PERFORMANCE ENHANCEMENT OF BOILER USING ENERGY CONSERVATION PRACTICES

By

Rahul K. Shah

12MMET24

DEPARTMENT OF MECHANICAL ENGINEERING

INSTITUTE OF TECHNOLOGY

NIRMA UNIVERSITY

AHMEDABAD-382481

May 2014

PERFORMANCE ENHANCEMENT OF BOILER USING ENERGY CONSERVATION PRACTICES

Major Project Report

Submitted in partial fulfillment of the requirements

For the Degree of

Master of Technology in Mechanical Engineering (Thermal Engineering)

By

Rahul K. Shah

(12MMET24)

Guided By

Assi. Prof. N. K. Shah

DEPARTMENT OF MECHANICAL ENGINEERING

INSTITUTE OF TECHNOLOGY

NIRMA UNIVERSITY

AHMEDABAD-382481

May 2014

Declaration

This is to certify that

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

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

Rahul K. Shah

12MMET24

Undertaking for Originality of Work

I, **Rahul Shah**, Roll. No. 12MMET24, give undertaking that the Major Project entitled "**Performance Enhancement of Boiler using Energy Conservation Practices**" submitted by me, towards the partial fulfillment of the requirements for the Degree of Master of Technology in Mechanical Engineering (Thermal 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 disseratition work elsewhere: it will be result in severe disciplinary action.

Signature of Student

Date :
Place: Nirma University, Ahmedabad

(Signature of Guide) Endorsed By:

Certificate

This is certify that the Major Project entitled **" Performance Enhancement of Boiler using Energy Conservation Practices**" submitted by **Mr. Rahul K. Shah** (Roll no :12MMET24), towards the partial fulfillment of the requirements for the Degree of Master of Technology in mechanical engineering (Thermal Engineering) of Institute of Technology, Nirma University, Ahmedabad is the record of work carried out by his 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 my knowledge, haven't been submitted to any other university or institution for award of any degree.

Prof. N. K. Shah
Project Guide (Institute),
Assi. Professor,
Department of Mechanical Engineering,
Institute of Technology,
Nirma University,
Ahmedabad.

Dr.R.N.Patel
Professor and Head,
Department of Mechanical Engineering,
Institute of Technology,
Nirma University,
Ahmedabad.

Dr K Kotecha
Director,
Institute of Technology,
Nirma University,
Ahmedabad.

Acknowledgments

With immense pleasure, I would like to present this report on the dissertation work related to "**Performance Enhancement of Boiler using Energy Conservation Practices**". I am very thankful to all those who helped me for the successful completion of the dissertation and for providing valuable guidance throughout the project work. It is a pleasant aspect that I have now the opportunity to express my gratitude towards them.

I offer my sincere and heartfelt gratitude to **Mr. D.M.Jethva** (Industry Guide) EE, GSECL Wanakbori TPS for his valuable guidance, motivation and encouragement. My M.Tech dissertation work would not have been possible without his constant support and helpful discussions. His constant support and constructive criticism has been invaluable assets for my project work. He has shown keen interest in this dissertation work right from beginning and has been a great motivating factor in outlining the flow of my work. I would also like to thank **Mr. B.R.Patel**, DE, GSECL Wanakbori TPS, **Mr. Hitesh Nyalan**i JE, Wanakbori TPS and **Mr. Pratik Panchal**, DE, GSECL Wanakbori TPS, for providing continuous encouragement in making this dissertation work successful and for constant support and constructive criticism which has been invaluable assets for my project work.

I grab the opportunity to express my sense of gratitude and indebtness to **Shri. N.C.Patel**, Chief Engineer, GSECL Wanakbori TPS, for giving me the opportunity to carry out my dissertation work at Wanakbori Themal Power Station and for allowing me to be a part of that reputed research center.

I hereby take the opportunity to express my deep sense of gratitude to my respected institute guide **Prof.N. K. Shah**(Assi. Prof.), Mechanical Engineering Department, Institute of Technology, Nirma University, Ahmedabad for his valuable and precious guidance as well as active support during the project.

My sincere thanks and gratitude to **Dr.V.J.Lakhera**, P.G. Co-ordinator, Mechanical Engineering Department, Institute of Technology, Nirma University, Ahmedabad and **Dr.R. N. Patel**, Head of Mechanical Engineering Department, Institute of Technology, Nirma University, Ahmedabad for their continual kind words of encouragement and motivation throughout the Dissertation work.

I would also like to thank **Dr K Kotecha**, Director, Institute of Technology, Nirma University for allowing me to carry out my project work in industry. I am thankful to Nirma University for providing all kind of required resources.

Rahul K. Shah

Abstract

All the power plants are facing the problem of boiler performance deterioration. GSECL Wanakbori Thermal Power Station is also facing the same problem of boiler performance reduction frequently. The consequence of which affects the working of power plant and national income in general. The aim of this research work is to do energy analysis of 210 MW boiler of Wanakbori Thermal Power Station and find the reduction in energy performance of the boiler. For that the effect of waterside deposits on the energy performance of the boiler, performance gain after installation of burner tilting device, gain in boiler efficiency after correcting the fuel/air ratio and exergy analysis of power plant are considered in this analysis.

In analysis of effect on energy performance due to water side deposits in the waterwall of boiler, for simplicity the overall system is considered at 100% maximum continuous rating (MCR) and linear growth rate of deposition is assumed. The composition of water wall deposits is used for energy conservation analysis. The quantity of water wall deposits were found in the range of 36.8 - 55.5 mg/cm^2. The reduction in heat transfer in water wall system and efficiency of boiler were calculated. The boiler is rated as dirty and suggested for acid cleaning. Due to cleaning thermal efficiency is improved by approximately 1 %.

The analysis on performance gain after the installation of burner tilting device resulted in reduction in reheat spray. The reduction in spray is observed as 20.57 t/hr. Reduction in spray results in reduction of boiler heat supplied, increase of power output in HP turbine and loss of generator output in IP and LP turbine. The economical gain is observed around 350 lacs due to implementation of burner tilting device.

The appropriate position of air dampers were corrected to supply the optimum air/fuel ratio. The supply of optimum air/fuel ratio shows increase in boiler efficiency about 1%.

In exergy analysis of boiler, the exergy input such as exergy supply by fuel, water and air and exergy output such as exergy available with steam and flue gas were calculated to find out exergetic boiler efficiency. The exergy efficiency and irreversibility of all the boilers were found. The exergy efficiency were found between 34.45 % to 38.53 % and irreversibility found between 341.23 MW to 412.67 MW.

Keywords: Thermal power station, Waterside deposit, Burner tilting device, Waterwall, Reheat spray, HP cylinder, IP and LP cylinder, Air fuel ratio, Boiler efficiency, Air dampers, Exergy, Irreversibility

Contents

List of Figures

List of Tables

Nomenclature

C	Conversion factor
U	Overall heat transfer co efficient (W/m^2K)
f	fraction
AR	Area ratio
d	Diameter (mm)
h	Enthalpy(kJ/kg)
K	Thermal coductivity (W/m·K)
HR	Heat rate (kCal/KWH)
s	Entropy (kJ/kg·K)
T	Temperature (K)
m	mass flow rate (t/hr)
Q	Heat transfer rate
P	Pressure (N/m^2)
h	Enthalpy (kJ/kg)
Q_f	Feed water flow (t/hr)
Q_B	Bled steam flow (t/hr)
C_p	Specific heat
x_{ke}	Exegry of kinetic energy
x_{pe}	Exergy of physical energy
g	Gravitational acceleration
E_{pe}	Physical exergy
V	Velocity (m/s)
T_o	Reference Temperature (K)
I	Irreversibility
W	Work tansfer (kW)
z	Elevation of the system
e_x	Specific exergy (kJ/kg)
E_x	Exergy Rate (kW)

Greek Symbols

δ	Thickness(tube, deposits, etc..) (μm)
η	Efficiency(%)
λ	Thermal conductivity
μ	viscosity of medium ($\frac{kg}{ms}$)
ρ	Density of the fluid (kg/m^3)
ε	Exergy

Abbriviations

LMTD	Log mean temperature difference
fg	Flue gas
w/s	Water side
O	Clean condition
WTPS	Wanakbori Thermal power station
SH	Superheat
RH	Reheat
HP	High pressure
IP	Intermediate pressure
LP	Low pressure
in	Input
out	Output
MCR	Maximum continuos rating
BTD	Burner tilting device
i	inside
o	ou‘tside
f	Fuel
a	Air
CV	Calorific value
PA	Primary air
SA	Secondary air
AE	Available energy
UE	Unavailable energy
I	First law
II	Second law
w	water
max	Maximum
tm	Thermo - mechanical
ch	chemical

Chapter 1

Introduction

With increase in population and changing lifestyle, our dependency on energy sources has been increasing. The economic growth of the country generally depends on growth in the power generation capacity. Per capita energy consumption is a parameter for nation's growth. In India the per capita energy consumption has raised from 15.6 units in 1950 to 879.22 units in 2011-12 and it is expected to increase over 1000 units per annum in 2012 which is still below the per energy consumption of the developed countries[1].

To meet this target we are primarily depends on the coal based thermal power plant. As per available data, at present coal based power plant generates 57% of the country's total power generation. Most of these power plants, which have passed their operating/economical life, are less efficient and polluting. With increase in power generation the environmental conditions has also deteriorated. Global warming is becoming a major issue. These less efficient power plants are contributing a lot in global warming. Apart from this, because of these less efficient power plants the coal consumption increases. The coal resources are also limited.

The requirement of less fuel consumption, environmental emission and generation cost are obtained by increasing the efficiency of the power plant. For finding out the different reasons for power plant performance deterioration and its effect on the efficiency, the GSECL Wanakbori thermal power plants are being selected for the present study.

As we know boiler is a heart of the power plant, so its efficiency is directly affect the overall efficiency of the power plant. The major concern of this study is on boiler of the power plant. In present study, the different reasons for the performance deterioration and its effect on the boiler efficiency are found. The various methods for the solution of these problems has been discussed and after implementation of these methods the improvement in performance and efficiency is obtained.

1.1 Motivation

By cosidering the available data, the rate at which power demand increases is extremely very high compared to the rate at which generation capacity increase. To develop new generation capacity few years are required however by improving performance of present power plant certain increase in demand can be fulfilled with least time period.

Boiler is a heart of power plant and efficiency of it also affect the overall efficiency of the power plant. By obtaining better performance capacity of the boiler and the power generation capacity of the unit incrases and coal consumption also reduced. Due to this, it is possible to lower the fuel consumption, environmental emmision and generating cost.

1.2 Objective of Study

The different parameters which affect the performance and boiler efficiency are considered and from that, the parameters which have major effect on the efficiency of the power plant are included in the present study. The specific objectives of the work are the following :

- To analyze effect on the energy performance of boiler of the thermal power plant under fouled and cleaned conditions.
- To analyze improvement in performance of boiler after installation of burner tilting device.
- Optimization of fuel air ratio through SADC (Secondary Air Damper Control).
- Exergy analysis of WTPS boiler

1.3 Outline of Thesis

First chapter gives us basic information about the importance of energy saving in Boiler. The motivation and objective of this project also given in this chapter.

Second chapter includes the basic information about boiler. The working principle and efficiency evaluation techniques for boiler are also discussed.

Third chapter tells us about the effect of waterside deposits on the performace of boiler. The gain in boiler performance as removal of waterside deposites explained in this chapter.

In fourth chapter the gain in power plant after installation of burner tilting device is discussed. The effect of it on boiler efficiency, turbine efficiency, reduction in CO_2emmision and coal consumption is explianed.

Fifth chapter includes the role of excess air in combustion. The effect of correct air supplied on the combustion and so on the boiler efficiency included in this chapter. The measurement for the correct damper opening for gaining proper air supply is discussed in this chapter.

The final chapter includes the exergy anlaysis of boiler. The importance of exergy analysis and the techniques for power plant exergy calculation are explained. The exergetic efficiency and exergy destruction for all the boilers are claculated in this chapter.

Chapter 2

Literature Review

2.1 Boiler

Boiler is a closed vessel which use to transfer the heat produced by combustion into water untill it is converted into heated water or steam. The thermal energy comeout by combustion of fuel is transferred to water, so it vaporizes and obtain steam at the required pressure and temperature. In other words, Boiler is a closed vessel which convert water into steam using heat.

As per Indian boiler regulation(IBR), boiler is any closed vessel exceeding 22.75 liters in capacity and which is used for generating steam under pressure and includes any mounting or other fitting attached to such vessel, which is wholly or partly under pressure when the steam is shut off[2].

Heat is transferred from one body to another body by three ways : (1) Conduction - heat is transferred by the physical contact from one molecule to another (2) Convection - heat is transferred through conveying medium like air or water and (3) Radiation - heat is transferred from one body to other through temperature difference, no conveying required for this heat transffer.

The steam produced is used for :

- To get mechanical work by expanding steam in steam engine or turbine.
- For heating of the residential and industrial buildings.
- To perform various operations in indusries like sugar mills, chemical and textile industries.

Water is a useful and cheap medium for transferring heat to a process. When water get converted into steam its volume increases to1600 times which produces a force that is similar to that in explosive as gunpowder. Because of this, the boiler is a very dangerous equipment which must be used with maximum care[2].

2.2 Boiler Types

Boilers are classified into different types in many ways. The boilers are classified according to the application, operating pressure, fuel used, draught system adopted, boiler setting arrangement etc.

2.2.1 Based on Application

Under this category boilers are classified as a) Utility or Power boilers b) Industrial or process steam boilers c) Others

(a) Utility boilers Boilers meant for supplying steam to a steam turbine for electricity generation are known as utility boilers or power boilers. Generally the fuel used in these boilers is coal.

(b) Industrial boilers Boilers on industries like petroleum refineries, fertilizer plants, sugar plants, paper mills etc. to provide steam for the various processes in that industry comes under this category. They are also called as process steam boilers.

(c) Others Boilers used for locomotive steam engines, marine boilers etc. are comparatively smaller to power and industry boiler.

2.2.2 Based on Operating Pressure

Based on the pressure, at which steam is generated, the boilers are classified as a) Sub Critical boilers b) Super Critical boilers

(a)Sub Critical Boilers: Boilers operating below the critical pressure (225 kg/cm^2) are termed as sub critical boilers. These boilers can be either recirculation type or once through.

(b)Super Critical Boilers Any boiler operating above critical pressure (225 kg/cm^2) is called super critical boiler. These boilers are necessarily once through type.

2.2.3 Based on Fuel Used

According to the fuel used boilers are classified as a) Solid fuel fired boilers b) Oil fired boilers c) Gas fired boilers

2.2.4 Based on Draught system

In this category the boilers are classified as a) Natural Draught Boilers b) Mechanized Draught Boilers

(a)Natural Draught The draught required for the flow of air and gas inside the boilers is created by chimney alone in this type of boilers.

(b)Mechanized Draught In this type of boilers fans are used to create the draught.

2.2.5 Based on No. of passes

According to the number of paths through which the gases flow inside the boiler, the boilers are classified into a) Single Pass Boilers b) Multi Pass Boilers

(a)Single Pass Boiler :The boiler setting is in the form of a tower and all the pressure parts such as superheater, reheater and economizer are arranged as horizontal coils in a single pass of flue gas which is the vertical extension of the furnace. These boilers are also known as tower type boiler.

(b)Multi Pass Boilers: In these boilers the flue gas pass through more than one pass and the pressure parts are distributed in all the passes to recover the heat from the gas.

2.3 Specifications of Wanakbori TPS boiler

In Wanakbori thermal power station, there are 7 power generation units and each units have capacity of 210 MW. All of the power generation units are fabricated and erected by M/s Bharat Heavy Electricals(India) Limited. There is one boiler per unit. The boilers in Wanakbori thermal power station are regenerative reheat water tube boiler. The type of circulation in boiler is natural circulation. Coal is used as fuel and it is supplied to the furnace in pulverized form. The tangential firing system is used which is mostly common in pulverised coal fired boiler. The draught of the system is maintain through balanced draught arrangement. The furnace in the boiler is dry bottom furnace.

2.3.1 Water tube boiler

In water tube boilers, hot combustion gases are circulate around the outside of water filled tubes. Feed water from a lower drum is circulated through a bundle of tubes called tube bank. The hot products of combustion from the furnaces pass over this tube called tube bank and heat is transferred to water through the tube metal. Depending on the amount of heat added either all the water in the tubes will be evaporated into steam by the time it comes out of the tubes or part of water will be converted into steam with a water steam mixture coming out of the tubes. In the letter case the mixture from the tubes will be collected in a drum and after separating the steam the remaining water will be returned back to the inlet of the tubes.

Small water tube boilers, which have single or two burners, are mostly constructed and trnasported as enclosed units. Large water tube boilers generally are heavy in size and more in weight and because of that they are fabricated in small units and erected at the field. In water tube boilers, water flow inside tubes and the hot gases circulate outside these water carrying tubes. These boilers have single- or multiple-drum. These boilers can be fabricated for several steam capacity and pressures and higher in efficiency than fire tube boilers.

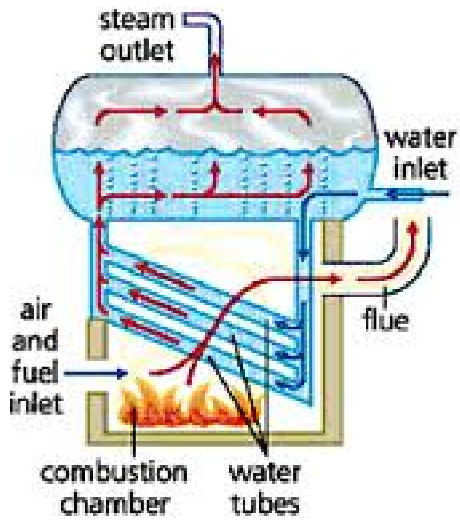

Figure 2.1: Water tube boiler[3]

2.3.2 Regenerative reheat cycle

All the modern steam turbine power plants are equipped with reheat and regenerative feed heating arrangement. In regenerative feed heating, the feed water supply to the boiler at a temperature much higher than the temperature of the condensate in the condenser by heating the feed water in one or more feed heaters with the help of steam bled off thereby increasing the average temperature heat addition in the cycle which in turn increases the thermal efficiency of the cycle. Reheating of steam is done by taking out the whole of the steam from the turbine at a suitable point (where the steam tends to become wet) and then it is reheated by further supply of heat which is given by the flue gas or live steam until it is super heated to near about the same initial temperature after which the steam is again supplied to the turbine and expanded to the condenser pressure. By reheating of steam erosion and corrosion are avoided.

2.3.3 Natural circulation system

Water delivered to a steam generator from feed heaters is at a temperature well below the saturation value corresponding to that pressure. Entering first, the economizer, it is heated to about 30 to 40 deg C below saturation temperature. From economizer the water enters the drum and thus joins the circulation system. Water entering the drum flows down through the down comer and enters ring header at the bottom. In the water walls a part of the water is converted to steam and the mixture flows back to the drum.

