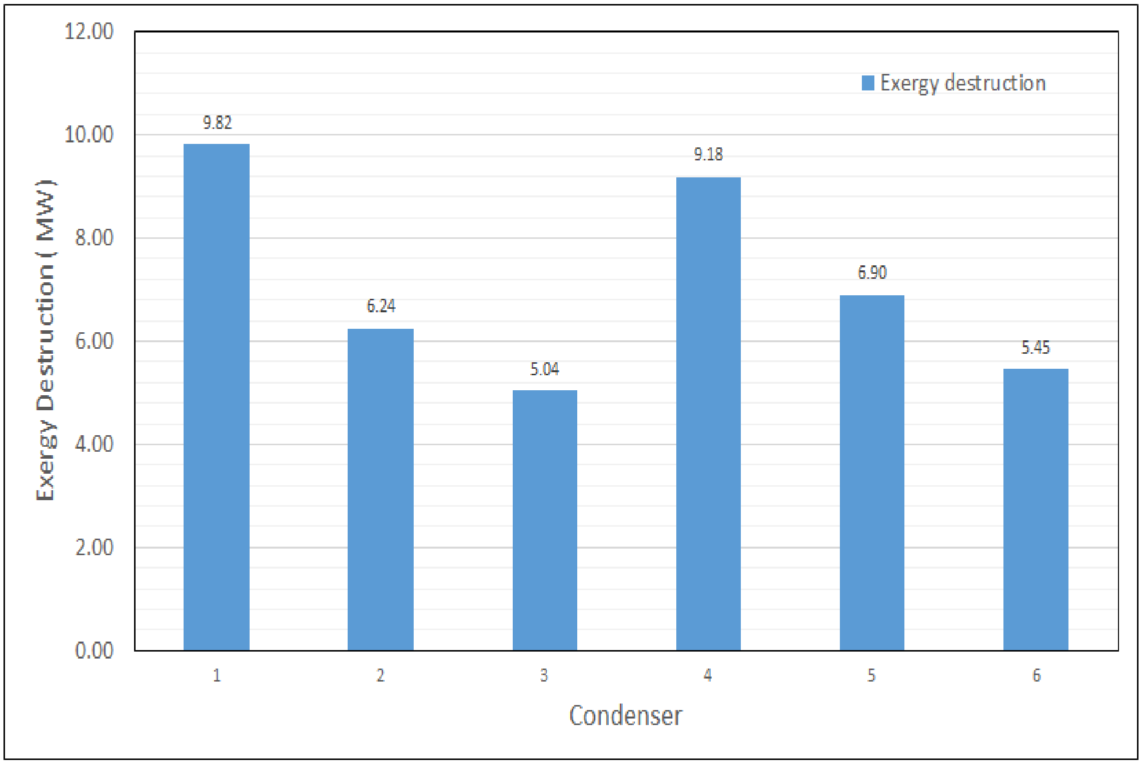

Figure 5.4: Exergy Destruction in Condenser

5.8.3 Second Law Efficiency of Condenser

Following figure 5.5 shows the second law efficiency of condenser.