The circulation, in this case, takes place on the thermo-siphone principle. The down comers contain relatively cold water, whereas the riser tubes contain a steam water mixture, whose density is comparatively less. The density difference is driving force for the mixture. Circulation takes place at such a rate that the driving force and frictional force in water wall are balanced. As the pressure increases, the difference in density between water and steam reduces. Thus the hydrostatic head available will not be overcome the friction resistance for a flow corresponding to the minimum requirement of the minimum requirement of cooling of water wall tubes. Therefore natural circulation is limited to boiler with drum operating pressure around 175 kg/cm^2.

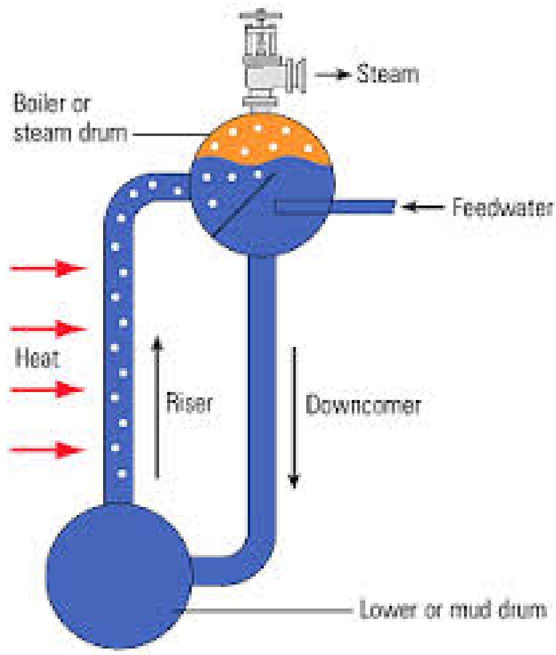

Figure 2.2: Natural circulation system[4]

2.3.4 Pulverized fuel

Most of the coal fired power plant boilers are using pulverized coal and number of large industrial boilers are pulverised fuel fired. This technology is well established and thousands of units around the world are using fuel in pulverised form.

In pulverized coal firing system, coal is first ground to dust like size and powdered coal is then carried in a stream of primary air to be fed through burners into the furnace of boiler. With the heating of coal particles in high temperature flames in the furnace, the volatile matter is distilled off resulting in the formation of minute sponge-like masses of fixed carbon and ash. After mixing of volatile gases with air, they get ignited and burn quickly releasing heat energy. [5]

2.3.5 Tangential fired

This type of firing system is widely used in pulverized coal fired boilers. It directs the mixture of fuel and air to an imaginary circle 1-2.5 m in diameter in the furnace centre. Both fuel and air are projected horizontally from the centre of the furnace along lines tangent to a vertical cylinder at the centre of the furnace. At the meeting points of two streams, intensive mixing occur which is desirable for complete combustion. A rotating motion, similar to that of a cyclone, is imparted to the flame body, which spreads out and fills the furnace area. The furnace employing tangential burner arrangement may be employed in a furnace having w/d ratio equal to 1 to 1.2. [5]

Figure 2.3: Tangential fired boiler[2]

2.3.6 Balance draught system

In this system both forced draught fan and induced draught fan are used. The forced draught fan is utilized to draw the controlled quality of air from atmosphere and force the same into the furnace. The induced draught fan sucks the products of combustion from the furnace and discharge into the chimney. The point where the draught is zero is known as the balancing point. This balancing point is normally maintained at the tip of the burners/nozzles where the air enters the furnace. Thereby a very small negative pressure is maintained at the furnace and other downstream paths of the boiler to prevent the hot flame/gas coming out of the boiler.

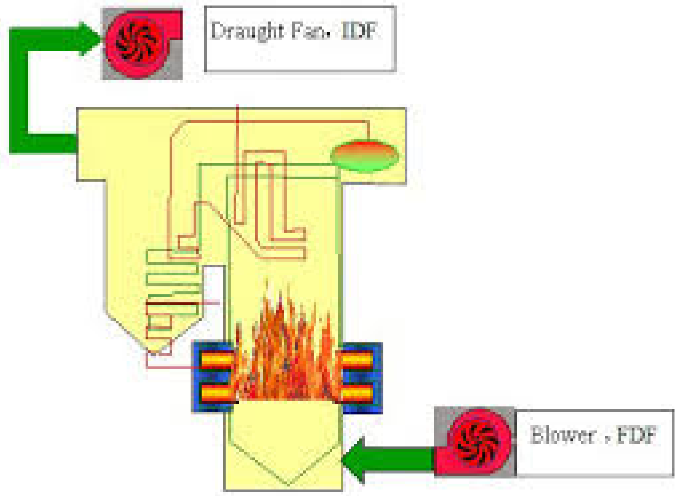

Figure 2.4: Balanced draught boiler

2.3.7 Dry bottom furnace

In dry bottom furnace ash or slag removed in the dry state. This is possible by allowing the exit temperature of gases below the ash fusion temperature. About 85% of ash, called fly ash flows with the furnace and get collected in electrostatic precipitator (ESP) while the remaining 15% falls through the furnace bottom to the clinker grinder and then the hydraulic sluice to which also falls the ash collected in dust collector and ESP. A hopper is provided at the furnace whose walls are inclined (50° to 60°) to allow gravity fall. The dry bottom furnace is cooler to cool molten slag and the solid slag falls to a slag pit containing water.

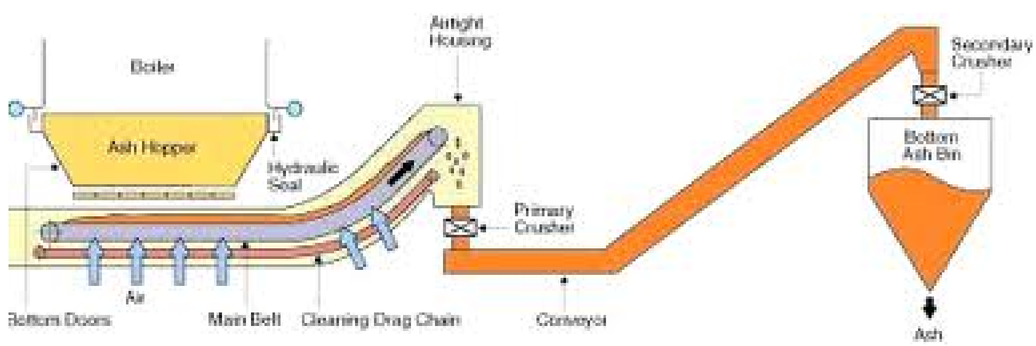

Figure 2.5: Dry bottom furnace[6]

2.4 Boiler Operation

The boiler is use to convert water into steam. This operation looks simple but it is more difficult. In boiler the operation starts with unloading coal from coal wagons. After unloading,

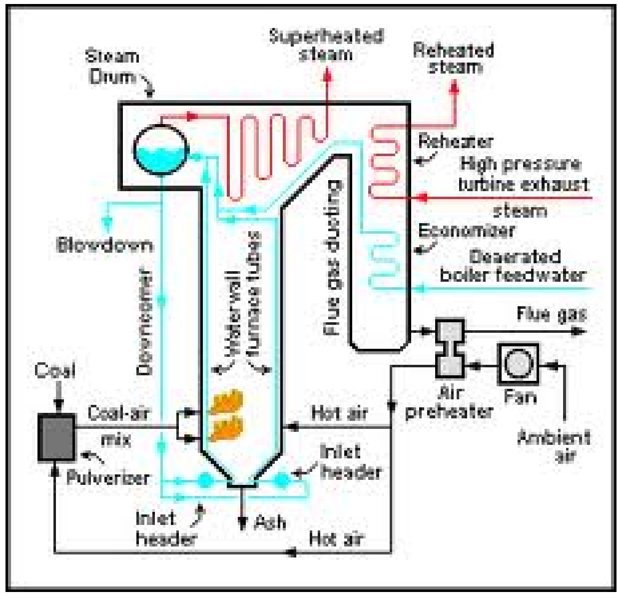

Figure 2.6: Boiler[7]

coal is supplied to the coal bunker using the belt conveyor. From bunker, it is supplied to the coal mill through coal feeder which supplies the measured quantity of coal. The coal is crushed to pulverized form into coal mill. The coal from coal mill supply to the furnace using the primary air which supply from PA fan. PA fan take the atmospheric air and supplied to the air pre heater where its temperature get increased. FD fan take air from atmosphere from which it is supplied to air pre heater where its temperature gets increased and after that it is supplied to the combustion chamber.

Water is supplied to the boiler from water reservoir and it is pumped through boiler feed pump. The water is first pass through economizer and then it is supplied to boiler drum. The water from boiler drum is supplied to the bottom ring header. From here it is flow through water wall tubes where it get heated and mixture of water and steam goes into the drum. The water in drum again repeats the cycle and steam goes to super heater. In super heater the steam get super heated and supply to the turbine.

The flue gas from furnace is removed using the induced draft fan. The ID fan sucks the flue gas by maintaining the negative draft in the furnace. The flue gas carries some useful amount of heat which can be transferred into various superheater, economizer and air pre heater. The ash particles from the flue gas are absorbed by electrostatic precipitator. So, only the gas is rejected to the atmosphere through chimney. The ash is mixed with water and pumped into the ash pond[8].

2.5 Performance Evaluation of Boilers

The purpose of perfromance evaluation of boiler is to find out the actual performance and efficiency of the boiler and compared it with design values. Performance evaluation is used to

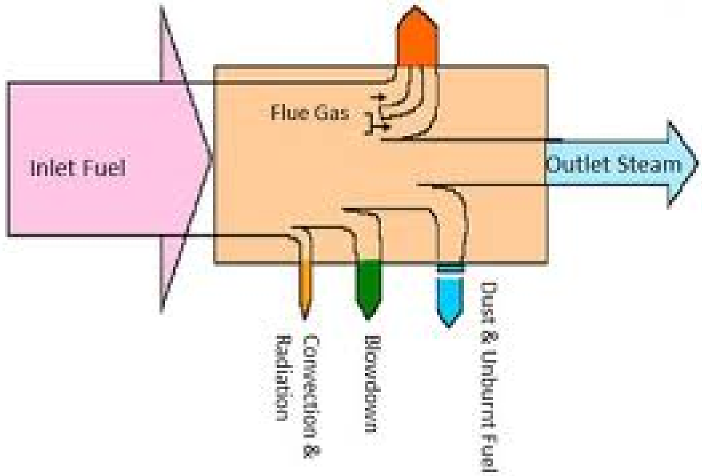

Figure 2.7: Diag. of boiler efficiency[2]

obtain the variations in energy performance and efficiency of the boiler. According to power test code ASME standered PTC 4-1 for steam generating units, the boiler efficiency can be evaluated by the following methods[2]:

1. The Direct Method(Input-output method): Which compare energy available with the working fluid to the energy supply by the boiler fuel.
2. The Indirect Method(Heat loss metod): Which shows the difference between energy input and various heat losses.

2.5.1 The Direct method

In this method, to obtain efficiency the output and input are only required. Beacause of this, it is also called as "Input-Output method". The efficiency can be find out through direct method using the following formula:

$$Boiler\,efficiency = (Heat\,output)/(Heat\,input) \times 100 \tag{2.1}$$

$$Boiler\,efficiency = \frac{(Quantity\,of\,steam\,generated \times (steam\,enthalpy - feed\,water\,enthalpy))}{(Quantity\,of\,fuel\,used \times Gross\,calorific\,value\,of\,coal)} \times 100 \tag{2.2}$$

2.5.2 The Indirect method

In this method, the boiler efficiency is evaluated by obtaining the various heat losses available in the boiler. This method overcome the disadvantages which happen with direct method can be overcome by this method. The efficiency is obtained by subtracting the various heat losses from 100. The efficiency get through Indirect method does not shows drastic change beacuse of the presence of error. The efficiency through this method can be calculated by the following formula:

$$Boiler\, efficiency = 100 – (various\, heat\, losses) \tag{2.3}$$

$$= 100 - (1 + 2 + 3 + 4 + 5 + 6 + 7 + 8) \tag{2.4}$$

The various heat losses occurring in the boiler are

1. Loss due to dry flue gas (sensible heat)
2. Loss due to hydrogen in fuel (H_2)
3. Loss due to moisture in fuel (H_2O)
4. Loss due to moisture in air (H_2O)
5. Loss due to carbon monoxide (CO)
6. Loss due to surface radiation, convection and other unaccounted
7. Unburnt losses in fly ash (Carbon)
8. Unburnt losses in bottom ash

The methodology to find out these different losses in the boiler is presented in energy efficiency booklet published by bureau of energy efficiency[2].

Chapter 3

Effect of water side deposits on the energy performance of water wall of boiler

For accurate working of thermal power plant, it is necessary to control contaminants in the power plant. In power plant, the water get contaminated from[9]:

1. Improper treated water stream results due to poor quality of input stream.
2. Because of improper operation and maintenance of water equipments.
3. Unable to handle damaged equipment such as deaerator, spray nozzles etc.
4. Due to leakage of condenser cooling water into DM water.
5. Because of leakage of untreated water into auxiliary cooling water.
6. Because of formation of chemicals into the steam during evaporation.
7. Because of accumulation of chemicals which is used for contamination control.
8. Presence of corrosion products in the feed water and condenser system.
9. Due to desuperheating spray in the steam for temperature control.

The presence of water contamination results in scaling on the inside surface of the water/steam pressure parts of the boiler, turbine, regenerative feed water heaters and condenser, coolers etc. The rate of deposition is generally depend upon the water/steam mixture composition , flow velocity of steam and water, composition of tube material and the temperature of the process[9]. As per IS 10391:1998[10], in power plants, deposits quantity up to 30µm both sub critical and super critical boiler are considered as in clean condition. For sub critical boiler, deposit quantity in the range of 30-80 µm and for super critical boiler deposit quantity in the range of 30-50 µm is considered as less dirty surface. When the deposit quantity reaches above 80 μm for subcritical boiler and 50 μm for super critical boiler, boiler is as very dirty and immediate cleaning is suggested in next maintenance work[10].

The effects of deposition[9] can be obtained as follows:

- It reduces the heat transfer in the boiler unit. The reduction in heat transfer rates affect the thermal response.
- Increase in resistance in the flow circuits causes increase in pumping power in the pump system and lower power output from the turbine.
- Failure of tube caused due to corrosion and hot spot becasuse of the presence of CuO and Na_2O respectively.
- Increases continuous/intermittent blow down to control contaminants which results in energy loss.
- Incorrect reply of thermostats.

3.1 Literature review

3.1.1 Literature

M. S. Bhatt[9] carried out analysis on 210 MW coal fired thermal power plant components by considering the overall system working at 100% maximum continuous rating and linear growth of deposition is assumed. He was carried out the analysis for whole power plant. The effect of scaling on the boiler component was shown in form of reduced heat transfer, which was in range of 0.2-2.0% for normal scaling.For pump, the increase in pumping power was finding out, which was in range of 0.6 - 7.6% for the boiler component. for boiler feed pump circuit increased pumping power was 29%. Low pressure heating circuit raise the pumping power upto 26%, for high pressure heating circuit it wae 21% and 18% for the lube oil cooler circuit. His work also shows the effect of overall scale build up which results in increased power input of 68 kW/μm and reduction in power output was about 25 kW/μm. The techniques to control comtaminant are also discussed in his work.

Quan Zhenhua et al.[11] investigated the effect of fouling on the heat transfer surface. In their work the heat transfer was considered as the forced convection heat transfer and the scaling happen due to calcium carbonate was considered. The different factors such as wall temperature, fluid velocity solution temperature, hardness and alkalinity were also considered to determine the fouling behavior. They examined the fouling. The variation in parameter of fouling observed from 380 to 2600 $kg{\cdot}W{\cdot}(m^4{\cdot}K)^{-1}$. The variation in thermal conductivity of fouling ranges from 1.7 to 2.2 $W{\cdot}(m{\cdot}K)^{-1}$.

Mohamed A. Antar and Syed M. Zubair[12] found out effect of fouling on the performance of multi-zone feed water heaters. In their work, it come to know that the condensing zone handled maximum amount of heat duty and least amount heat duty handled by subcooling zone. The desuperheating, condensing and subcooling zone of feedwater heater was affected by the fluid mass flow rate and overall heat transfer coefficients. In this analysis to evaluate the effect of fouling on heat exchanger performance, two types of fouling models were considered. The reduction in heat transfer for the heat exchanger was 2.7% in 3 year. The increase in outlet shell-side fluid temperature was about 6.3%.

Xu Zhi-Ming et al.[13] considered Huaneng Dalian Power Plant and Changshan Power Plant for their work. In this paper the method to found out fouling cost for boiler and turbine are shown. Using the data available from the power plant the cost of fouling for increase of product costs, excess surface area, operating maintenance and product loss were calculated. The analysis shows that economical loss in boiler and turbine because of fouling reaches to 4.68 billion dollers for China and it is about 0.169% GDP of China in 2006.

Ahmet fertelli and Ertan buyruk[14] presented the numerical study for effect of deposit formation on a single tube of cross flow heat exchanger. Calculations were carried out by using ANSYS software program. Heat transfer efficiency of the tube was calculated using thermal resistance approach for clean and foulded condition.It was examined that the deposit formation leads to reduction in the heat transfer rate strongly.

A. J. Karabelas et al.[15] investigated the heat exchanger to found out the effect of liquid side on it. The analysis was carried out study to modify the design standards for novel and conventional heat exchangers. In this paper, for plate type heat exchanger, new fouling data are recorded for particles of mean size of 5 μm. The analysis shows that the fouling resistance were mostly affected by the flow passage geometry and the fluid velocity.

P. Jain[16] carried out failure investigation of water wall tubes of different capacity boiler and indicated the reasons of tube failure. He examined the presence of deposits due to raw water ingress in the condenser water. The main reason for the tube failure is given by the overheating of tube which results due to the presence of deposits in the internal surface.

3.1.2 Water wall system

Any boiler needs primarily an evaporating surface for the conversion of water into steam. In the early periods of boiler development the evaporating surfaces are formed by placing many coils of tubes or tube banks across the flow path of the hot gases from the furnace and circulating water through these tubes. With the need for increase in steaming capacity of boilers and to minimize the furnace heat losses by radiation, in modern boilers the evaporating surface is made of water walls, which forms the major part, if not all, of furnace enclosure. The water walls are tube panels through which the water from steam generation will be circulated. In the water wall only heat is added to evaporate the feed water to steam.