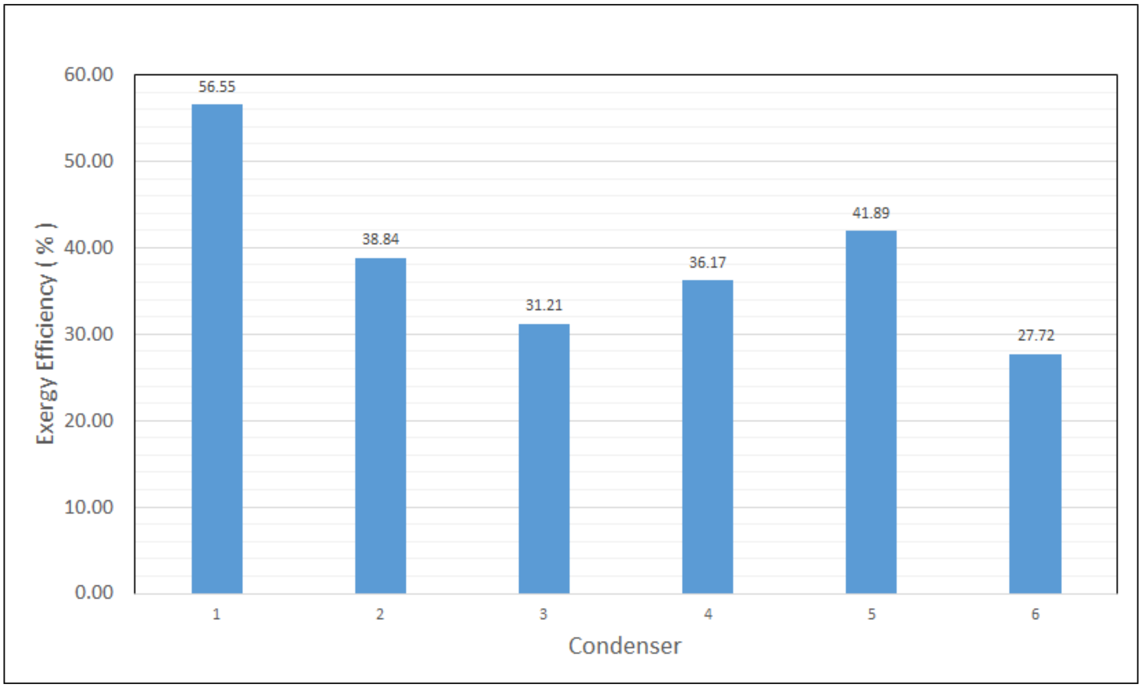

Figure 5.5: Second Law Efficiency of Condenser

5.8.4 Cooling Water Parameters Influence on Condenser Performance

Condenser heat ransfer rate strongly depends on condensing pressure, cooling water flow rate and temperature. In an ideal situation, when the venting system properly removes air from the steam condenser, the achievable condensing pressure is determined by temperature of the cooling water. For the steam power plant with closed cycle cooling system, cooling water temperature is determined by natural water source or ground temperature. This means that cooling water temperature is changing with weather conditions in particular region, and cannot be changed in order to achieve better condenser performances (i. e. higher vacuum in the condenser). Still, cooling water temperature directly affects condenser performances. Suitable parameter for on-line control is cooling water flow rate, and it can be varied in a wide range, with appropriate circulation cooling water pumps. During plant operation the objective is to operate at the optimum cooling water flow rate, which depends on cooling water temperature and power demand. In that manner, cooling water temperature and flow rate are considered variable parameters in the simulation of the plant operating conditions.

5.8.5 Condensing Pressure and Cooling Water Temperature

With cooling water temperature rise, the mean temperature difference in the condenser decreases, and condenser heat transfer rate for the same condensing pressure will also decrease.

It means that this particular condenser is designed at its maximum heat transfer rate and with increased cooling water temperature it cannot achieve required value. Increasing of cooling water flow rate will increase the condenser heat transfer rate for the given cooling water temperature.

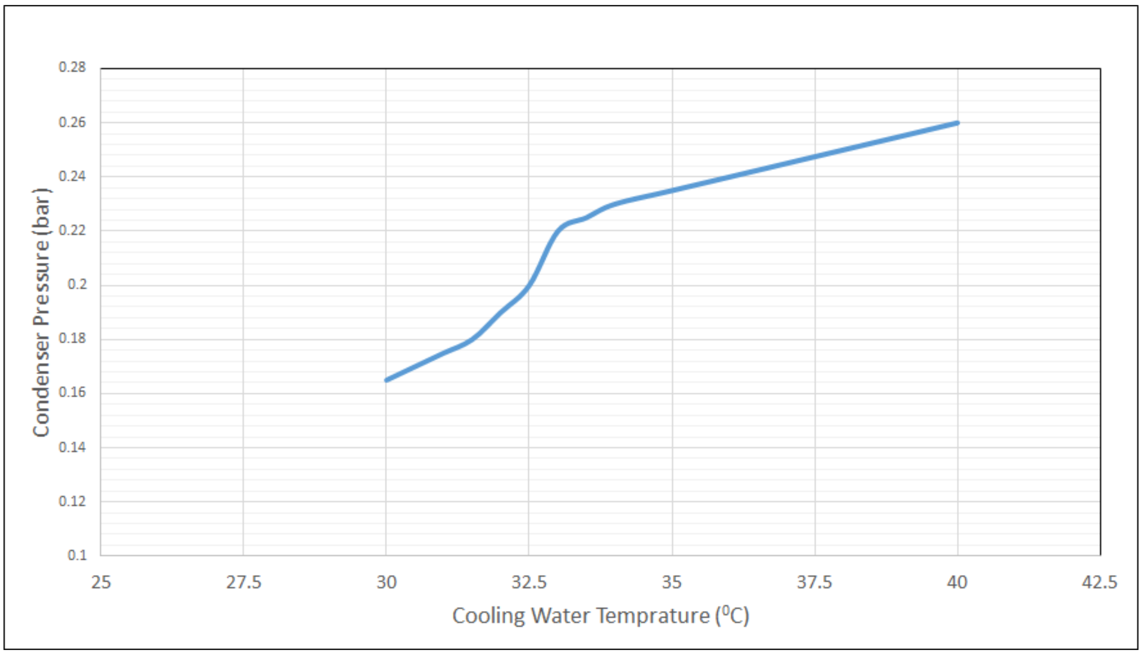

Figure 5.6: Variation of Condenser Pressure with Cooling Water Temperature

Condensing pressure dependence on cooling water temperature is obtained for the given water flow rate and steam load of the condenser. Steam load is considered constant, in order to obtain a clear illustration of this dependence, as it is shown in figure 5.6. It is obvious that with cooling water increasing, pressure in the condenser will also increase.

Steam Load of the Condenser With cooling water temperature increasing, in order to maintain designed heat transfer rate of the condenser, condensing pressure will increase. As this plant is working under turbine-follow mode, the turbine governor will increase the throttle and exhaust flows in order to set the generated power at the designed level, but with increasing of heat rate.

Condenser operating conditions influence on the plant performances

With increasing of condensing pressure and flow rate of cooling water,steam flow through the condenser, and thus through the low-pressure turbine is increasing.This will increase net power output, however, is correlated to the increasing of the heat rate.

Loss factor Loss factor in condenser is the ratio of heat loss by steam to heat gain by water. As shown in figure 5.7 loss factor is higher when condenser efficiency is lower and loss factor is lower when condenser efficiency higher.

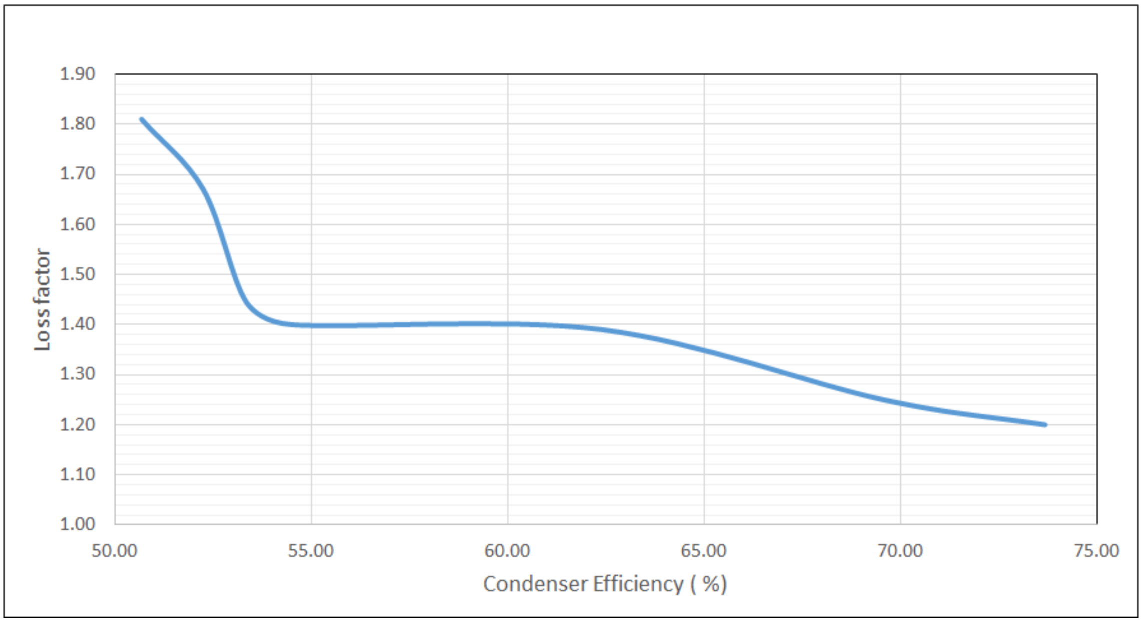

Figure 5.7: Loss Factor vs. Condenser Efficiency

Energy Efficiency and Specific heat rate

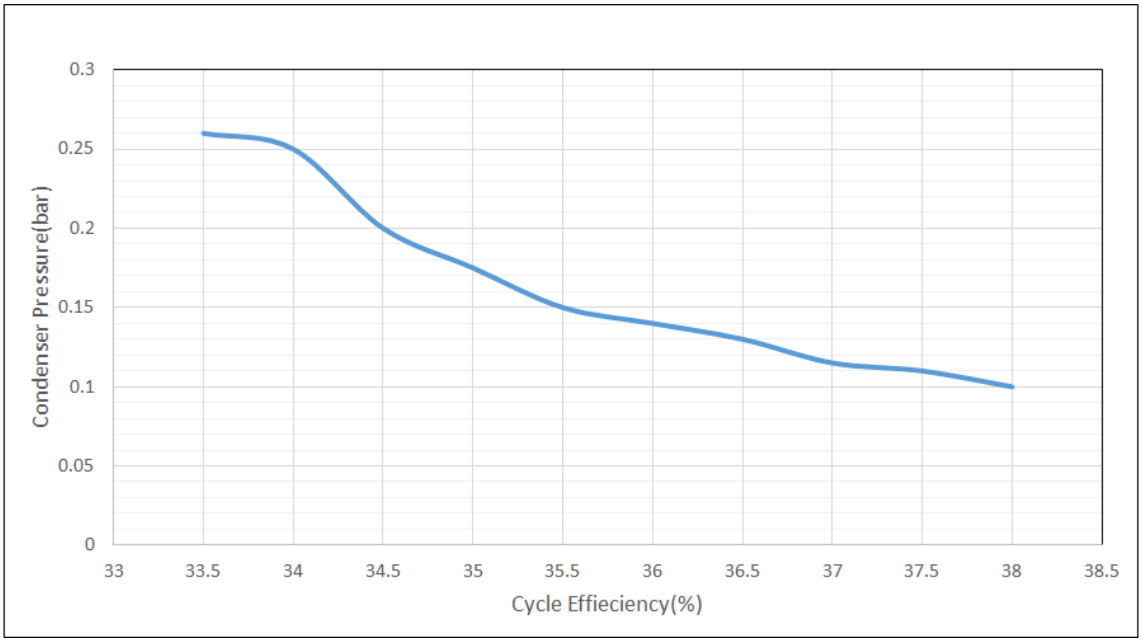

Figure 5.8: Variation of Cycle Efficiency with Condenser Pressure

Specific heat rate change due to condensing pressure change, With decreasing of the pressure in the condenser,specific heat rate decreases. With pressure decreasing to bellow the point where the exhaust annulus becomes choked, excessive condensate sub-cooling will result, tending to reduce the improvement in heat rate resulting from the lower condensing pressure. Condensate sub-cooling is defined as the saturation temperature corresponding to the pressure in the condenser minus the condensate temperature in the hot well.