In boilers, water walls completely cover the interior surface of the furnace providing practically complete elimination of exposed refractory surface. Water walls serve as the only means of heating and evaporating the feed water supplied to the boiler from the economizers. Water walls usually consist of tangential vertical tubes and are connected at the top and bottom to headers. These tubes receive water from the boiler drum by means of down comers connected between drum and water walls lower header. In a boiler approximately 50 percent of the heat released by the combustion of fuel in the furnace is absorbed by water walls[8]. Heat so absorbed by the water walls is used in evaporation of water supplied to the boiler. The mixture of steam and water is discharged from the water wall tubes into the upper wall header and then passes through riser tubes to the steam drum.

The steam is separated and accompanying water together with the incoming feed water is returned to the water walls through the down comers.

Advantages of water walled furnace

A boiler with water walled furnace has many advantages compared to furnace of any other type.

The major advantages of water walled furnaces[8] are:

- In furnace not only combustion but also heat transfer is taking place simultaneously.
- The maintenance work involved in repairing the firebricks (which is otherwise necessary) is completely eliminated.
- Due to heat transfer in the furnace, temperature of the flue gas leaving the furnace is reduced to the acceptable level of the superheating surfaces.
- Higher heat loading in the furnace is possible, as heat is being simultaneously removed by heat transfer, and hence economy in surfacing.
- Providing a gas tight seal to the combustion chamber to prevent air infiltration.

3.1.3 Uniform scale thickness

The formation of scale is unpredicted because it is uneven and non uniform. The higher temperature zone has higher thickness of scale and lower temperature zone has lesser thickness of scale. The scale is higher in low velocity zone and lower in high velocity zone. Even in a given local zone, it deposits unevenly. To relate the system performance to deposition, it is assumed that there is a scale of uniform thickness on the entire heat transferring surface. Because it is not possible to determine scaling in all pipes at all locations so in present work uniform scaling is assumed.

3.2 Experimental work

The current analysis belong to WTPS Boiler No.4. The analysis is carried out at full plant load and scale thickness is considered as uniform scale thickness. In present work, the effect of waterside deposits on the water wall by finding its effect on heat transfer rate and pumping power and boiler efficiency. The laboratory analyses of deposits were carried out by BHEL, Tiruchirappalli.

3.2.1 Composition of deposits

BHEL has carried out remaining life assessment[17] study in GSECL Wanakbori thermal power station boiler No.4, 210 MW, against order placed by M/s GSECL Wanakbori TPS. The study was carried out for boiler pressure parts taking care of the stipulated condition specified in IBR Amendment 391A. The uncertainity in chemical composition found as $\pm 0.2\%$.

BHEL Tiruchirappli[17] carried out analysis of the deposit sample. The composition of the deposits given in table 3.1. Table 3.1 shows that the water side deposits are comoposition of SiO_2, Copper and Fe_3O_4. The measurements of thermal conductivity of the deposit samples were not given in the remaining life assessment study carried by BHEL used in present work. As composition of deposits on waterwall in WTPS is similar to work presented by M.S.Bhatt[9] as given in table 3.1. So thermal conductivity values provided in his work is used in present work. Hence, it was decided the thermal conductivity of deposits of Wanakbori TPS boiler no.4 were taken as in the range of 0.5 -2.0 W m^{-1} K^{-1}.

Table 3.1: Result of chemical analysis of deposits from internal surface of water wall [17]

Si No.	Particular	Water wall deposits	
		WTPS Boiler No. 4	Bhatt[4]
1	Silica(SiO_2)	1.5	0.70
2	Iron Oxide(Fe_3O_4)	90.4	91.7
3	Copper(Cu)	5.8	5.80
4	Calcium Oxide(CaO)	Traces	1.2
5	Magnesium Oxide(MgO)	Traces	Traces
6	Sodium Oxide(Na_2O)	0.3	0.2
7	Phosphate	1.2	Traces
8	Loss in ignition at 600°C	0.7	0.5

3.2.2 Deposit build up

Deposit build up is mmeasured in mg/cm^2 or in meter. A deposit of 190 mg/cm^2 is similar to scale of 1 mm. As per IS 10391:1998[10], in power plants, deposits quantity up to 30µm both sub critical and super critical boiler are considered as in clean condition. For sub critical boiler, deposit quantity in the range of 30-80 µm and for super critical boiler deposit quantity in the range of 30-50 µm is considered as less dirty surface. When the deposit quantity reaches above 80 µm for subcritical boiler and 50 µm for super critical boiler, boiler is as very dirty and immediate cleaning is suggested in next maintenance work[10]. The maximum time required for accumulation of deposits in boiler parts and regenerative feed water heaters are considered as 200,000 hours, for turbine it is 60,000 hours and 20,000 h for condensers. The offline cleaning of bolier and feed heaters were taken at 20 years, for turbine 5 years and condensor must be clean annually

When the quantity of deposit exceeds 40 mg/cm^2, the tube surfaces are considered to be very dirty surfaces as per the Indian Standard 10391-1998 and the chemical cleaning is suggested to improve the heat transfer and reduce the overheating. The guidelines are given in table 3.2.

Table 3.2: Acid cleaning guideline(IS 10391:1998)[?]

Boiler Type	Internal Deposit Quantity Limits		
	Clean $\frac{mg}{cm^2}$/µm	Less Dirty $\frac{mg}{cm^2}$/µm	Very Dirty $\frac{mg}{cm^2}$/µm
Supercritical Units >220kg/cm^2	<15 / <30	15-25 / 30-50	>25 / >30
Subcritical Unit s <220kg/cm^2	<15 / <30	15-40 / 30-80	>40 / >80

In remaining life assessment of Wanakbori TPS boiler, it is shown that the deposits quantity of water wall varies from 36.8 to 80.5 mg/cm^2 with 5.8 % of copper. The quantity of deposits exceeds 40 mg/cm^2 and the boiler is rated as "Dirty" and suggested the chemical cleaning in the next available maintenance work. The quantity of water wall deposits are given in table 3.3.

Table 3.3: Waterwall deposit quantity[?]

	Deposit quantity(mg/cm^2)
1. Front waterwall	36.8
2. Rear waterwall	54.1
3. LHS waterwall	55.5
4. RHS waterwall	40.0

3.3 Result and discussion

With the presence of deposits inside the tube of waterwall, the following parameters are to be affected:

- Effect on heat transfer of waterwall
- Effect on boiler efficiency

3.3.1 Effect on heat transfer

The presence of the waterside deposits in the waterwall has effect on heat transfer in the component of boilers. The reduction in heat transfer is occur due to two reasons,

1. Presence of an extra thermal resistance.
2. Reduced in heat trnasfer area because of the scaling inside tube.

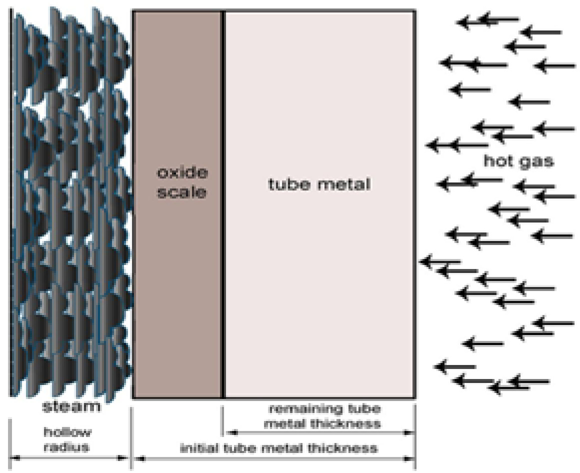

Figure 3.1: Heat transfer through waterwall[18]

The effect of scales of different compositions on the relative heat transfer rate is quantified in Bhatt[9].

Bhatt[9] gives the reduction in heat transfer (in tube type heat exchange surfaces) can be quantified by the decrease in the overall heat transfer coefficient (based on LMTD and tube inside diameter) as,

$$\frac{Q_S}{Q_O} = \frac{1}{1 + \frac{C\delta_S U_O f_a}{\lambda_s AR}} \tag{3.1}$$

Where,

C = Conversion factor (1E-06 if the scale thickness is in μm; 1E-03 if it is in mm).
δ_s = Scale thickness
U_o = Overall heat transfer coefficient in clean condition (W m^{-2} K^{-1})

$$U_O = \frac{1}{\frac{d_o}{di}\frac{1}{h_i} + \frac{d_o ln(\frac{d_o}{d_i})}{2k} + \frac{1}{h_o}} \tag{3.2}$$

f_a = fraction of the area of the heat exchanger affected by scaling
λ_S = thermal conductivity of the deposit
AR = ratio of surface area reduced by scaling

$$AR = \frac{d_o\ -\ 2\delta_s}{d_o} \tag{3.3}$$

Where,

d_o = Tube inside diameter in clean condition

The maximum area reduction ratio is around 0.98 in the boiler pressure parts. The boiler parameters are shown in table 3.4.

Table 3.4: Boiler Parameter

Parameter	
$T_{w/s\,in}$ (^oC)	290
$T_{w/s\,out}$(^oC)	340
$T_{fg\,in}$(^oC)	1250
$T_{fg\,out}$(^oC)	850
d_o(mm)	57.2
δ_o(mm)	0.5
U_o(W/mK)	23
AR	0.982

The value of Q_S/Q_O is given for different λ_S are given table 3.5.

Table 3.5: Q_S/Q_O for different λ_S

λ_S	Q_S/Q_O
0.5	0.927
0.8	0.953
1.3	0.970
1.6	0.976
2.0	0.980

As per data from the table 3.5 we can observe that as the thermal conductivity of the water side deposite is increasing ratio of Qs/Qo is increasing which shown in figure 3.2.

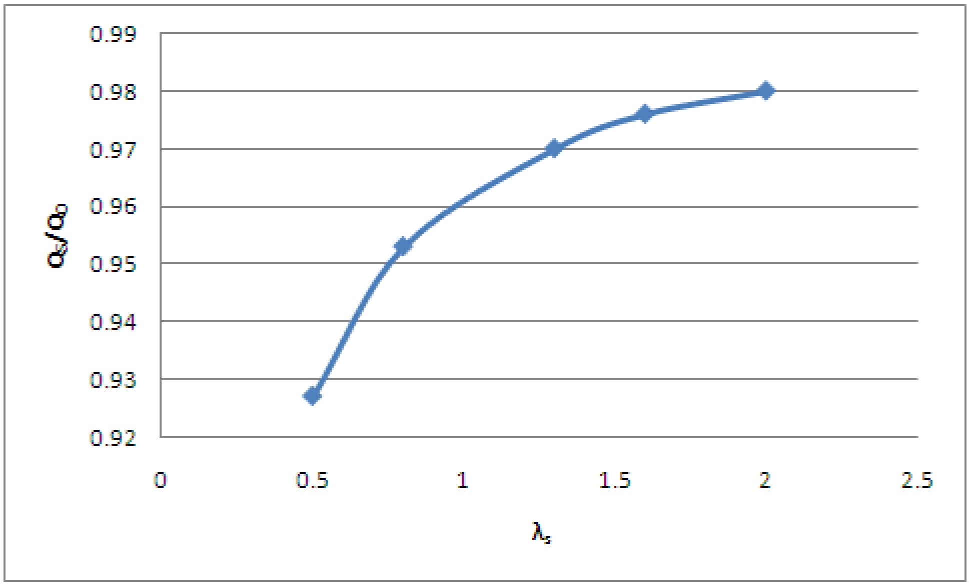

Figure 3.2: variation of Qs/Qo with thermal conductivity of water side deposite

3.3.2 Effect on boiler efficiency

The effect of contaminants on the waterwall can be obtain by considering its effect on boilers. The boiler shows multiple and inter connected effect. For given boiler the efficiency is found which is shown in section 3.3.2.1 .The efficiency is calculated using indirect method.

3.3.2.1 Boiler efficiency by Indirect method

Calculation for the efficiency of boiler unit number 4 of WTPS through indirect method is given below. In this method the efficiency of boiler is obtain by subtracting different heat losses from the 100. So, the different heat losses are given to calculating the boiler efficiency. The different heat losses are considered according to design condition.

$$Boiler\,efficiency = 100 - (various\,heat\,losses) \tag{3.4}$$

The various heat losses occurring in the boiler are

Loss due to dry flue gas (sensible heat)	= 5.48 %
Loss due to hydrogen and moisture in fuel (H_2 and H_2O)	= 5.43 %
Loss due to moisture in air (H_2O)	= 0.14 %
Loss due to Combustible in Ash	= 1.22 %
Loss due to Sensible Heat in Ash	= 0.38 %
Loss due to surface radiation, convection and other unaccounted	= 0.20 %
Unaccounted ,manufacturer's margin	= 0.50 %

Boiler efficiency of unit no.4 by Indirect method	= 100 – (1+2+3+4+5+6)
	= 86.65 %

3.3.2.2 Effect of deposits on efficiency

The presence of deposits generally affect the heat transfer, it reduces the heat transfer and as a result boiler efficiency also get reducing. We are considering efficiency obtained through indirect method for the calculation purpose because indirect method is more accurate than direct method. In M. Bhatt[9] the sensitivity indices are given for the deposit thickness. The thickness of per μm of deposits reduces 0.0021% of the boiler efficiency. So, the overall effect of scaling on the boiler efficiency (%) is given by,

$$Boiler\,efficiency(scale\,condition) = Boiler\,efficiency(clean\,condition) – 0.0021(Scale\,thickness) \tag{3.5}$$

$$\eta_{B,s} = \eta_{B,c} - 0.0021(\delta_S) \tag{3.6}$$

Where, δ_S is the uniform scale width in the boiler (0–500 μm).

The scale thickness of the deposits in the boiler is around 500 μm.

$$\eta_{B,s} = 85.60\ \%$$

The reduction in boiler efficiency due to the waterside deposits of 500μm is around 1%.

Chapter 4

Improvement in performance of boiler after installation of burner tilting device

For boilers in power plant, It is essential to achieve the better efficiency for reducing the cost of generation and so the operators do exercise to maintain the superheat and reheat steam temperature to the rated value. The coordinated water flow and spray attemperation are use to control the rated steam temperature. The Tilting of burner, gas recirculation (GR), divided back pass dampers, excess air and steam bypass methods are used to control temperature[19]. Traditionally, spray attemperation system is used for the superheat/reheat steam temperature control. Boiler outlet steam temperature control through spray attemperation is one of the most challenging control problems in thermal power plants due to the complexity and multitude of influencing variables and non-linear load-dependent process response. However the use of spray attemperation for temperature control results in the loss of efficiency. So, through the burner tilting method, we keep away the effect of reheater spraying on the boiler efficiency.

4.1 Literature Review

4.1.1 Literature

Ravindra Kumar P. et al.[19] shown the different ways to control the reheater steam temperature instead of using the spray water attemperation. The different methods given to control reheat steam temperature are use of burner tilting device, gas recirculation, devided back pass dampers, excess air and steam by pass. It is suggested to use the burner tiling device and gas recirculation system instead of spray attemperation because use of spray attemperation causes loss in pressure and loss of heat energy due to spray of cold water in the steam.

T. K. Seal et al.[20] was carried out analyses of improvement of steam temperature control through improved control system at NTPC – Rihand stage of 2×500 MW. They found reduction in desuperheater spray is around 9 t/hr and because of this the reduction in energy loss is around 1.195 kcal/KWH. This results in the saving of around 13 lacs and reduction in CO_2 is around 1500 MT.

Wang yan-jun et al.[21] did study on temperature control by tilting Burner in 1025 t/hr controlled circulating boiler. In 102 5t/hr controlled circulating boiler, temperatures of superheated steam and reheated steam are controlled by tilting of burner. After commission, tilting devices of burner nozzle ever broke down and failed to control the temperature. Discharged steam temperature of high pressure cylinder in turbine is 10 ~ 20°C higher than heat designed value. Emergency spray has to be applied to control the reheated steam temperture. The large heating surface of reheater in boiler makes the attempering water in reheater be 20 ~ 30 t/hr more than the designed value, which obviously affects the economy of unit operation. Through investigating study and boiler test, improving measures were taken on the tilting structure.

Yin Jie and Wang Pei-hong[22] shown the influence of different spray desuperheating modes on the thermal economy of unit in power plant. The spray desuperheating system, an important component of thermodynamic system in thermal power plants, directly impacts the thermal economy of the unit .On the hypothesis that the temperature of the main steam and the reheating steam experience the same temperature drop and the spray desuperheating flow rate are also of the same, by using the equivalent enthalpy drop model that evaluates the influence of different spray desuperheating modes on thermal economy, the influence of spray desuperheating of superheaters and reheaters on unit thermal economy were calculated respectively under three typical operational conditions of the domestically-made subcritical 300 MW unit.

In T.M. Peas[23] the method of energy balance for reheater is discussed. The energy balance method for reheater is use to find out the loss of energy due to the desuperheater spray. The analysis show the reduction in boiller because of the spray attemperation in reheater. In this paper, author shows that reduction of spray in reheater results in proper heat transfer and reduces the flue gas temperature upto 46.6 $^\circ$C.

David Lindley[24] distinguished the different spray attemperation methods, the information about temperature control through burner tilting mechanism and detail about gas recirculation system is discussed. In his paper the information of mechanically atomized attemperator, variable area attemperator, variable annulus desuperheater, and variable orifice attemparator is given.

4.1.2 Need for steam temperature control

Load cycling and reduced minimum points raise important operational issues for fossil - fuel power plants. The limits on maneuverability and the minimum load for a specific power plant often are governed by steam temperature control. Operation beyond these limits results in excessive variations in steam temperature. The component that handle high temperature steam are among the most costly in the power plant, include the main steam turbine and the superheat section of the boiler. Lengthy and costly outages can result from damage to this equipment. Large transients of steam temperature can cause such damage and reduce equipment life through the process of thermal-cycling induced stress cracking.

The temperature of superheated and reheated steam should not be higher than the rated value because it causes the metallurgical problems in boiler tubes and also in the turbine component. However, the steam temperature below the rated value resutls in increasing in heat energy for specific power generation. Generally reduction in teperature of 100 $^{\circ}$C in higher capacity power plant will result in increases of 0.3 % of plant heat rate[24]. Hence it is necessary to maintain the superheated and reheated steam temperature within the small range of rated values.

In a boiler, superheated and reheated steam temperature depend upon many factors such as fuel quality, cleanliness of boiler and scaling of heat transfer regions etc,. The clean furnace as per design condition increases the transfer of heat in furnace and so the temperature of flue gas reduces and hence the superheat and reheat temperature also reduces. On the other hand, the dirty furnace results due to reduced coal quality, fouling etc., results in higher flue gas teperature and hence the superheat and reheat temperature increases.

Improved steam temperature control can benefit a power plant[8] by

- Reducing the probability of catastrophic failure during startup by providing a better match of steam temperature to turbine metal temperature.
- Increasing the life of high temperature sections of the turbine and boiler by reducing by reduced temperature cycling.
- Increasing the maneuvering capability of the unit, resulting in reduced cycling demand on less responsive units.
- Increasing the steam temperature set points, achieving higher efficiencies based on demonstrated reduction in steam temperature variations
- Increasing the range of automatic steam temperature control, thereby reducing the probability of temperature cycling due to manual control.

4.1.3 Temperature control with tilting burners

In tilted burner arrangement, the burning fuel in a corner-fired boiler forms a large swirling fireball which can be moved to a higher or lower level in the furnace by tilting the burners upwards or downwards with respect to a mid position. The repositioning of the fireball changes the pattern of heat transfer to the various banks of superheater tubes and this provides an efficient method of controlling the steam temperature, since it enables the use of spray water to be reserved for fine-tuning purposes and for emergencies.

In addition, the tilting process provides a method of controlling furnace exit temperatures. With such boilers, the steam temperature control systems become significantly different from those of boilers with fixed burners. The boiler designer is able to define the optimum angular position of the burners for all loads, and the control engineer can then use a function generator to set the angle of tilt over the load range to match this characteristic. A temperature controller trims the degree of tilt so that the correct steam temperature is attained.

Tilted burners are generally used in tangential fired furnace. In tilted burners, the burner can be move upword or downword and accordingly change the heat transfer in furnace. When the reheat steam temperature is above the rated value, the burner move downward which reduces the heat aborption and the temperature start to reduces. On other hand for lower steam temperature the burner moves upward. The different position of burner is shown in figure 4.1.

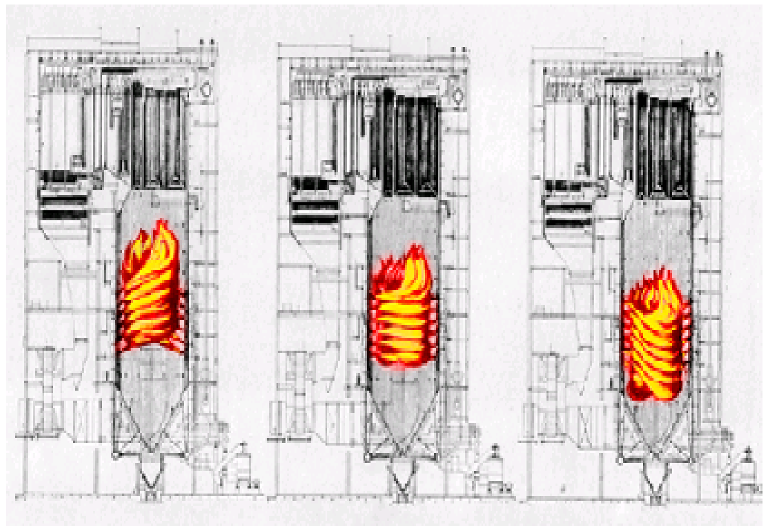

Figure 4.1: Furnace with burner tilting device

4.1.4 Reheat steam temperature control

In boilers with reheat stages, changes in firing inevitably affect the temperature of both the reheater and the superheater. If a single control mechanism were to be used for both temperatures the resulting interactions would make control-system tuning difficult, if not impossible, to optimise. Such boilers therefore use two or more methods of control.

Because of the lower operating pressure of reheat steam systems, the thermodynamic conditions are significantly different from those of super heaters, and the injection of spray water into the reheater system has an undue effect on the efficiency of the plant. For this reason, it is preferable for the reheat stages to be controlled by tilting burners (if these are available) or by apportioning the flow of hot combustion gases over the various tube banks. However, if the superheat temperature is controlled by burner tilting, gas apportioning or spray attemperation must then be used for the reheat stages.

In boilers with fixed burners, steam-temperature control may be achieved by adjusting the opening of dampers that control the flow of the furnace gases across the various tube banks. In some cases two separate sets of dampers are provided: one regulating the flow over the superheater banks, the other controlling the flow over the reheater banks.

Between them, these two sets of dampers deal with the entire volume of combustion gases passing from the furnace to the chimney. If both were to be closed at the same time, the flow of these gases would be severely restricted, leading to the possibility of damage to the structure due to overpressurisation. For this reason the two sets are controlled in a so-called 'split-range' fashion, with one set being allowed to close only when the other has fully opened.

These dampers provide the main form of control, but the response of the system is very slow, particularly with large boilers, where the temperature response to changes in heat input exhibits a second-order lag of almost two minutes' duration. For this reason, and also to provide a means of reducing the temperature of the reheat steam in the event of a failure in the damper systems, spray attemperation is provided for emergency cooling.

The spray attemperator is shut unless the temperature at the reheater outlet reaches a predetermined high limit. When this limit is exceeded, the spray valve is opened. In this condition, the amount of water that is injected is typically controlled in relation to the temperature at the reheater inlet, to bring the exit temperature back into the region where gas-apportioning or burner tilting can once again be effective. The relationship between the cold reheat temperature and the required spray water flow can be defined by the boiler designer or process engineer.

4.2 Experimental work

The present analalysis is related to Wanakbori thermal power stations Boiler No.5 of 210 MW. The analysis is carried out at full plant load. We analyzed the effect of burner tilting device by considering its effect on desuperheater spray in reheater. The burner tilting device control the outlet steam temperature through its upward and downward movement which results in the reduction of desuperheater spray in the reheater device. The reduction in reheat spray results in coal consumption, economical gain and reduction in CO_2 emission.

4.2.1 Reheat spray

The main effect of burner tilting device shows on reheater spray and reduces the amount of spray. So, to get the benefit of burner tilting device, the amount of rehaeter spray is measured before and after the installation of burner tilting device. The burner tilting device was installed in April-2013.

The amount of RH spray before burner tilting device is given in table 4.1.

Table 4.1: RH spray before BTD

RH Spray (t/hr)	Sept -12	Oct -12	Nov-12	Dec-12	Jan-13	Feb–13	Mar-13
	24	18	23	25	21	24	23

The amount of RH spary after installing burner tilting device is given in table 4.2

Table 4.2: RH spray after BTD

RH Spray (t/hr)	May-13	June -13	July-13	Aug-13	Sept-13	Oct–13	Nov-13
	0	2	3	3	2	2	2

The difference in RH spary before and after the installation of burner tilting device is shown in table 4.3.

Table 4.3: Reduction in RH spray

Avg. RH spray before BTD (t/hr)	Avg. RH spray after BTD (t/hr)	Reduction in RH spray (t/hr)
22.57	2.0	20.57

By considering difference between average RH spray, the reduction in spray is obtained as 20.57 t/hr which is consider due to the installation of burner tilting device.

4.3 Results and discussions

4.3.1 Effects of reheat spray

The reheat spray results in the performance deterioration[25] which is mainly due to:

1. A change in the heat added to the feed water in the boiler because of the difference in the heat given to the spray compared to that given to the feed.
2. A reduction of power output from the HP cylinder because of the reduced steam flow, as every kilogram of spray water flow results in a comparable reduction of HP cylinder steam flow.
3. A reduction of feed flow through the feed water heaters because the spray water is tapped of the feed system before the feed water heaters. This will results in a reduction of bled steam flow through heaters and hence there will be an increase in steam flow through the IP and LP cylinders. So there will be an increase in the power output from these cylinder.

4.3.2 Economical Perspective

The plant load factor of Wanakbori TPS is 0.75. So, the generation of electricity per year

$$= 210000 \times 365 \times 24 \times 0.75$$

= 1379700000 kW/year

For unit heat rate deviation (1 kcal/kWh)

The deviation in heat rate means the more amount of energy required to obtain the desired energy generation. Therefore deviation of heat rate for the power plant results in extra coal consumption for the required energy generation. The energy generation per year of WTPS boiler is 1379700 kU/year and the gross calorific value of the coal is considered as 4100 kCal/kg.

For deviation of 1 kCal/kWh the additional coal cosumption is:

$$Additional\,coal\,consumption\,(1kcal/kWh) = \frac{Heat\,rate\,deviation \times\ Energy\,generation\,per\,year}{GCV\,of\,coal} \tag{4.1}$$

= 336.512 MT

The additional coal consumption results in economical loss and increase in CO_2 emmision.

The cost of coal is around 3800 Rs/MT. The additional coal consumption of 336.512 MT results in economical loss per year.

$$Economical\,loss/year = Additional\,coal\,consumption \times cost\,of\,coal \tag{4.2}$$

= 12.80 lacs

The additional coal consumption of 336.512 MT increarse in CO_2emmision per year is

$$CO_2\,emmision/year = Additional\,coal\,consumption \times carbon\,in\,coal \times CO_2\,in\,coal \tag{4.3}$$

= 336.512 × 30% carbon × 44/12

= 369 MT of CO_2emmision

4.3.3 Change in boiler heat supplied to steam

The conditions of steam flow, design heat rate, actual heat rate., are given below,

Main steam flow Q_S	= 690 T/hr
Design heat rate HR	= 1988.81 kCal/KWH
	= 8327.147 kJ/KWH
Actual heat rate HR_s	= 9600 kJ/KWH

The reheat spray results in loss of feed flow which results in reduce in heat rate but at the same time that amount of flow is spray into the reheater which results in the increase in heat rate. Therefore the final heat rate is obtained by combining both the heat rate. Because of 20.57 t/hr of RH spray increase in heat rate is:

$$Combined\,heat\,rate = \frac{[(Q_s - 20570) \times HR] \,+\, [(20570 \times HR_s)]}{Q_s} \tag{4.4}$$

$$= 8365.093 \text{ kJ/KWH}$$

$$= 1997.87 \text{ kCal/KWH}$$

By comparing this obtained heat rate with design heat rate of the system, the loss of energy is obtained.

The loss in energy

$$= 1997.87 \text{ - } 1988.81$$

$$= 9.06 \text{ kCal/KWH}$$

For Wanakbori TPS boiler reduction of 20.57 t/hr of reheat spray results in 9.06 kCal/KWH of energy gain. As shown in section 4.4.2, the deviation of 1 kCal/KWH energy increases coal consumption of 336.512 MT.

The saving in 9.06 kCal/KWH of energy reduce the additional coal consumption.

$$\text{The reduced coal consumption} = 3048.80 \text{ MT}$$

$$\text{Economical gain/year} = 115.96 \text{ lacs}$$

$$CO_2 \text{ emission reduction per year} = 3343 \text{ MT of } CO_2 \text{ emission}$$

After installation of burner tilting device, the gain in energy is obtain as 9.06 kCal/kg and the reduction in coal consumption is obtained as 3343 MT per year.

4.3.4 Increase in power output from HP cylinder

The reduction of 20.57 t/hr spray flow, all of which bypasses the HP cylinder, so increase the flow of steam through the HP cylinder by the same amount. An increased steam flow results in a comparable increase of HP cylinder flow, which increase of power output from the HP cylinder.

Data of main steam		**Data of R/H inlet**	
Pressure P_O	$= 154.6$ kg/cm^2	Pressure P_{ri}	$= 37.56$ kg/cm^2
Temp. T_O	$= 540°C$	Temperature T_{ri}	$= 336.9°C$
Enthalpy h_O	$= 3419.6$ kJ/kg	Enthalpy h_{ri}	$= 3069.8$ kJ/kg

$$Heat\,drop\,around\,HP\,cylinder = Steam\,enthalpy\,at\,inlet - Steam\,enthalpy\,at\,exhaust \tag{4.5}$$

$$= h_o\text{-}\ h_{ri}$$

$$= 349.8 \text{ kJ/kg}$$

With reduction in 20.57 t/hr of spray gain in electrical power

$$= \frac{20570 \times 349.8}{3600}$$

$$= 1988.72 \text{ kW}$$

The reduction in spray flow which bypasses the HP cylinder results in increase of power output from the HP cylinder about 1988.72 kW.

4.3.5 Loss of power output from IP and LP cylinder

As shown in figure 4.2 with spray water, the feed flow through the feed water heaters reduces because the spray water is tapped of the feed system before the feed water heaters. This will results in a reduction of bled steam flow through heaters and hence there will be an increase in steam flow through the IP and LP cylinders. So with reheat spray there will be an increase in the power output from these cylinder.

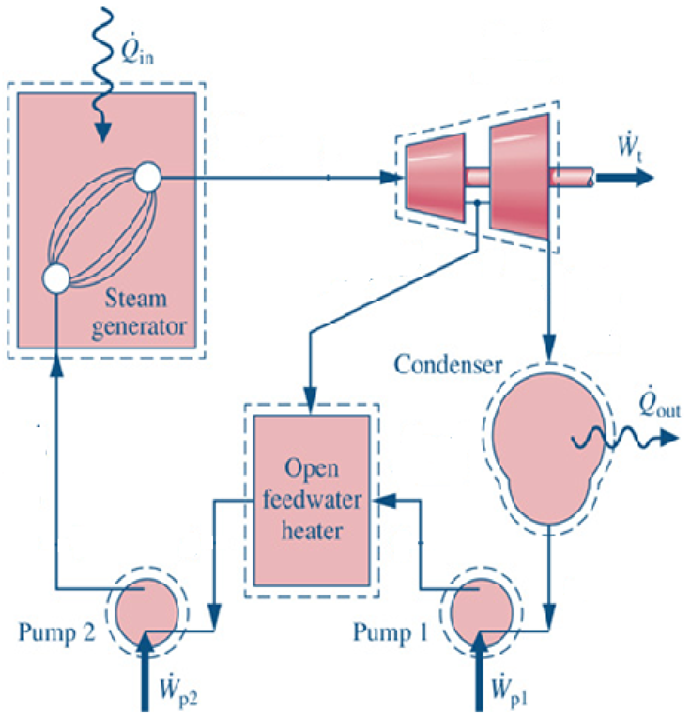

Figure 4.2: Flow of water in power plant [26]

After the installation of burner tilting device, the reheat spray reduces which means the more feed flow through the feed water heaters and decrease in steam flow through the IP and LP cylinders which results in loss in power output from these cylinders.

The design feed quantity through HP feed heaters (Q_f) is 191.67 kg/s.

HP feed heater is a kind of heat exchangers. In these the the feed water is heated by obtaining the heat from the steam which is generally bled steam.

heat lost per kg of steam = heat accepted per kg of feed water

$$Q_B(h_b - h_c) = Q_f(h_o - h_i) \tag{4.6}$$

Where,

Q_B = Quantity of bled steam, kg/s

Q_f = Quantity of feed water, kg/s

h_b = Enthalpy of bled steam at inlet, kJ/kg

h_c = Enthalpy of bled steam at outlet, kJ/kg

h_o = Enthalpy of feed water at inlet, kJ/kg

h_i = Enthalpy of feed water at outlet, kJ/kg

Assume the heat surrended per kilogram of steam and the heat accepted per kilogram of feed water remains constant.

Then,

$$Q_B \propto Q_f \tag{4.7}$$

So,

$$\frac{Q_{B1}}{Q_{B2}} = \frac{Q_{f1}}{Q_{f2}} \tag{4.8}$$

Where,Q_{B1} and Q_{f1} refer to design condition and Q_{B2} and Q_{f2} refer to actual condition respectively.

The design feed quantity (Q_{f1}) = 191.67 kg/s.

In actual condition there is spray flow of 20.57 t/hr.

The actual feed quantity (Q_{f2}) = 176.50 kg/s

$$Q_{B2} = Q_{B1} \times 0.97$$

Which means the bled steam to each HP heater will be reduced to 0.97 of its design value which is shown in table 4.4.

Table 4.4: HP heaters

	HP heaters	
	5	6
Design bled steam flow kg/s	10.09	16.27
Reduced bled steam flow kg/s	9.78	15.78
Reduction in bled steam flow kg/s	0.31	0.49

As mention above, HP heater is a kind of heat exchanger in which temperature of feed water is increased throgh the heat gain from the bled steam.

Heat drop of steam around HP heater 5

$$= \text{Enthalpy of steam at inlet - Enthalpy of steam at outlet}$$

$$= 3323.64 - 2870.2$$

$$= 453.44 \text{ kJ/kg}$$

The reduced reheater spray results in reduction of extra generator output because of reduction in steam bled at HP heater.

Loss of extra generator output due to increased bled steam at HP heater 5

$$= \text{Reduction in bled steam flow} \times \text{Increase in heat drop}$$

$$=140.57 \text{ kW}$$

Heat drop of steam around HP heater 6

$$= \text{Enthalpy of steam at inlet - Enthalpy of steam at outlet}$$

$$= 3070.32 - 2973.5$$

$$= 96.82 \text{ kJ/kg}$$

Loss of extra generator output due to increased bled steam at HP heater 6

$$= \text{Reduction in bled steam flow} \times \text{Increase in heat drop}$$

$$= 0.53 \times 96.82$$

$$= 47.44 \text{ kW}$$

The total reduction in extra generator output is the combination of loss of generator output at HP heater 5 and 6.

Hence the total reduction in extra electrical output because of reduced bled steam flow

$$= 188.01 \text{ kW}$$

The reduction in reheat spary reduces the flow of steam through IP and LP cylinder which results in loss of extra generator output of 188.01 kW.

Hence the total gain in energy and saving of cost in energy generation are shown below.

Gain in HP cylinder	= 1988.72 kW
Loss in IP and LP cylinder	= 188.01 kW
Total gain	= 1800 kW
Unit cost of electricity generation	= 3 Rs/unit
Saving in generation	= 350 lacs

Chapter 5

Optimization of air fuel ratio through SADC (Secondary air damper control)

Combustion is a chemical reaction between fuel and oxygen present in air which causes release of heat energy. In power plant when coal is fired into the furnace, the combustible substances such as carbon, hydrogen, sulphur etc. present in the coal get chemically combined with oxygen. The oxygen is obtained from the air which is mixture of 23.2 % oxygen and 76.8 % nitrogen by weight. This air is as necessary as fuel itself to achieve combustion.

Under complete combustion, the fuel and oxygen are so chemically combined that both the constituents are totally consumed with no combustible or uncombined oxygen present in the flue gas. However practically if there is not sufficient amount air available to complete the combustion process, some of the fuel is left unburned which results in reduction in efficiency and undesirable emission. To avoid this situation excess air is to be supplied however if the supplied air is more then it results in the excess amount of oxygen present in the flue gas because it does not react during the combustion and carried out the usable heat in form of stack losses. So it shows that some amount air above and beyond stoichiometric requirements is needed for the complete combustion of fuel.

5.1 Literature review

5.1.1 Secondary air

It is known that coal is a combination of carbon, volatile matter, moisture and ash particle. The volatile matter is mainly comprises of gases such as methane, hydrogen, nitrogen, oxygen and moisture. When the coal particles enter the furnace because of the intense radiation present its temperature starts increasing, causing its expansion. Volatile matter comes out from the coal particle and starts burning. Oxygen in the primary air gets consumed in the combustion of volatile matter. Since volatile matter is gaseous in nature, it mixes easily with the air and hence its combustion is very rapid. The solid particles, which are without of volatile matter are known as soot particle.

Combustion air requirement for soot particles is fulfilled by secondary air. Secondary air is the another stage of supply of air to the combustion chamber, it is provided to complete the combustion which is initiated by primary air. It is injected into the combustion chamber of boiler with sufficient pressure to produces turbulence required for proper mixing in combustion and so complete combustion is possible.