With increasing pressure in the condenser of 0.11bar, efficiency decreases to 2.0-2.7% and considering that in this particular case the reduction is 2.7%,dependece of the energy efficiency in the function of the cooling water temperature rise is obtained and is shown in figure 5.8 . Cooling water temperature rise causes the reduction of energy efficiency and power of steam power plants. The problem is more severely set for plants with closed-cycle cooling system in comparison to the plant with an open cycle cooling system. Dry Bulb Temperature, the wet bulb temperature, the atmospheric pressure, flow rate of the circulating water. It means that cooling water temperature is changing in amuch wider range in one day in comparison to cooling system.

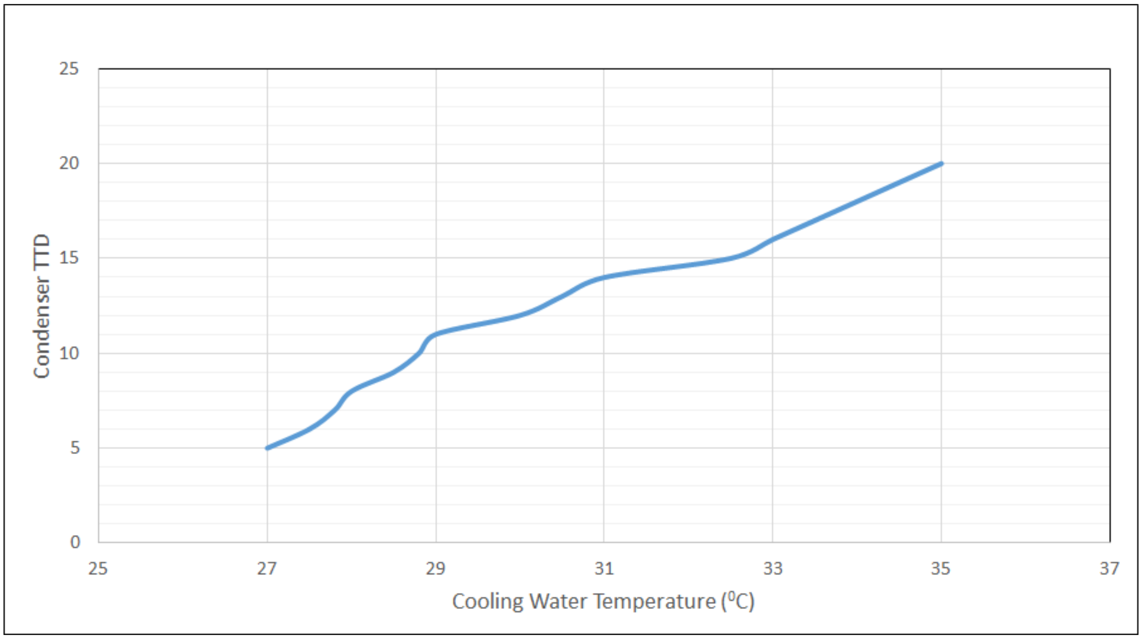

Figure 5.9: Variation of Condenser TTD with Cooling Water Temperature

Condenser TTD and cooling water temperature The analysis is show that the condenser TTD is function of water temperature, figure 5.9 show that the condenser TTD is increase with increasing the temperature of inlet cooling water in condenser and reduce the condenser performance.

Chapter 6

Plant Heat Rate

6.1 Introduction

The heat rate of a coal fired thermal power plant is a measure of how efficiently it converts the chemical energy contained in the coal into electrical energy. This conversion is accomplished in four major steps.[43]

- Chemical energy in the coal is converted into thermal energy.
- Thermal energy is converted into kinetic energy.
- Kinetic energy is converted in mechanical energy.
- Mechanical energy is converted to electrical energy.

In each of these processes,

- Some energy losts are occured,
- Some of the fuel is not burnt completely,
- Some of the thermal energy is rejected to the cooling water,
- Some of the kinetic and mechanical energy produces heat instead of electricity,
- Some of the electricity that is produced is used by auxiliaries in plant.

The heat rate of a power plant is the amount of chemical energy that must be supplied to produce one unit of electrical energy. It is an important parameter for assessing the efficiency of a thermal power plant.

6.1.1 Why is Heat Rate Important?

High temperature rate and thermal execution change are indispensable parts of any genuine exertion for expense regulation in an electric generating station. As the electric power industry grows deregulation and rivalry, cost control and the capacity to give energy at the most minimal conceivable expense get to be imperative issues. The power producer must give a lower-cost energy than the opposition yet still stay productive in the long haul. Fuel-cost diminishment and expanded dependability and accessibility through proficiency change are key techniques for enhancing benefit.