5.1.2 Calculation of air required for combustion

Theoretical air required for complete combustion of the coal can be known if the constituents of coal are known. The coal generally consist carbon, hydrogen, sulphur, oxygen and nitrogen. Theoretical air requirement, also called stoichometric air requirement, for complete combustion of 1 kg of coal can be computed as follows.

Carbon burnt to carbon dioxide

Substance	C	$+$	O_2	$=$	CO_2
Atomic or moleculer weight	12		32		44
Weight in Kg	1		2.67		3.67 + 8084 kCal heat release

Thus the oxygen required is 2.67 weight of carbon and the CO_2 produced is 3.67 weight of carbon and the heat release is about 8084 kCal.

Combustion of hydrogen

Substance	$2H_2$	$+$	O_2	$=$	$2H_2O$
Atomic or moleculer weight	3		32		36
Weight in Kg	1		9		9 + 34444 kCal heat release

The oxygen required is 8 times weight of hydrogen and the water vapor produced is 9 times weight of hydrogen. Heat release is around 34444 kCal.

Sulphur burn to sulphur dioxide

Substance	S	$+$	O_2	$=$	SO_2
Atomic or moleculer weight	32		32		64
Weight in Kg	1		1		2 + 1229 kCal heat release

The oxygen required is the same as the sulphur and the sulphur dioxide produced is twice the weight of the sulphur. Heat release is around 1229 kCal .

If considered the above chemical reactions, the total oxygen required for any fuel is :

$$Mass\,of\,O_2\,required = 2.67C + 8H + S - O \tag{5.1}$$

However, there is one complication. Fuel often contains oxygen and available for combustion in just the same way as oxygen from air and so allowance must be made for it. This is done by assuming that all the oxygen in the fuel will combine with necessary amount of hydrogen.

For supplying 1 kg of oxygen 4.32 kg of air is required. Air required for combustion of other constituents is given by:

$$kg\,of\,air\,required = 11.53C + 34.56(H - \frac{1}{8}O) + 4.32\,S \tag{5.2}$$

Where C, H, O and S are carbon, hydrogen, oxygen and sulphur contents present in the coal in percent by weight.

5.1.3 Effect of Air on combustion

Once the theoretical air required is determined the actual air supplied can be adjust to avoid excessive heat losses. If the amount of air supplied is higher than required, it will result in lowering the furnace temperature, increasing the smoke and increase the heat loss in chimney gases and as a result dry flue gas loss increase. When amount of air supplied is lower, it will result in incomplete combustion. To obtain maximum boiler efficiency it is necessary to supply optimum amount of air.

The boiler losses affected due to air supplied are explained below.

5.1.3.1 Dry Flue gas loss

This loss is due to residual thermal energy contained in the dry flue gas when its temperature is too low for further useful work. At this point it is exhausted to atmosphere.

$$Dryfluegasloss = \frac{mC_p(T_f - T_a)}{CV} \times 100 \tag{5.3}$$

This loss constitutes the largest portion of boiler losses and is dependent upon:

- Quantity of dry combustion gases.
- Temperature rise between forced draft fan inlet and gas exit temperature.
- Mean specific heat of the flue gases.

The amount of air supplied over and above the theoretical air, has the greatest bearing upon this loss as this causes both quantity and temperature of the flue to deviate [2].

5.1.3.2 Incomplete Combustion Loss

Loss due to incomplete combustion seldom occurs in modern high capacity pulverized fuel fired boilers under normal operating conditions due to less amount of air supplied for combustion. However, this loss is calculated by the following formula [2].

$$Incompletecombustionloss = \frac{\%CO \times \%C}{\%CO + \%CO_2} \times \frac{5744}{CV} \times 100 \tag{5.4}$$

5.1.3.3 Combined heat loss

If the above losses are added then the result is as shown in figure 5.1. The loss get reduced as optimum level of air supplied is attained, reaches minimum at optimum level and start to increase as more amount of air is added. Thus there is only one quantity of excess air which will give the lowest loss for the combustion of a particular fuel. In other word, there is only one value of air which will give maximum efficiency.

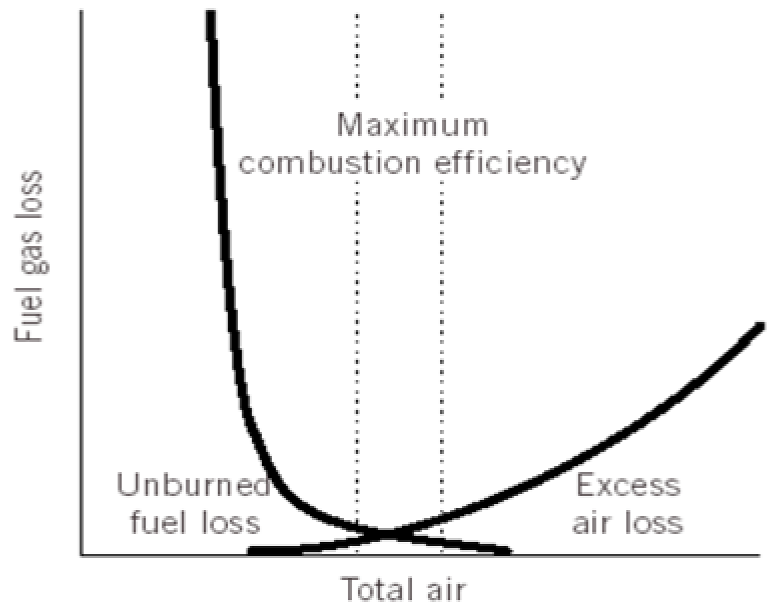

Figure 5.1: Combined heat loss v/s Excess air[5]

5.2 Secondary air damper

As mentioned earlier, Combustion air requirement for soot particles is fulfilled by secondary air. Secondary air fans are used for this, discharge of which is connected to air preheaters, where air temperature is increased to 325°C. These air then flows to wind box, from where it is admitted to the furnace through damper called secondary air dampers. The construction of damper used in WTPS is shown in fig 5.3.

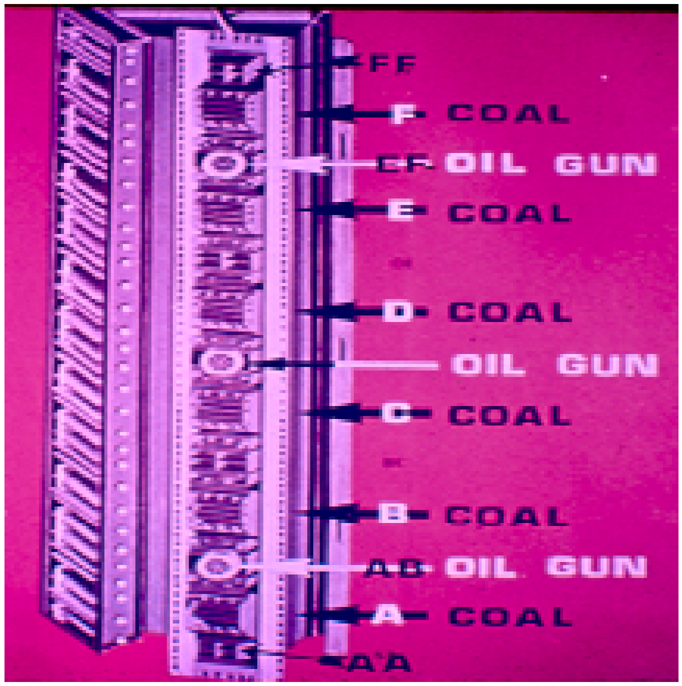

Figure 5.2: Furnace[27]

The secondary air dampers are named after the elevation on the boiler as follows:

Dampers A, B, C, D, E and F are provided at four corner of the furnace.These are also called fuel air dampers. The opening of these dampers is modulated as per mill loading. These dampers supply the secondary air combustion of coal and hence open only for those elevations, which are in service.

Dampers AB, CD and EF are placed between A and B elevation, C and D elevation, E and F elevation respectively. For the oil elevations in service, these dampers modulate as per the oil pressure. For the elevation where oil guns are not in service, these dampers modulate to supply the required quantity of air.

Dampers AA, BC, DE, FF are auxiliary air dampers and placed between B and C elevation, C and D elevation and at bottom and top of furnace which provide the additional air required for combustion. These dampers are manually controlled and supply the additional secondary air which is required for complete combustion.

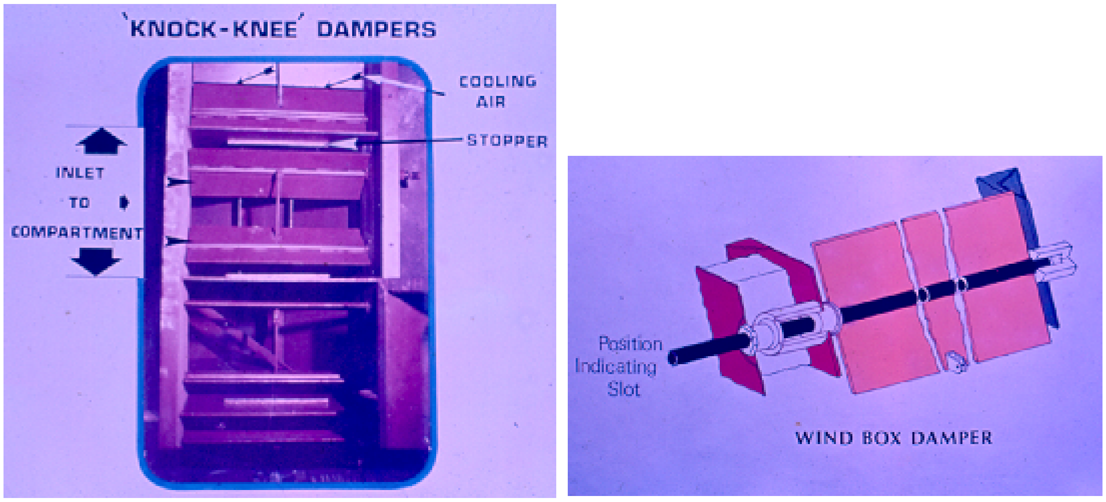

Figure 5.3: Secondary Air damper

5.3 Experimental work

The present work carried out at Wanakbori Thermal Power Station. The analysis is at full plant load. In present work, air dampers were adjusted to supply the correct amount of air for combustion and finding out the gain in boiler efficiency due to correct damper position. The laboratory analyses of coal were carried at chemical laboratory of GSECL Wanakbori thermal power station.

5.3.1 Calculation for Optimum Air

By determining the pecentage change of heat loss with percentage change in quality of excess air supply, optimum quality of excess air supply be found. When heat losses is minimum at that point excess air supply is optimum.The design parameters of boiler are considered to find out the air supplied at which heat losses are minimum. The boiler parameters for the different amount of air supplied and calculation for different heat losses at different amount of air supplied are shown in the appendix V.

5.3.1.1 Dry flue gas loss

Dry flue gas loss basically depends upon the mass supplied and the flue gas temperature. The figure 5.4 shows that as the amount of air supplied increases, the amount of mass supplied and the outlet flue gas temperature increases, dry flue gas increases from 3 -7 %.

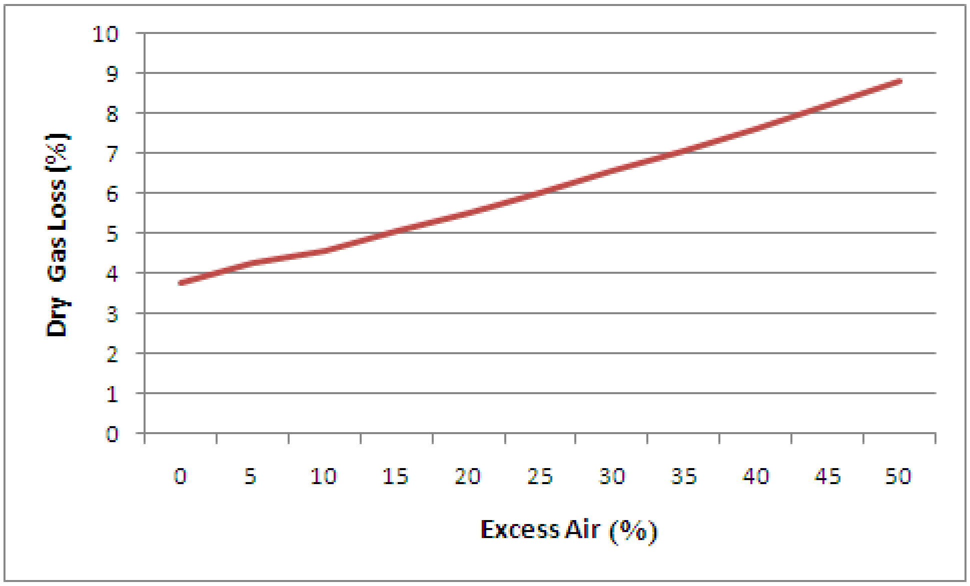

Figure 5.4: Excess air v/s Dry flue gas loss

5.3.1.2 Incomplete combustion loss

The incomplete combustion loss depends upon the formation of CO during the combustion. The lesser amounts of air supplied results in more amount of CO being produced due to incomplete combustion. As shown in figure 5.5, when the amount of air supplied is higher then the optimum value the incomplete combustion loss is very less and reduces at constant rate but where it is below the optimum value of air quantity it increases very sharply.

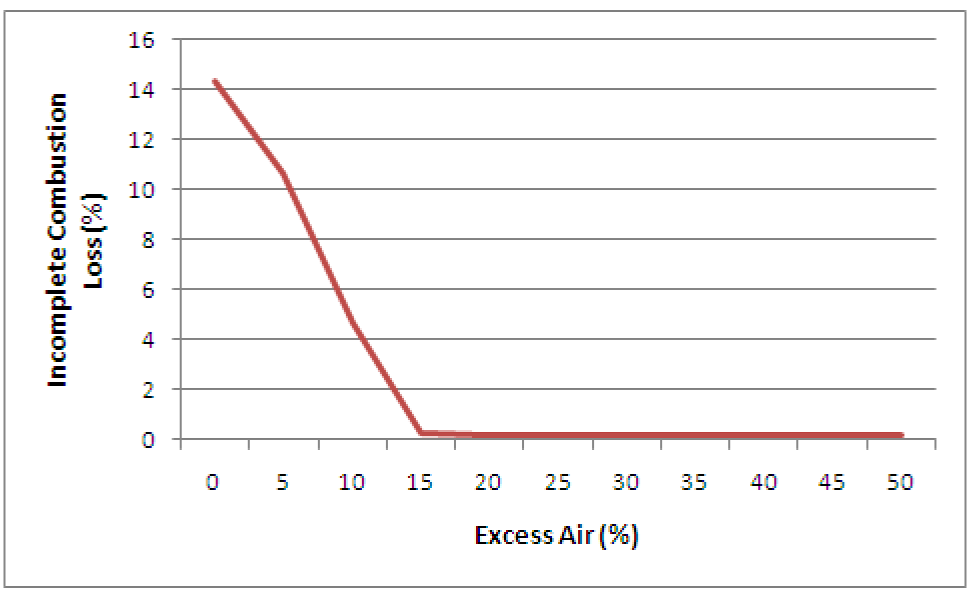

Figure 5.5: Excess air v/s Incomplete combustion loss

5.3.1.3 Combined heat loss

The result of combination of these losses is shown in the figure 5.6. The loss is minimum when the amount of air supplied is 15% excess of its theoretical supplied as shown in figure 5.6 . Higher and lower of that air supplied give higher heat loss. Thus the maximum boiler efficiency for WTPS boiler is obtained when 15% of excess air is supplied.

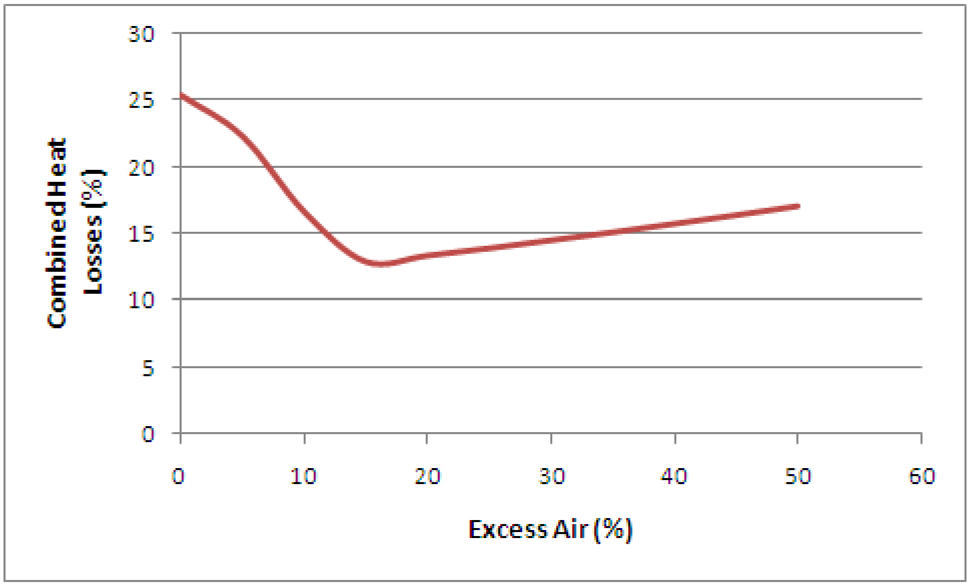

Figure 5.6: Excess air v/s Combined heat loss

5.3.1.4 Effect on O_2 and CO_2

In thermal power plant, the air supplied is basically determined from availability of O_2 and CO_2 in the flue gas. It is shown earlier that the maximum efficiency for WTPS boiler is obtained when the amount of excess air supplied is around 15%. As shown in figure 5.7 and 5.8 the excess air of 15% results in the 2.74% of O_2 and 15% CO_2 in the flue gas. The effect of different supply of air on the presence of O_2 and CO_2 in the flue gas are shown in the figure 5.7 and 5.8. Thus by knowing percentage of O_2 and CO_2 in the flue gas at exit, percentage of excess air is determined.