Heat rate improvement requires the support of personnel at all levels, in addition to support from both station and corporate management. By monitoring and acting on many of the items identified here and decreasing the amount of controllable losses, plant heat rate can be improved to an optimum level and maintained at that optimum level.[43]

Controllable misfortunes, frequently called administrator controllable misfortunes, are characterized as "those heat rate misfortunes that can be specifically affected (either absolutely or contrarily) by the activities of the administrator." In numerous cases, the real "control" is taken care of by the control system, however regularly, administrator intercession can affect the size of the losses.

6.1.2 Factors Affecting on Heat Rate

LP steam turbines are designed to operate with specific values of turbine back pressure. The turbine back pressure increases above the design value as the steam temperature in the condenser increases above the design value, which results in a reduction in MW produced and leads to increases in turbine cycle and unit heat rates.For units equipped with cooling towers, factors such as condenser fouling, maintenance related cooling tower performance deterioration, and increases in ambient temperature and humidity can all cause increases in back pressure. For full-load operation, increases in turbine cycle heat rate of more than 2% are typical for an increase in exhaust pressure of 50.8 mm of Hg above design. It is not uncommon to find units operating with turbine back pressures approaching 127 mm ofHg, which results in even larger heat rate disadvantage.

The condenser suffers from a low vacuum mainly because of one or more of the following reasons[43]:

- Reduced Circulating Cooling Water flow rate.
- Increased Circulating Cooling Water Inlet temperature.

- Tube fouling.
- Tube flooding by condensate which has not been pumped further in the circuit by the condensate extraction pump.
- Accumulation of gases in the condensate atmosphere.
- Abnormally large thermal load on the condenser (i.e. over and above the normal condenser thermal load for a given unit output).

6.1.3 Heat Rate Improvement Opportunities

Heat rate improvement opportunities for existing units includes:

- Reductions in heat rate due to process optimization.
- More aggressive maintenance practice.
- Equipment design modifications.
- Boiler, turbine cycle and heat rejection system.
- The overall level of heat rate improvement which can be achieved varies with unit design, maintenance condition, operating conditions, and type of coal.

6.2 Methods for Heat Rate Evaluation

There are two methods for calculation of heat rate of a thermal power stations:

- Indirect Method
- Direct Method

6.2.1 Indirect Method

Indirect method of plant heat rate measurement is an quick method,which is short duration and not possible throughout year. Though it is very complex and loss based method of measurement.

6.2.2 Direct Method

Immediate technique for heat rate estimation is carried out where coal estimation is accessible. In this strategy, coal utilization information is arrived at the midpoint of for long length of time to accomplish at exact estimation of coal utilization.. This technique is utilized at very nearly all thermal power stations as standard practice. Working parametric amount, for example, general era, aggregate coal utilization, normal general caloric estimation of the coal, particular coal utilization have been gathered from Thermal Power Station powers on month to month premise. From that point, working station high temperature rate for every month is figured as under.

$$Heat\,rate = Specific\,coal\,consumption * G.C.V.of\ coal$$

Where,

$$Specific\,coal\,consumption = \frac{Total\,coal\,consumption\,in\,amount}{Gross\,generation\,in\,month}$$

Thus Operating Heat Rate is calculated and compared with respect to design plant heat rate and % deviation is calculated for evaluation of the performance of Thermal Power Station :

$$Heat\,Rate\,deviation = \frac{(Operating\,Heat\,Rate - Design\,Heat\,Rate)}{Design\,Heat\,Rate} * 100$$

6.3 Calculation for Plant Heat Rate by Direct Method

Plant heat rate evaluation process required the details which are described below:

- Total coal consumption of each unit during evaluation period.
- Average overall calorific value of coal.
- Gross generation of elecricity (in kWh).
- Gross Calorific Value.

Unit wise evaluation is shown in figure 6.1.