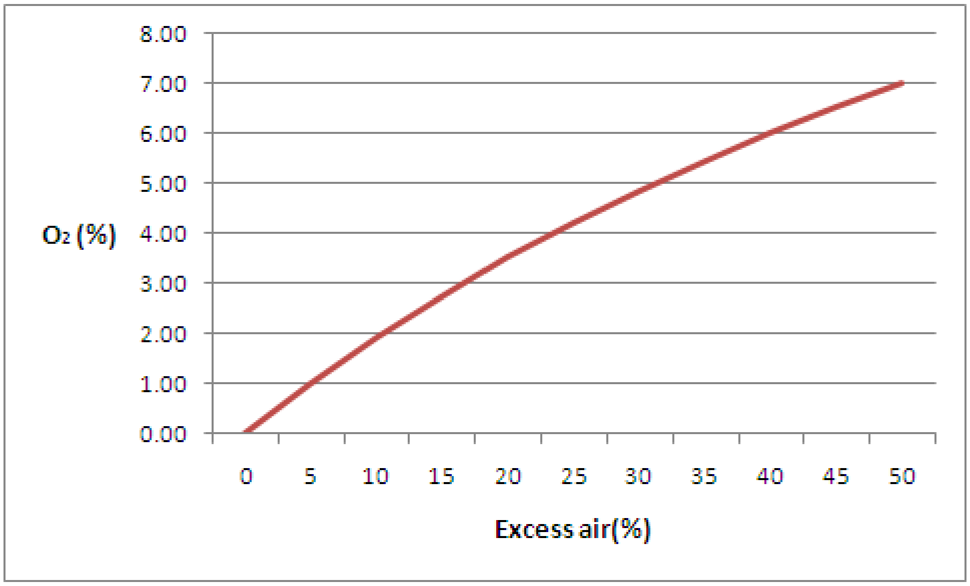

Figure 5.7: Excess Air v/s O_2 in flue gas

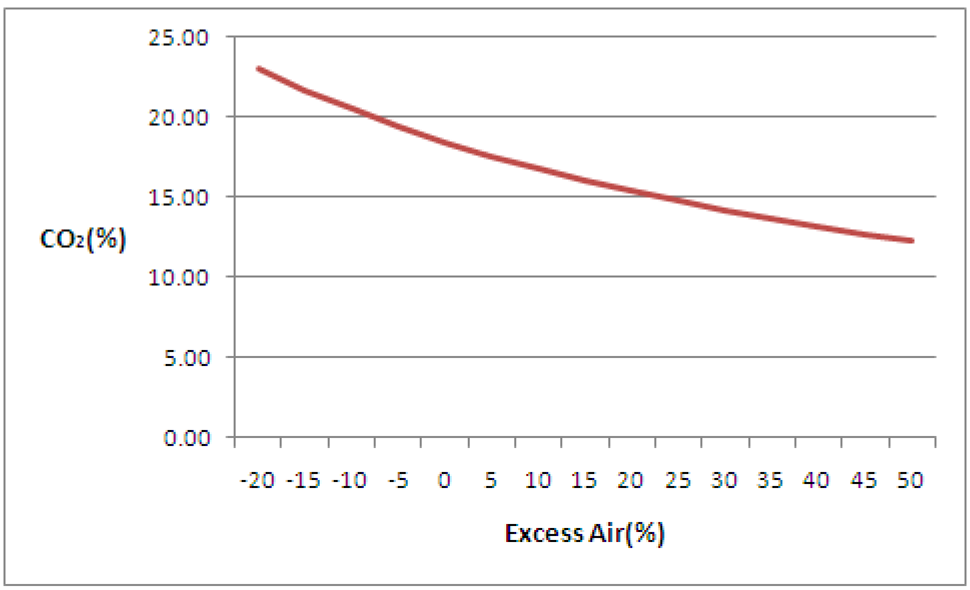

Figure 5.8: Excess air v/s CO_2 in flue gas

The figure 5.7 and 5.8 shows that the amount of oxygen increase continuously and amount of CO_2 reduces as the air supplied is increases.

5.3.2 Boiler efficiency before correcting damper position

5.3.2.1 Coal analysis

The proximate analysis of coal was carried out at WTPS chemical laboratory. The ultimate analysis is obtained using relations of proximate to ultimate conversion. The conversion relation are shown in appendix IV. Table 5.1 shows the composition of coal used.

Table 5.1: Coal analysis

Proximate Analysis	
%Moisture	11.78
%Ash	32.23
%Volatile matter	23.52
%Fixed Carbon	32.47
GCV in Kcal/kg	4071
Ultimate Analysis	
%Carbon	44.57
%Hydrogen	3.09
%Oxygen	5.99
%Sulphur	0.53
%Nitrogrn	1.82
%Moisture	11.78
%Ash	32.23

5.3.2.2 Other operating parameter

The operating parameter are taken when the plant running on full load conditions. The parameters are shown in table5.2.

Table 5.2: Operating parameter

Flue Gas Temp(°C)	179.25
% Oxygen in flue gas	2.30
% CO_2 in flue gas	12.01
Dry bulb temp(°C)	40.00
% CO in flue gas	0.02
Specific Humidity	0.01
% Combustible in bottom ash	2.36
% Combustible in fly ash	0.97
Mill reject (kg/hr)	795.00
GCV of mill reject	1500

5.3.2.3 Boiler efficiency

The boiler efficiency were calculated according to "Heat Loss Method" [2]. The various losses are shown in table 5.3.

Table 5.3: Boiler efficiency before correcting damper position

Losses in Boiler		
1. Dry gas loss	%	6.63
2.Loss due to comb. in ash	%	0.84
3.Loss due to H_2in coal	%	4.41
4.Loss due to moisture in fuel	%	1.87
5.Loss due to sesnsible heat in ash	%	0.48
6.Loss due to moisture in air	%	0.104
7.Loss sue to radiation	%	0.20
8.Mill reject loss	%	0.22
9.Carbon monoxide loss	%	0.13
10.Unaccounted loss	%	0.50
Total losses	%	15.38
Boiler efficiency	%	84.62

5.3.3 Correction in Air supplied

5.3.3.1 Theoretical air required

Theoretical air required for complete combustion can be known if constituent of coal are known. In coal carbon, hydrogen, sulpher, oxygen and nitrogen are present. The air required for complete combustion different quality of coal is calculated by using the following equation.

$$kg\,of\,air\,required = 11.53C + 34.56(H - \frac{1}{8}O) + 4.32\,S \tag{5.5}$$

Air required for the complete combustion of coal used in WTPS is shown in the table 5.4.

Table 5.4: Theoretical air required

Theoretical Air Required (kg/kg of fuel)	6.01

5.3.3.2 Actual air required

The theoretical air required for given quality of coal has obtained from the available coal composition. The theoretical air required is 6.01 kg/kg of fuel. But our optimum excess air analysis shows that maximum efficiency is obtained when excess air supplied is 15%. So in present work, the position of air dampers are set to supply 15% of excess air. The calculation of actual air required to get the maximum boiler efficiency is shown in the table 5.5.

Table 5.5: Actual air required

Actual air required (15% EA) kg/kg of fuel	Total fuel supplied (t/hr)	Total air required (t/hr)
6.91	135	933

5.3.3.3 Correction in air supplied

In Wanakbori thermal power plant, the air is supplied in three different stages. Primary air is supplied as a mixture with coal. Secondary air is divided in two parts: through the fuel air damper and through auxiliary air damper as discussed in section 5.3. To vary the supply of air, the air intake from different stages has to controlled using secondary air damper.

Correction in Primary air supplied

The primary air fan supply the air to the coal mill from which mixture of coal and air supplied into the furnace through burner. Basically primary air is supplied based on the fuel flow but in this power plant the primary air supplied is fixed. Through one coal mill 60 T/hr of air flow is set as constant. The five mills B, C, D, E and F were supplying coal. Thus the total PA flow is shown in table 5.6.

Table 5.6: Primary Air flow

No. of mill in working condition	PA flow from one mill(t/hr)	Total PA flow(t/hr)
5	60	300

Correction in Secondary air supplied

Fuel air damper

The air flow from fuel air damper is maintained using the chart which is shown in figure 5.9. The graph shows the 50% of fuel air damper opening. Total 20 fuel air dampers are open when five mills are in working conditions. The air flow when the damper opening is 50% is shown in the table 5.7.

Table 5.7: Air Flow throgh FA damper

No. of mill in working conditions	SA flow (FAD 50% open) (t/hr)	Total SA flow (t/hr)
5	14.85	297

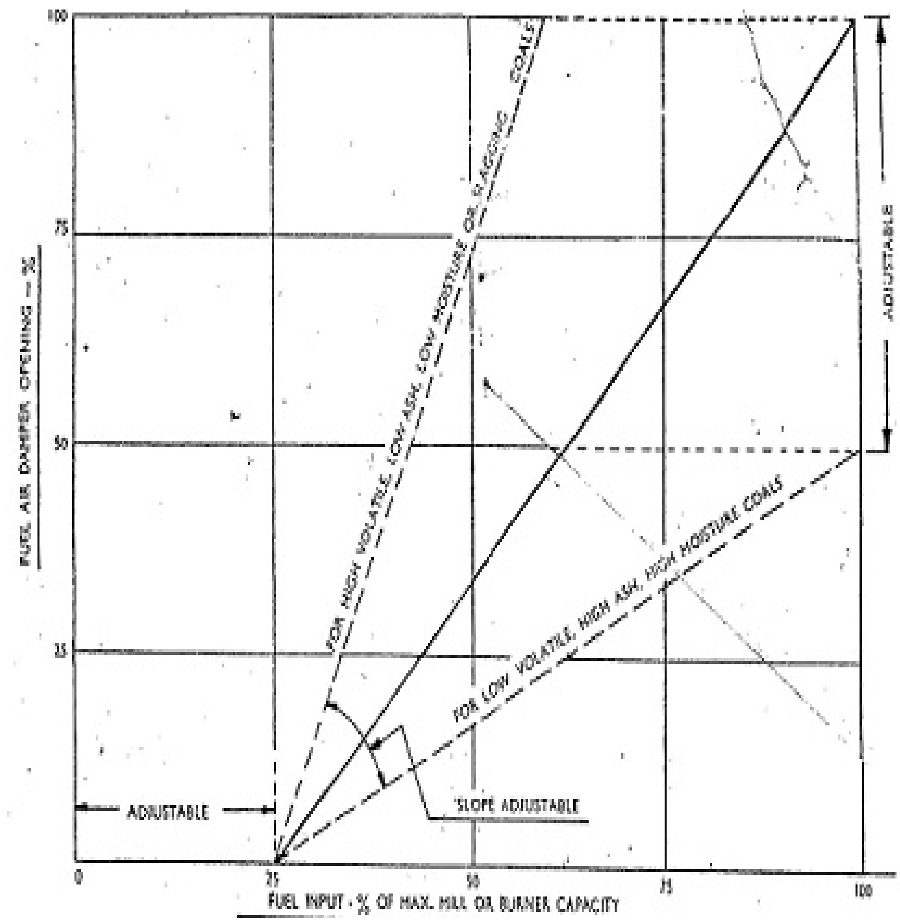

Figure 5.9: Fuel supplied v/s Fuel air damper openinig[27]

Auxiliary air damper

The total air requirement with 15% of excess air is around 933 t/hr. The PA flow is 300 t/hr, SA flow from top/bottom AA/FF dampers is 118 t/hr and SA flow from fuel air damper is around 297 t/hr. The auxiliary air dampers now need to supply 218 t/hr of air. The number of auxiliary air damper is 20 and so to supply the required flow the damper has to open around 37%.

5.4 Result and discussion

The correction in damper opening gives the correct amount of air supplied for maximum efficiency. The fuel air dampers open to 50% which was 20% and the auxiliary air dampers open to 37% which was 15% as opened earlier. The air supplied before correcting damper position is 750 T/hr which increases to 930 T/hr as damper position get corrected. The effect of correcting air supply leads towards complete combustion. Its effect shows on the heat losses.

5.4.1 Boiler efficiency after correcting the damper position

5.4.1.1 Coal analysis

The proximate analysis of coal was carried out. The ultimate analysis is obtained using relations of proximate to ultimate conversion. Table 5.8 shows the composition of coal used.

Table 5.8: Coal Analysis

Proximate Analysis	
%Moisture	11.78
%Ash	32.23
%Volatile matter	23.52
%Fixed Carbon	32.47
GCV in Kcal/kg	4071
Ultimate Analysis	
%Carbon	44.57
%Hydrogen	3.09
%Oxygen	5.99
%Sulphur	0.53
%Nitrogrn	1.82
%Moisture	11.78
%Ash	32.23

5.4.1.2 Other operating parameter

The operating parameter are taken when the plant running on full load conditions. The parameters are shown in table 5.9.

Table 5.9: Operating Parameter

Flue Gas Temp(°C)	167.75
%Oxygen in flue gas	2.78
%CO_2in flue gas	11.50
Dry bulb temp(°C)	40.00
%CO in flue gas	0.02
Specific Humidity	0.01
%Combustible in bottom ash	2.24
%Combustible in fly ash	1.05
Mill reject (kg/hr)	795.00
GCV of mill reject	1500

5.4.1.3 Boiler efficiency

The boiler efficiency were calculated according to “Heat Loss Method”[2]. The various losses are shown in table 5.10.

Table 5.10: Boiler efficiency after correcting air damper

Losses in Boiler		
1. Dry gas loss	%	6.21
2.Loss due to comb. in ash	%	0.81
3.Loss due to H_2in coal	%	4.38
4.Loss due to moisture in fuel	%	1.87
5.Loss due to sesnsible heat in ash	%	0.48
6.Loss due to moisture in air	%	0.09
7.Loss sue to radiation	%	0.20
8.Mill reject loss	%	0.22
9.Carbon monoxide loss	%	0.11
10.Unaccounted loss	%	0.50
Total losses	%	14.40
Boiler efficiency	%	85.60

After correcting the damper position, the boiler efficiency is measured as 85.60%. The gain in efficiency is obtained by comparing it before and after correcting damper position. The improvement in efficiency shown in table 5.11.

Table 5.11: Improved in efficiency

Boiler eff. before correcting damper position (%)	Boiler eff. before correcting damper position (%)	Improved Effi- ciency(%)
84.62	85.60	1.0

As shown in table 5.11, after correcting the position of secondary air damper the gain in efficiency is observed as around 1%.

Chapter 6

Exergy analysis of WTPS Boiler

The exergy method is a relatively new technique uses the concept of exergy which define as the maximum work that can be extracted from the system till it reaches the state of thermodynamic equilibrium with its surrounding. The exergy balance applied to the process gives us information about the potential of work as input and the exergy available with the output. The loss in exergy which is also called irreversibility gives the measure of process unavailability. The concept of exergy is very important in all energy producing, energy consuming and energy conveying systems. The first law provides an energy analysis of a thermodynamic system without making any discrimination of its quality. The second law gives attention towards all forms of energy are not of same quality and the quality of energy always degrades while conserving its quantity. Both first law and second law analysis require to be carried out for any energy conservation or energy conveying system, to make it more exergy efficient which ultimately leads to energy saving or reduced energy consumption for given task.

6.1 Literature review

6.1.1 Literature review

Pradeepsingh hada and Ibrahim Hussain Shah [29] carried out exergetic analysis on 30 MW thermal power plant. The energy and exergy flows in a boiler have been calculated by them. The energy and exergy efficiency also have been calculated. In their work, the energy and exergy efficiency of boiler were found as 84.52% and 33.73% respectively. Irreversibility(exergy destruction) for boiler was also calculated and found as 93MW.

Sarang Gulhane and Amitkumar Thakur [30] shows the exergy analysis of 6 MW captive thermal power plant. In their analysis, the exergy destruction found more at lower load than running at higher load. The increase in 1st law and 2nd law efficiency as the load increase were shown in this paper.

P. Regulagudda et al. [31] carried out thermodynamic analysis of 32 MW coal fired power plant. The developed energy and exergy formulation are shown in this paper. The exergy efficiency for the system was found as 25.38%. The maximum exergy destruction found in boiler.

H. Ravi kulkarni et al. [32] analysed 32 MW coal fired thermal power plant. In their analysis, calculations were made for various power plant components such as boiler, turbine and heat exchangers. In boiler, the loss was calculated around 42% and the turbine had 38% loss. The overall plant efficiency was around 30%.

Yong Li and Lei Liu [33] carried out analysis of 300MW thermal power plant based on second law of thermodynamic and fuel and product concept. The analysis was carried out when power plant running on 75%, 50% and 30% load conditions. The decrease in exergy efficiency when load decrease was shown in their work.

6.1.2 Exergy

Exergy is the maximum amount of useful work that is obtain as the systems are interact to each other to get to the equilibrium. Exergy can be measured as it is leaving of the state of a system relative to that of which is considered the environment. Therefore exergy is a feature of both the system and the environment together. Exergy is generally destroyed and not conserved. Exergy is considered as an extensive property of the system. In the first law, energy can neither be created nor destroyed. The production or consumption of energy is impossible. The second law tells us that the quality of a particular amount of energy, that is, the amount of work, or action that it can do diminishes for each time this energy is used. The usable form of energy in a system is called exergy. Exergy is considered as the total of the free energies in the system. Unlike energy, exergy can be consumed[28].

Exergy is always destroyed and not conserved when work is done; therefore the exergy input is always higher than the exergy output. This imbalance comprises of utilized output which is known as exergy destruction and non-utilized output, which is referred to as exergy loss.

The exergetic efficiency from a thermodynamic view point gives a accurate measurement of the performance of a system. Since exergy is generally destroyed, denoted by irreversibilities in a process, the output is less than the input.

6.1.3 Energy v/s Exergy

Table 6.1: Comparision between Exergy and Energy

Energy	Exergy
Energy is always conserved in a process	Exergy is always conserved in reversible process, but is always consumed in an irreversible process and destroyed by irreversibility
Energy is motion or ability to produce motion	Exergy is work or ability to produce work
Depends upon the state of the systems	Depends upon the state of the system and of surroundings.
Calculated on the basis of reference state.	Reference state is considered as environment
Increase as the temperature increases.	For iso-baric process, it attains its minimum value at t_o ; at lower temperature ; it increase as the temperature drops.
Never depends on pressure.	Depends on pressure.

6.1.4 Second law analysis

6.1.4.1 Limitations of First law

The first law states that when a system undergoes a thermodynamic cycle, then the net heat supplied to the system from the surrounding is equal to the net work done by the system on its surrounding. According to first law heat and work are mutually convertible but since energy can neither ne created nor destroyed the total energy associated with an energy conversion remain constant.

The concept of 1st law efficiency for heat engines or in general for thermodynamics cycle representing the fractional part of the heat supplied to a cycle, which is converted into work.

$$Efficiency = \frac{Desired\,output}{Work\,supplied} \tag{6.1}$$

Therefore the 1st law efficiency is the ratio of quantitative value of desired output to the quantitative value of input used to produce the result. The efficiency of different energy system involving different types of energy input and output cannot be compared directly. Identification of particular energy quantities as output and input gives rise to different definitions for efficiency.

Limitations of first law

- It does not help to predict whether the certain process is possible or not
- Not provide sufficient condition for a certain process to take place
- A spontaneous process can be proceed in a particular direction only but first law does not gives information about to direction
- It establishing equivalence between the amount of heat used and mechanical work

6.1.4.2 Maximum work available to cycle

The maximum work obtainable from a certain heat input in cyclic heat engine is called available energy (exergy) or the available part of energy supplied. The minimum energy that has to be rejected by the second law is called the unavailable energy or the unavailable part of energy supplied[28].