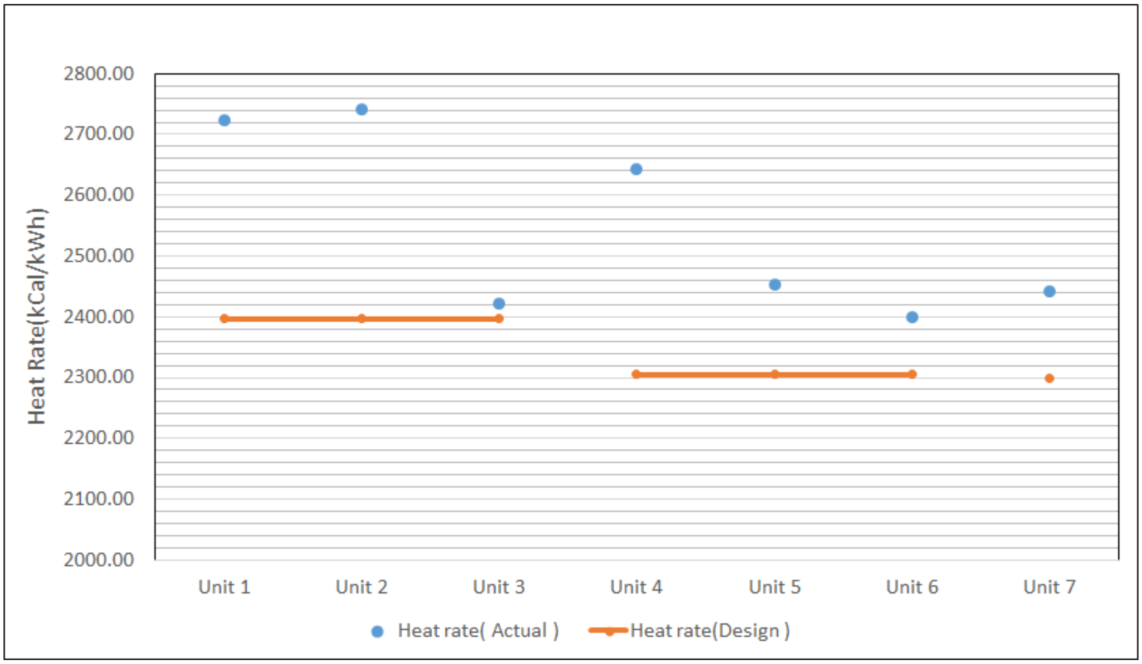

Figure 6.1: Unit wise Heat Rate Evaluation

As we can see from the figure 6.1 that the heat rate is higher than its design value in unit 1,2,4,5,7. It is slightly higher than its design value in unit 3 and 6.

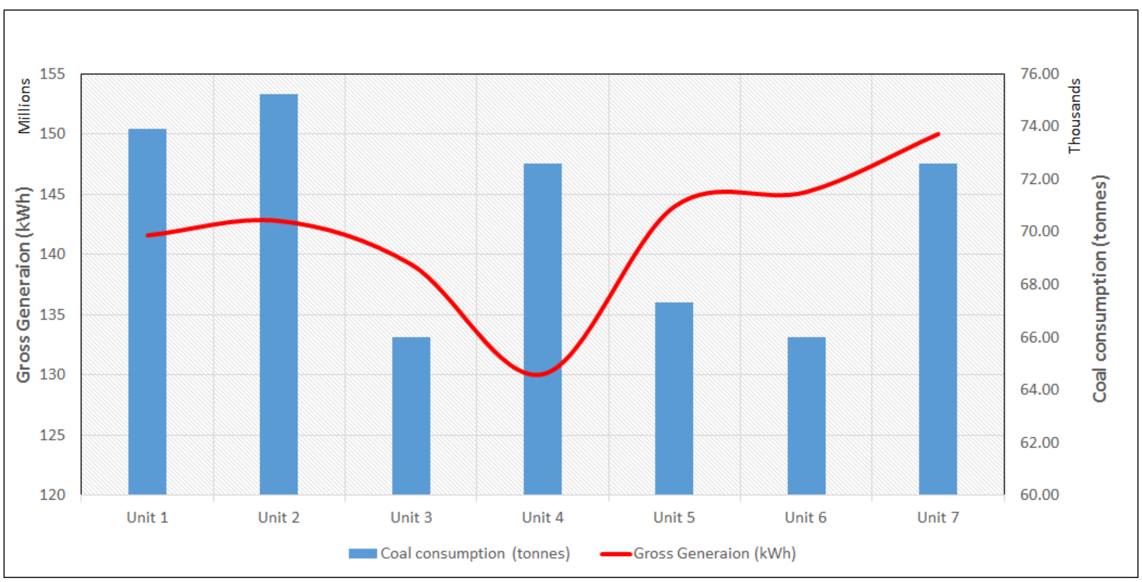

Figure 6.2: Unitwise Gross Generation and Total Coal Consumption

This figure 6.2 shows unitwise gross generation and total coal consumption. In that coal consumption in unit 3,5 and 6 are less while gross generation is not more affected as compared to other units.

6.4 Suggestion

The evaluated value of heat rate shown in appendix F indicates that power plant unit heat rate is more than design value. Poor performance of cooling tower result in increase of cooling water outlet temperature By re-duce the cooling water outlet temp. approximately 4.0 to 5.0 °C which will improve the condenser vacuum by 20-25 mmHg for each cooling tower and because of vacuum heat rate save approximately 50 to 60 kCal/kWh.

6.5 Discussion

From evaluated data for heat rate we can concludes that heat rate affecting parameter is condenser vacuum, condenser back pressure, high water inlet temperature at condenser inlet , not operating the equipments on design parameter, and poor cooling tower effectiveness.

Chapter 7

Conclusions and Future Work

7.1 Conclusions

7.1.1 Energy Audit and Exergy Analysis of Cooling Tower

Present operating effectiveness of NDCT number 4 & 5 are 57.14 and 59.32% respectively which are lower as compare to its design value of 66.67 %. As per design data range is 5 °C and aprroach is 10 °C but at actual condition range is 8°C and 7°C and approach is 14°C and 11.80°C respectively. Its also observed that cold water temperature (CWT) never cross the dry bulb temperature(DBT) toward approaching to wet bulb temp. This show that heat transfer taken place inside the cooling tower mainly due to sensible heat transfer, evaporative cooling (latent heat transfer) is not contributing in decreasing cooling water temperature. Using ultra ever dry coating helps to repel water on fill surfaces corrosion can be prevented caused by moisture. By replacing old spray nozzle with new cooling spray nozzle made of ABS/PVC we can improve water distribution in cooling tower which is currently non-uniform distribution.

7.1.2 Energy Audit and Exergy Analysis of Circulating Cooling Water Pump

Present audit of circulating cooling water pump concludes that overall effieciency of all nine pumps is lesser then design value. It is possible to improve pumps performance by trimming the impeller, replacing worned gear, precaution against worn out gear and bearings. These improvements makes it possible to improve efficiency of pumps and operation nearer to Best Efficiency Point. Second law efficiency is lower because of large variation in exergy destruction,this destruction is mainly because of varying pressure, friction and aging of pump etc.

7.1.3 Energy Audit and Exergy Analysis of Condenser

Energy audit of condenser concludes that first law efficiency of all condenser are lesser then design value of 74%. Performance of condenser 1 and 4 are satisfactory because thier efficiency are nearer to design efficiency,while condenser 2,3,5 and 6 are not working efficiently. Second law efficiency of all condenser is less than design efficiency of 60%. Exergy efficiency is lower than design because exergy destruction in condenser is more. Reasons for lower effieciency is lower vacuum and cooling water temperature difference than design value. Cooling water flow rate through condenser directly affects the efficiency. Higher cooling water inlet temperature in condenser leads to lower efficiency.

7.1.4 Heat Rate Analysis

By analysis we can conclude that the current heat rate of WTPS is more than design value which is not acceptable in terms of efficiency and economically as well as environmentally, Higher heat rate increases coal requirement increases for same generation of electricity . Improvement in the plant heat rate is possible by taking care of cooling water outlet temperature from cooling tower, condenser back pressure and vacuum. Further studies shall be focusing on improving plant heat rate by improving the effectiveness of cooling tower and efficiency of condenser without vacuum loss.

7.2 Scope of Future Work

Present work provides one sided data for comparison of natural draft cooling tower with forced draft cooling tower. Further methods to improve second law efficiency of cooling tower in the context of technological challenges can be investigated. Performance analysis of cooling tower based on location and climatic conditions shall carried and the cluster for the same can derived. Also thermodynamic analysis can be done in cooling tower by replacing nozzles for uniform water distribution to optimize the performance.

Present analysis of condenser is limited to condenser efficiency, further the relation between condenser pressure, generated power and condenser efficiency can be investigated. As heat transfer effectiveness of condenser is highly depended on tube fouling, the metallurgical challenges shall indentify and overcome. Also this analysis provides preliminary inputs for modeling and heat transfer analysis of condenser.

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Appendices

Appendix A : Stability of Test Condition During Summer Season for Cooling Tower

Sr. No	Measured	Design Value	Op. avg.	Max. Value	Min. Value	Max op. deviation	Code permissible
1	Wind Velocity	2.22	4.00	5.0	1	±2.0	5.0 Avg Max value
2	Wet Bulb Temp, °C	28.50	24	27	22	3.0	±5°C of design
3	Relative Humidity	50.00	49.62	58	41.60	8.38	Should not fall below
4	Dry Bulb Temp, °C	37.78	32.60	34.50	30.2	1.9	±5°C of design
5	Range, °C	10.00	7.3	NA	NA	2.7% from design	±20% of design
6	Heat Load	383.00	251	NA	NA	34% of design	±20% of design
7	Flow Rate	33000.00	29500	NA	NA	10.5% of design	±10% of the flow

Appendix B : Air Side Measurements for Cooling Tower

S.No	Cooling Tower Parameters	Unit	At Inlet Condition					
			Unit 1	**Unit 2**	**Unit 3**	**Unit 4**	**Unit 5**	**Unit 6**
1	DBT	°C	28.5	27	27.6	33.3	33.1	27.2
2	RH	%	76.5	38	28.67	40.22	412	31.56
3	WBT	%	25.1	17.3	15.9	22.50	22.6	16.2
4	Absolute Humidity	kg/kg of dry air	0.019	0.0085	0.006	0.013	0.0132	0.0071
5	Humid Volume of dry air	M^3/kg of dry air	0853	0.849	0.851	0.867	0.866	0.85
6	Density of dry air	kg/$M^3$of dry air	1.172	1.178	1.175	1.154	1.154	1.177
7	Enthalpy of Humid air	kJ/kg of dry air	77.33	48.96	44.73	66.96	67.27	45.59
8	Climatic Factor for Evaporation Loss	Factor	0.85	0.85	0.85	0.85	0.85	0.85

Appendix C : Water Side Measurements for Cooling Tower

S.No	Cooling Tower Parameters	Unit	Value					
			Unit 1	**Unit 2**	**Unit 3**	**Unit 4**	**Unit 5**	**Unit 6**
1	Circulating Water Flow Rate	kg/hr	27750000	36500000	23450000	29360000	24250000	22132000
2	Hot Water Temperature	°C	43.8	37.6	39.3	44.5	14.1	38.6
3	Cold Water Temperature	°C	35	29	29.8	36.5	34.4	28.9
4	Enthalpy of The Hot Water	kJ/kg of water	183.3	157.2	164.5	186.1	173.3	161.5
5	Enthalpy of The Cold Water	kJ/kg of water	146.5	121.4	124.5	152.9	143.9	121
6	COC in Makeup Water	%	1	1	1	1	1	1
7	Latent Heat of the Vapor at Avg Temperature	kJ/kg	2406	2406	2406	2406	2406	2406