Consider a system working on reversible heat engine as shown in figure 6.1. Engine withdraws heat energy Q from the infinite heat source which is at temperature T and rejects heat Q_o at surrounding which is at constant temperature T_o. The maximum work obtained from the engine is

$$W_{max} = Heat\, supplied \times Maximum\, efficiency \tag{6.2}$$

$$= Q_1 \times (1 - \frac{T_o}{T}) \tag{6.3}$$

$$= Q_1 \times \frac{Q_1}{T} T_o \tag{6.4}$$

$$= Q_1 - T_o dS \tag{6.5}$$

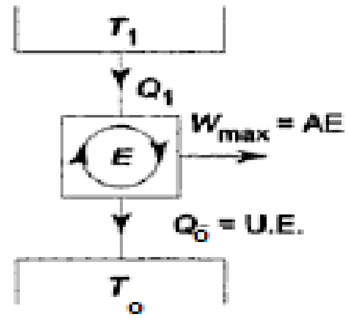

Figure 6.1: Reversible cycle with infinite heat source[28]

W$_{max}$ is the available energy and it is represented on T-S diagram by area x-y-1-2 is shown in figure 6.2. The unavailable energy is represented by area 1-2-3-4. The unavailable energy is the energy rejected from engine and hence it is part of energy that is not converted into work[28].

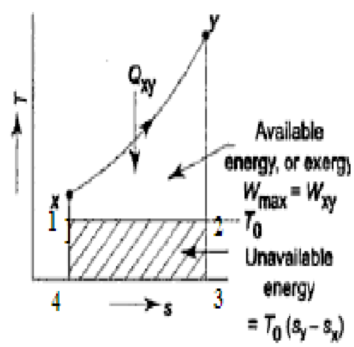

Figure 6.2: T-S diag for reversible cycle[28]

$$Heat\,supplied = Available\,energy + Unavailable\,energy \tag{6.6}$$

$$Q_1 = AE + UE \tag{6.7}$$

$$Q_1 = W_{max} + UE \tag{6.8}$$

$$W_{max} = Q_1 - UE \tag{6.9}$$

From the above equation,

$$Available\,energy\,AE = W_{max} = Q_1 - T_o(s_2 - s_1) \tag{6.10}$$

$$Unavailable\,energy\,UE = T_o dS = T_o(s_2 - s_1) \tag{6.11}$$

6.1.4.3 Irreversibility

The maximum work is possible when the processes are totally reversible and satisfy the condition of reversibility. But we know that there is no perfect reversible process is possible in actual and all the processes in nature are irreversible. Therefore the actual work done by the by the system is always less than theoretical idealized work done due to the presence of irreversibility with the processes. So irreversibility is the difference of reversible maximum work and actual work[28].

Mathematically,

$$I = W_{max} - W \tag{6.12}$$

This is also considered as degradation or dissipation.

6.1.4.4 Forms of exergy

There are four types of exergy[34]

1. Kinetic exergy
2. Potential exergy
3. Physical exergy
4. Chemical exergy

$$Exergy = KE + PE + PhE + CE \tag{6.13}$$

Kinetic and Potential Exergy

The kinetic and potential energies of a system are organised form of energy and so it is completely convertible to work. Therefore, when evaluated the relation to the environmental reference datum levels, they are equal to kinetic and potential energy respectively[34].

Exergy of kinetic energy:

$$x_{ke} = ke = \frac{\vec{V}^2}{2} \tag{6.14}$$

where V is the velocity of the system relative to the environment.

Exergy of potential energy:

$$x_{pe} = pe - gz \tag{6.15}$$

where,

g is the gravitational acceleration

z is the elevation of the system relative to a reference level in the environment.

Physical exergy

Physical exergy is an obtainable work by receiving substance through reversible physical processes from its initial state to the environment.

$$Exp = Ex\triangle T + Ex\triangle P \tag{6.16}$$

Physical exergy = thermal component of physical exergy + pressure components of Physical energy

$$Exph = C_{ph}(T - To) - To[C_p ln(T/To) - Rln(P/Po) \tag{6.17}$$

Physical exergy plays an important role to optimize mechanical and thermal processes.

Chemical exergy

Chemical exergy is the maximum amount of work obtainable when the substance under consideration is brought from the environmental state to the dead state by processes involving heat transfer and exchange of substance only with the environment.

$$E_{xo} = X_i E_{oi} + RT_o X_i \eta X \tag{6.18}$$

6.1.4.5 Exergy balance

The exergy is the maximum useful work that is obtained from a system as it reaches the dead state. Conversely, exergy can be regarded as the minimum work required to bring the system from the dead state to given state. The value of exergy cannot be negative. If a closed system would be able to change its state other than dead state, the system would be able to change its state towards the dead state. This tendency would stop when the dead state is reached. Since any change in state of the close system to the dead state can be accomplished with zero work, the maximum work cannot be negative[28].

While energy is always conserved, exergy is not generally conserved, but it is destroyed by irreversibilities. When a closed system is allowed to undergo a spontaneous change from given state to dead state, its exergy is completely destroyed without producing any useful work. The potential to develop work that exists originally at the given state is thus completely wasted in such a spontaneous process[28].

Therefore, at steady state

1. exergy in = exergy out
2. exergy in – exergy out = exergy destroyed

6.1.4.6 Second law efficiency

A common measure of energy use efficiency is the first law efficiency, η_I. The first law efficiency is the ratio of output energy to input energy of the device. The first law is concerned only with the quantity of energy and disregards the forms in which energy exists. It is the second law of thermodynamics which provides a means of assigning a quality index to energy[28].

With second law, it is possible to analyze the means of minimizing the consumption of exergy to perform a given process, thereby ensuring the most efficient possible conversion of energy for the required task.

The second law efficiency of a process is defined as the ratio of the minimum available energy which must be consumed to do a task divided by the actual amount of exergy consumed in performing the same task[28].

$$\eta_{II} = \frac{Minimum\, exergy intake\, to\, peform\, the\, given\, task}{Actual\, exergy intake\, to\, perform\, the\, samet\, ask} \tag{6.19}$$

$$\eta_{II} = \frac{A_{min}}{A} \tag{6.20}$$

Where A is the availability or exergy.

A power plant converts a fraction of exergy A or $W_{max}$to useful work W. For the desired output of W, A_{min}= W and A = W_{max}.

Here,

I = W_{max}- W and η_{II}= $\frac{W}{W_{max}}$

6.1.5 Exergy formulation for power plant

The thermodmodynamic analysis of thermal power plant contain the balance of mass, energy, entropy and exergy. As earlier specified, the change in kinetic and potential energies will be neglected and steady state flow will be assumed[31].

For a steady state process, the balance of mass for a control volume system can be written as

$$\sum_i \dot{m}_i = \sum_o \dot{m}_o \tag{6.21}$$

The balance of energy for a control volume system is written as

$$\sum_i \dot{E_i} + \dot{Q} = \sum_o \dot{E_o} + \dot{W} \tag{6.22}$$

The balance of entropy for a control volume system is

$$\sum_i \dot{S} + \sum_i \frac{\dot{Q}}{T} + \dot{s_{gen}} = \sum_o \dot{s} + \sum_o \frac{\dot{Q}}{T} \tag{6.23}$$

The balance of exergy for a control volume is written as

$$\sum_i \dot{Ex_i} + \sum_k (1 - \frac{T}{T_k})\dot{Q_k} = \sum_o \dot{Ex_o} + \dot{W} + \dot{Ex_d} \tag{6.24}$$

where stream exergy rate is

$$\dot{E_x} = \dot{m}(e_x) \tag{6.25}$$

$$\dot{m}(e_x) = \dot{m}(e_x^{tm} + e_x^{ch}) \tag{6.26}$$

The exergy balance explianed above is in general form. For combustion process, the heat input also be considered when calculating chemical exergy of coal.

The specific exergy is given by

$$e_x^{tm} = (h - h_o) - T_o(s - s_o) \tag{6.27}$$

When combustion occurs in the boiler, coal is burned to form carbon dioxide, water vapor and other products of combustion. Since the chemical composition of coal changes, in addition to the thermodynamic state, the chemical component of exergy must also be considered. The combustion reaction that describes combustion of coal with the theoretical air component is

$$5.08C + 2H_2 + 6.08(O_2 + 3.76N_2) = 5.08CO_2 + 2H_2O + 22.86N_2 \tag{6.28}$$

The exergy balance equation for the reaction is

$$\sum N_p(\overline{h_{fo}} - \overline{h} - \overline{h_o} - T_o\dot{s})_p = \sum N_r(\overline{h_{fo}} - \overline{h} - \overline{h_o} - T_o\dot{s})_r \tag{6.29}$$

The exergy content of coal for the mass of carbon in coal is written as

$$E_{x,coal} = (E_{x,reaction} \times \eta_c \times m_f)/M_c \tag{6.30}$$

Then, the energy and exergy efficiencies of the power plant are written as

$$\eta_{energy} = \frac{W_{output}}{m_f \times CV} \tag{6.31}$$

$$\eta_{exergy} = \frac{W_{outpot}}{E_{x,coal}} \tag{6.32}$$

6.2 Experimental work

In present analysis, to carry out exergy analysis of Wanakbori thermal power station different parameters are noted at different working conditions. The design parameters are consider at the full load. The parameters of unit 1, 3, 4 and 5 noted when they running at $\frac{3}{4}$ of the full load. The pameters of unit 6 and 7 are taken at the full load condition. The unit 2 was not in running condition because of maintenance work so, not consider for the following analysis.

6.2.1 Coal analysis

Coal generally consist fixed carbon, volatile matter, moisture and ash particles. The ultimate analyses of coal supplied to the all units of WTPS are shown in the table 6.2. The comparison of currently available coal to the coal used during design is also shown.

Table 6.2: Coal Analysis

Coal Composition	Design Parameter	Unit 1	Unit 3	Unit 4	Unit 5	Unit 6	Unit 7
C (%)	51.13	42.97	43.13	42.90	41.80	43.09	43.26
H (%)	3.42	2.98	2.99	2.97	2.89	2.98	3.00
N (%)	1.04	1.83	1.83	1.83	1.83	1.83	1.83
S (%)	0.64	0.50	0.50	0.50	0.50	0.50	0.50
O (%)	7.76	5.65	5.69	5.64	5.30	5.68	5.72
M (%)	8.00	11.03	11.06	11.09	11.04	11.06	11.11
A (%)	28.00	35.04	34.80	35.07	36.64	34.86	34.58

6.2.2 Operating parameter

The other operating parameters required for the exergy analysis of boiler are mass flow rate and temperature of the input and output. The mass flow rate of fuel, water, air supplied, steam and hot product for all units are shown in table 6.3. The temperature of the water, air supplied, steam and hot product are shown in table 6.4.

Table 6.3: Mass flow rate of different supplied

	Mass flow rate(kg/sec)						
	Design Parameter	Unit 1	Unit 3	Unit 4	Unit 5	Unit 6	Unit 7
Fuel	33.33	28.06	28.33	26.94	27.22	37.5	37.22
Water	194.44	146.11	145.27	149.44	153.89	170.10	176.94
Air	244.44	169.44	147.22	144.44	147.22	198.61	198.61
Steam	194.44	132.22	131.66	138.89	141.66	155.55	161.66
Hot Product	277.78	197.50	175.55	171.38	174.44	236.11	235.83

Table 6.4: Temperature of different supplied

		Temperature(K)						
		Design parameter	Unit 1	Unit 3	Unit 4	Unit 5	Unit 6	Unit 7
Water		518	502	499	500	506	515	509
Air	PA	611	537	553	529	502	546	547
	SA	603	533	545	531	546	537	541
Steam		808	808	808	808	808	808	808
Hot Product		419	413	421.5	410.5	417	408	407

6.3 Result and discussion

In this work, exergy analyses of boiler were carried by considering:

1. Exergy of fuel supplied
2. Exergy of water supplied
3. Exergy of air supplied
4. Exergy available with steam
5. Exergy available with hot product

The formulas to find out exergy of these various component is given in Pradeepsingh hada and Ibrahim Hussian Shah [29].

6.3.1 Exergy of Fuel Supplied

The exergy related to fuel is obtained, if fuel composition is available. The composition of coal such as C, H, N, S and O are available with us. The formula for finding out the exergy of fuel is giev in[29]. Exergy of fuel is,

$$\varepsilon_f = 34183.16(C) + 21.95(N) + 11659.9(H) + 18242.90(S) + 13265.90(O)\, kJ/kgK \quad (6.33)$$

Where, C, N, H, O and S are percentage of carbon, nitrogen, hydrogen, oxygen and sulphur available in coal.

The exergy of coal calculated for all units are given in table 6.5. Kotas [34] suggests that the ratio (exergy of fuel / calorific value) should stay between 1.15 and 1.30; in this analysis, such values are between 1.010 -1.020.

Table 6.5: Exergy of Fuel Supplied

		Design parameter	Unit 1	Unit 3	Unit 4	Unit 5	Unit 6	Unit 7
Exergy of Fuel	$\frac{kJ}{kg}$	19023.09	15877.11	15938.27	15850.69	15420.24	15922.11	15978.86
	MW	634.04	529.24	531.28	528.35	514.01	597.08	595.10

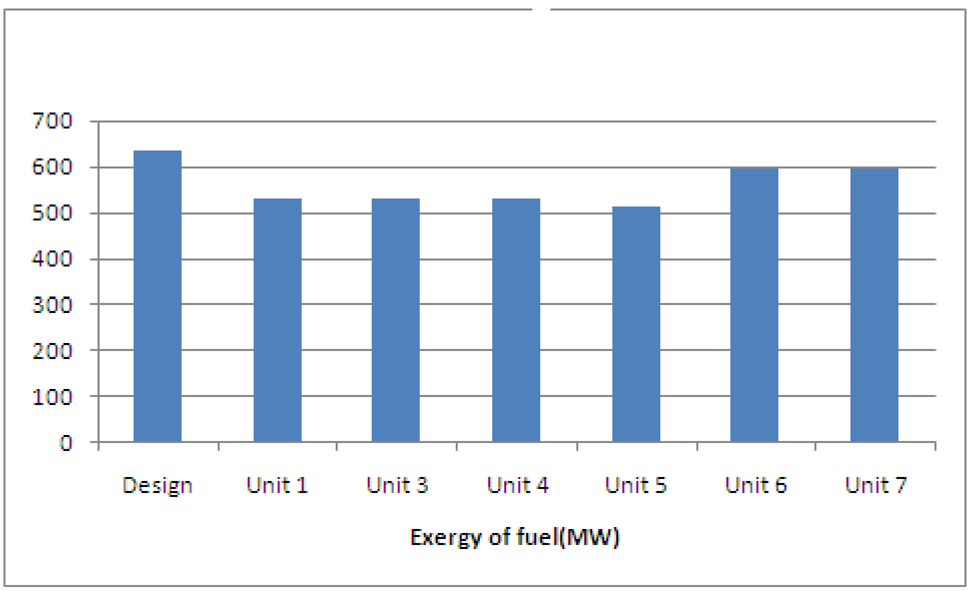

Figure 6.3: Exergy of Fuel Supplied

6.3.2 Exergy of Water Supplied

Exergy of water is given by,

$$\varepsilon_w = (C_p)_w \left[(T_w - T_o) - T_o ln(\frac{T_w}{T_o}) \right] \; kJ/kgK \quad (6.34)$$

T_w	Temperature of feed water
T_o	Reference temperature
$(C_p)_w$	Specific heat of water at constant temperature

The exergy of water calculated is shown in table 6.6.

Table 6.6: Exergy of Water Supplied

		Design parameter	Unit 1	Unit 3	Unit 4	Unit 5	Unit 6	Unit 7
Exergy of Water	kJ/kg	231.298	203.46	198.37	200.57	210.26	225.98	215.481
	MW	44.97	29.73	28.81	29.89	32.36	38.44	38.12

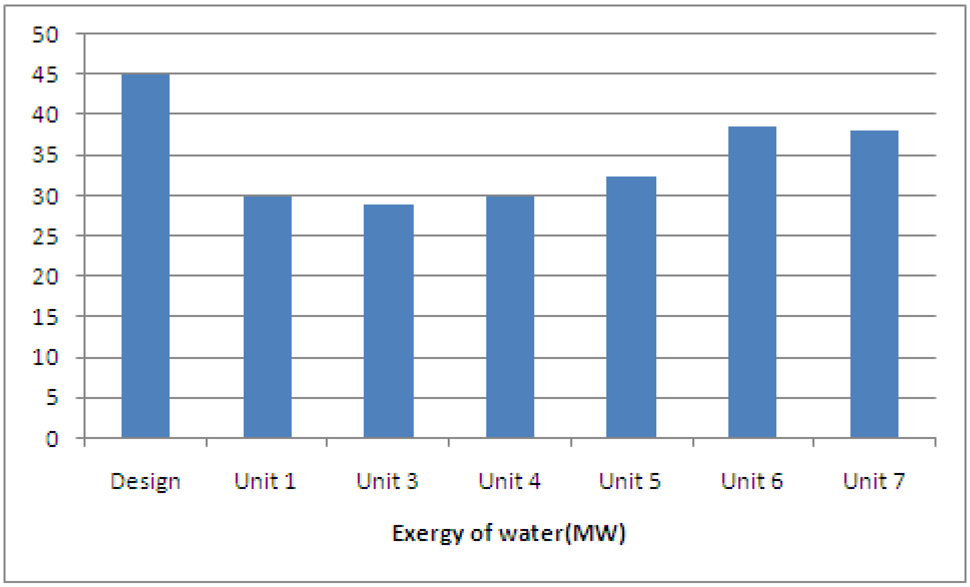

Figure 6.4: Exergy of Water Supplied

6.3.3 Exergy of Air Supplied

The air supplied to the boiler is in two forms: primary air and secondary air. The exergy for both kind of air supplied are calculated. Exergy of air is given by,

$$\varepsilon_a = (C_p)_a \left[(T_a - T_o) - T_o ln(\frac{T_a}{T_o}) \right] \; kJ/kgK \tag{6.35}$$

T_a	Temperature of air
T_o	Reference temperature
$(C_p)_a$	Specific heat of air at constant temperature

The total exergy of air calculated is shown in table6.7.