Appendix D : Performance Parameters Evaluation of Cooling Tower

S.No	Cooling Tower Parameters	Unit	Unit 1	Unit 2	Unit 3	Unit 4	Unit 5	Unit 6
			Value	Value	Value	Value	Value	Value
1	Heat Load	kJ/hr	$102*10^7$	$95*10^7$	$93*10^7$	$98*10^7$	$71*10^7$	$89*10^7$
2	Range	°C	8.80	8.60	9.50	8.00	7.00	9.70
3	Approach	°C	9.90	11.70	13.90	14.00	11.80	12.70
4	Effectiveness	%	88.89	73.50	68.35	57.14	59.32	76.38

Appendix E : Result Table of Circulating Cooling Water Pumps

Stage 1

Particulars	Units	Unit 1			Unit2			Unit 3		
		CW-1A	**CW-1B**	**CW-1C**	**CW-2A**	**CW-2B**	**CW-2C**	**CW-3A**	**CW-3B**	**CW-3C**
Unit load	MW	210	210	210	200	200	200	160	160	160
Feed Water Temp	°C	32	32	32	32	32	32	32	32	32
Suction head	kg/cm^2	-5.1	-5.2	-5.2	-5.2	-5.1	-5.2	-4.8	-4.8	-4.5
Discharge pressure	kg/cm^2	1.6	1.7	1.64	1.55	1.45	1.56	1.6	1.5	1.42
Power consumption	kW	1160	1170	1190	1133	1144	1290	1273	1144	1282
Head developed	mWC	21.1	22.2	21.6	20.7	19.6	20.8	20.8	19.8	18.7
Density	kg/m^3	1000	1000	1000	1000	1000	1000	1000	1000	1000
Flow rate	m^3/hr	7321	11320	6980	8380	7284	10740	8250	9200	7540
Hydraulic power output	kW	421	684	411	472	389	608	467	496	384
Overall efficiency	%	62.2	76.5	51.9	73.2	62.9	64.2	74.8	69.3	67
Motor efficiency	%	84	85	83	85	82	83	84	86	84
Pump efficiency	%	66.6	81.4	55.2	77.9	66.9	68.3	79.5	73.7	71.3
Specific Power Consumption	kW/m^3/hr	0.15	0.10	0.17	0.13	0.15	0.12	0.15	0.12	0.17
% Loading of Motor	%	45.1584	60.79	52.52	43.86	40.54	62.97	42	49.26	38.50
% Loading of Pump on Flow	%	73.21	73.21	69.80	83.80	72.84	89.5	82.50	92.00	75.40
% Loading of Pump on Head	%	98.64	98.64	100.98	96.77	91.63	97.24	97.24	92.57	87.42

Appendix F : Plant Heat Rate Evaluation during October-2014

- Coal Consumption taken as per average consumption during one month on partial load 160 MW and full load 210 MW:

Parameter	Unit	Unit 1	Unit 2	Unit 3	Unit 4	Unit 5	Unit 6	Unit 7
Coal Consumption	tonnes	73920.00	75240.00	66000.00	72600.00	67320.00	66000.00	72600.00
Gross Generaion	MWh	141600.00	142800.00	139200.00	130080.00	144000.00	145200.00	150000.00
G.C.V	kCal/kWh	3686000.00	3645000.00	3670050.00	3640000.00	3640000.00	3640000.00	3640000.00
Heat rate	kCal/kWh	2724.00	2742.00	2422.00	2642.00	2453.00	2400.00	2442.00
Design Heat rate	kCal/kWh	2397.00	2397.00	2397.00	2305.00	2305.00	2305.00	2300.00
Deviation heat rate	%	13.64	14.39	1.04	14.62	6.42	4.12	6.17

Appendix G : Evaluation of Effectiveness of Condenser

- Calculated parameters for condenser at 210 and 160 MW:

Load(MW)	**MW**	**210**	**210**	**210**	**210**	**210**	**210**	**160**	**160**	**160**	**160**	**160**	**160**
Vacuum(mmHg)	mmHg	698	699	700	705	698	696	711	700	705	699	700	701
CW Inlet Pressure	kg/cm^2	1.75	1.72	1.71	1.72	1.62	1.68	1.60	1.50	1.51	1.51	1.62	1.68
CW Outlet Pressure	kg/cm^2	1.10	1.20	1.10	1.10	1.15	1.25	1.20	1.20	1.10	1.10	1.10	1.20
D.P.	kg/cm^2	0.65	0.52	0.61	0.62	0.47	0.43	0.40	0.30	0.41	0.41	0.52	0.48
CW Inlet Temperature	°C	27.0	27.0	27.0	27.0	27.0	27.0	27.0	27.0	27.0	27.0	27.0	27.0
CW Outlet Temperature	°C	35.5	35.5	36.0	35.5	36.5	36.0	35.5	35.5	35.8	35.8	35.5	36.0
Temp.Diff.	°C	8.5	8.0	9.0	8.5	9.5	9.0	8.5	8.5	8.8	8.8	8.5	9.0
Effectiveness	%	52	43	49	46	52	49	46	46	48	48	46	49