Table 6.7: Exergy of Air Supplied

		Design parameter	Unit 1	Unit 3	Unit 4	Unit 5	Unit 6	Unit 7
Exergy of Air	kJ/kg	96.24	62.74	69.08	60.78	59.30	65.56	66.76
	MW	23.57	10.63	10.17	8.78	8.73	13.02	13.26

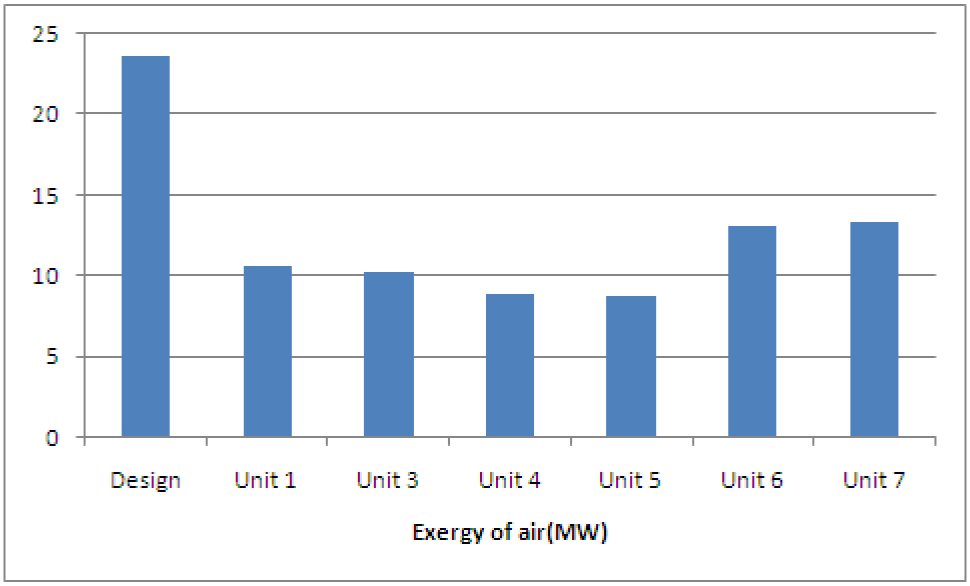

Figure 6.5: Exergy of Air Supplied

6.3.4 Exergy Available with Steam

Exergy of steam is given by,

$$\varepsilon_s = (h - h_o) - T_o(S - S_o) kJ/kgK \tag{6.36}$$

h = Enthalpy of steam
h_o = Enthalpy at reference temperature
s = Entropy of steam
s_o = Enthalpy at reference temperature

The exergy of steam calculated is shown in table 6.8.

Table 6.8: Exergy Available with Steam

		Design parameter	Unit 1	Unit 3	Unit 4	Unit 5	Unit 6	Unit 7
Exergy of Steam	kJ/kg	1429.43	1459.87	1456.64	1483.45	1483.45	1483.45	1483.45
	MW	277.94	193.03	191.79	206.04	210.16	230.76	239.82

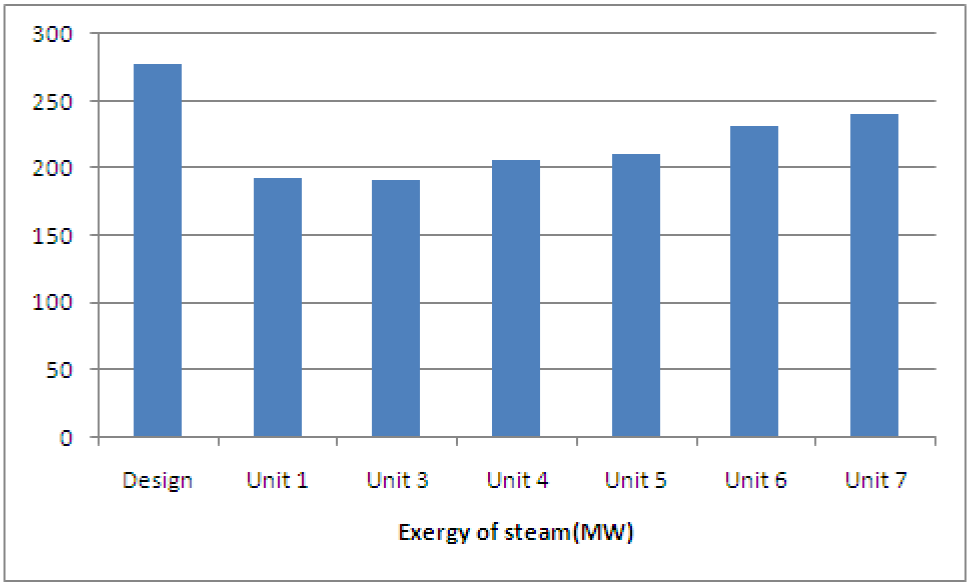

Figure 6.6: Exergy Available with Steam

6.3.5 Exergy Available with Flue Gas

Exergy of hot product is given by,

$$\varepsilon_g = (C_p)_{fg} \left[(T_g - T_o) - T_o ln(\frac{T_g}{T_o}) \right] \; kJ/kgK \tag{6.37}$$

T_g	Temperature of flue gas
T_o	Reference temperature
$(C_p)_{fg}$	Specific heat of flue gas at constant temperature

The exergy of hot product calculated is shown in table 6.9.

Table 6.9: Exergy available with Flue Gas

		Design parameter	Unit 1	Unit 3	Unit 4	Unit 5	Unit 6	Unit 7
Exergy of Flue Gas	kJ/kg	25.69	23.44	26.57	22.53	21.28	21.63	21.28
	MW	7.13	4.63	4.67	3.86	3.71	5.11	5.02

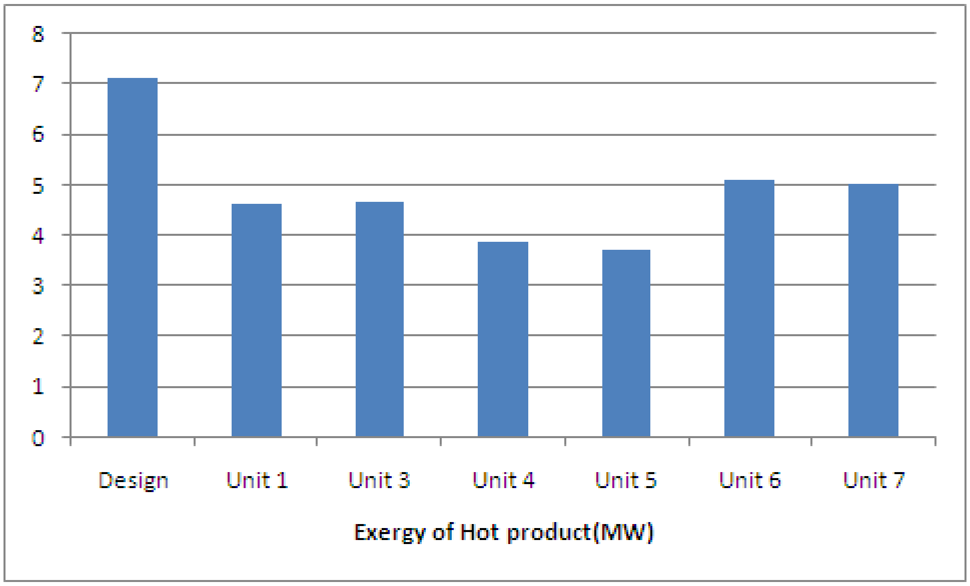

Figure 6.7: Exergy available with Flue Gas

6.3.6 Exergy Efficiency

Second Law efficiency (Exergetic efficiency) of the Boiler is given by

$$\eta_{II} = \frac{Total\,Exergy\,leaving\,the\,boiler}{Total\,Exergy\,entering\,the\,boiler} \tag{6.38}$$

On putting the respected values in above equation, the obtained efficiencies are shown in table 6.10. The comparison of exegetic efficiency are shown in the figure 6.8.

Table 6.10: Exergetic Efficiency

		Design parameter	Unit 1	Unit 3	Unit 4	Unit 5	Unit 6	Unit 7
Exergetic Efficiency	%	40.57	34.70	34.45	37.01	38.53	36.37	37.87

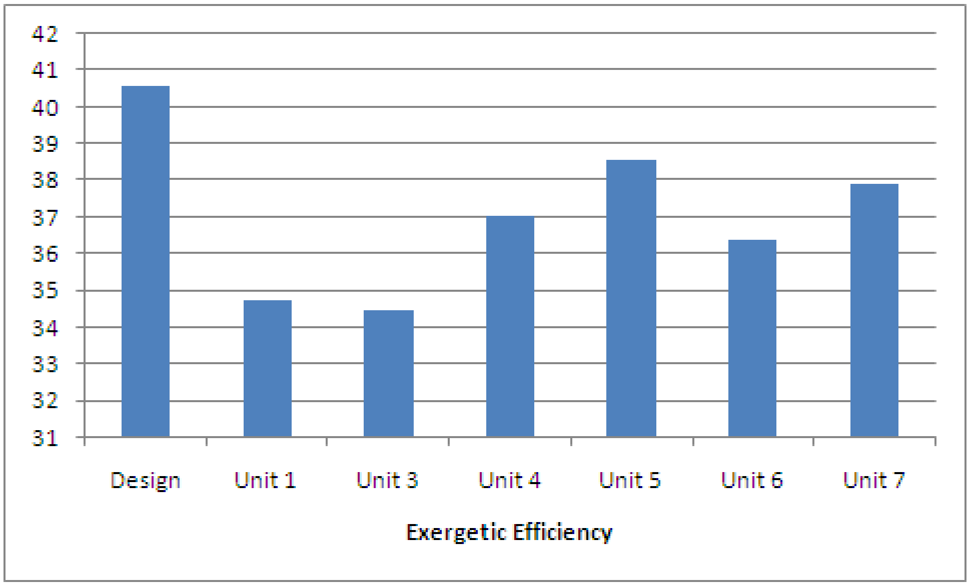

Figure 6.8: Exergetic Efficiency

6.3.7 Total Irriversibilieties in the steam Boiler

Total Irriversibilieties in the steam boiler are given by ,

$$I_b = Total\, exergy\, entering\, the\, boiler - Total\, exergy\, leaving\, the\, boiler \tag{6.39}$$

On putting the respected values in above equation, the obtained irreversibilies are shown in table 6.11. The comparison of irreversibilities are shown in the figure 6.9.

Table 6.11: Irreversibilies

		Design Parameter	Unit 1	Unit 3	Unit 4	Unit 5	Unit 6	Unit 7
Irreversibility	MW	417.51	371.93	373.79	357.17	341.23	412.67	401.64

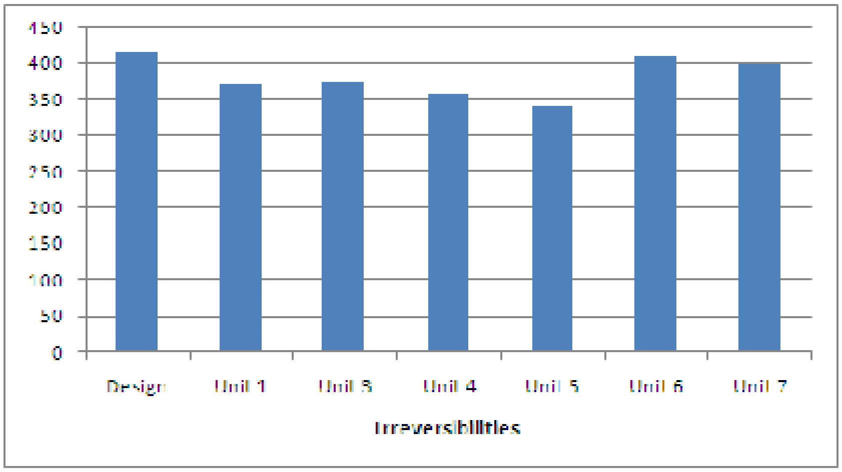

Figure 6.9: Irreversibility

The irriversibilities of the system vary from 358 - 417MW which results in loss of energy.

Chapter 7

Conclusions

7.1 Conclusions

By analyzing the effect of water side deposits on the energy performance of water wall of Boiler unit number 4 of WTPS, the following conclusion has been made:

- Deposit build up in boiler is around 500µm over a 20 year above period.
- The deposit quantity in the boiler tube is between 36.8 – 55.5 mg/cm^2. As the deposit quantity reaches above 40 mg/cm^{2}according to IS 10391:1998, boiler was rated as dirty and immediate cleaning of boiler tube was carried out.
- Reduction in heat transfer in water wall of boiler is observed because presence of additional thermal resistance and reduced tube diameter. The reduction in heat transfer in water wall of boiler is between 2.0-7.3% for different thermal conductivity of deposits.
- Water side deposits in water wall reduce the boiler efficiency around 1.05 %.

By analyzing the performance of the burner tilting device of WTPS Boiler number 5 of 210 MW capacity, the following conclusion has been made:

- After the installation of the burner tilting device the reduction in the reheater spray is around 20.57 t/hr.
- The reduction in loss of energy because of 1t/hr of reheater spray is 0.44 kCal/KWH. The total gain of energy after the reduction of 20.57 t/hr is 9.06 kCal/KWH.
- The reduction in 1kCal/KWH results in the economic loss of 12.80 lacs for Wanakbori thermal power station. Due to gain of energy is 9.06 kCal/KWH, the saving of 115 lacs per annum is possible.
- The reduction in 1kCal/KWH results in the 369.6 MT of CO_2 emission per year for Wanakbori thermal power station. By the gain of 9.06 kCal/KWH reductions in CO_2 emission is 3348 MT per year. This can be trade with other company.

- The power gain from HP cylinder is 1988.72 kW, loss of power from IP and LP cylinder 188.01 kW and net power gain is 1800 kW.
- The economic benefit determined was 350 lacs.

In analysis of maintaining the optimum fuel ratio using secondary air damper control, the following conclusion has been made:

- The supply of air has effect on combustion and so on boiler efficiency. There is only one specific supply of air where complete combustion has been obtained. For WTPS boiler, the maximum efficiency is obtain when the 15% excess air is supplied.
- To maintain the damper opening to supply required optimum air, the damper opening change from 15% to 37%. The flow of air corrected to 930 t/hr from previous 750 t/hr. The boiler efficiency improved 1% because of proper combustion due to sufficient air supply.

In exergy analysis of WTPS boiler, the following conclusion has been made:

- The exergy efficiency found in range of 34.45 - 38.53%. The irreversibilty ranges from 341.23 MW - 412.67 MW. The irreversibility shows that the higher amount useful energy is wasted.

Appendix A

Calculation for U_o

U_o is the overall heat transfer coefficient in clean condition (W m^{-2} K^{-1}).

$$U_O = \frac{1}{\frac{d_O}{d_i}\cdot\frac{1}{h_i}+\frac{d_O\cdot ln(\frac{d_O}{d_i})}{2k}+\frac{1}{h_O}}$$

For the calculation of heat transfer co efficient of water wall tubes under clean condition the following parameters are considered,

$T_{w/s\,in}$ = 563 K
$T_{w/s\,out}$ = 613 K
$T_{fg\,in}$ = 1523 K
$T_{fg\,out}$ = 1123 K
d_i = 57.2 mm
d_o = 63.5 mm

Calculation of h_i

In water wall tubes the water is flow for the heating purpose. The mean temperature of the water is 588 K. The properties of water at this temperature are given below.

μ = 6.26 $\times 10^{-5}$Pa·s
k = 0.363 W/m·K
Pr = 1.23
m_f = 191.63 kg/s

So, here Raynold's number

$$\text{Re} = \frac{4\times m_f}{\frac{N_T}{2}\cdot\pi\mu d_i}$$

$$= 203538.6$$

f = 0.00383

$$\text{Nu} = \frac{(\frac{f}{2})\cdot Re\cdot Pr}{1.07 + 12.7\cdot(\frac{f}{2})^{0.5}(Pr^{\frac{2}{3}} - 1)}$$

$$= 421.31$$

$$\text{Nu} = \frac{h_i \cdot d_i}{k}$$

$$\text{h}_i = 2673.69 \text{ W/m}^2\text{K}$$

Calculation of h$_o$

At the outer surface of water wall tube, the exhaust gases are flowing. The mean temperature of the exhaust gases are 1323 K. The properties of gases at this temperature are

ρ = 0.2715
μ = 4.98 $\times$ 10^{-5} Pa·s
k = 0.0849 W/m·K

$$\text{Reynold's no Re} = \frac{\rho v d_o}{\mu}$$

$$= 4154.28$$

$$\text{Nu} = \frac{(\frac{f}{2})\cdot Re\cdot Pr}{1.07 + 12.7\cdot(\frac{f}{2})^{0.5}(Pr^{\frac{2}{3}} - 1)}$$

$$= 16.97$$

$$\text{Nu} = \frac{h_o \cdot d_o}{k}$$

$$\text{h}_o = 22.68 \text{ W/m}^2\text{K}$$

The calculation of U$_o$is given below

$$\text{U}_O = \frac{1}{\frac{d_O}{d_i}\cdot\frac{1}{h_i} + \frac{d_O \cdot ln(\frac{d_O}{d_i})}{2k} + \frac{1}{h_O}}$$

$$\text{U}_O = 23 \text{ W/m}^2\text{K}$$

Appendix B

Design parameter for boiler

Table B.1: Coal Analysis

Coal Analysis				
Proximate analysis			Ultimate analysis	
% Moisture	8.00		% Carbon	51.13
% Ash	28.00		% Hydrogen	3.42
% Fixed Carbon	25.00		% Oxygen	7.76
% Volatile Matter	25.00		% Sulphur	0.64
GCV in kCal/kg	4440		% Nitrogrn	1.04
			% Moisture	8.00
			% Ash	28.00

Table B.2: Parameter

Parameter	
Flue gas temperature (^oC)	146.00
% O_2in flue gas	3.50
% CO_2in flue gas	13.70
Dry bulb temperature(^oC)	38.00
% CO in flue gas	0.02
% Combustible in bottom ash	5.60
% Combustible in fly ash	1.50

Appendix C

Calculation of HR_s

Data of main steam

Flow Q_S	= 690 T/hr
Pressure P_O	= 154.6 kg/cm^2
Temp. T_O	= 540°C
Enthalpy h_O	= 3419.6 kJ/kg
Entropy s	= 6.482 kJ/kg k

Data of R/H inlet

Pressure P_{ri}	= 37.56 kg/cm^2
Temperature T_{ri}	= 336.9°C
Enthalpy H_{ri}	= 3069.8 kJ/kg

Data of R/H outlet

Pressure P_{ro}	= 36.14 kg/cm^2
Temperature T_{ro}	= 540°C
Enthalpy h_{ro}	= 3540.3 kJ/kg
Entropy s_{ro}	= 7.263 kJ/kg k

Data of final feed water

Pressure P_f	= 170.3 kg/cm^2
Temperature T_f	= 241.6°C
Enthalpy h_f	= 1047.0 kJ/kg
Entropy s_f	= 2.689 kJ/kg k

Data of spray water

Pressure P_s	= 49 kg/cm^2
Temperature T_s	= 241.6°C
Enthalpy h_s	= 1045.4 kJ/kg
Entropy s_s	= 2.7136 kJ/kg k

Additional data

Condensate temperature T_c	= 46°C
Design heat rate HR	= 1988.81 kcal/KWH
	= 8327.147 kcal/KWH

Ideal cycle efficiency(Without spray)

$$\text{Ideal cycle efficiency} = 1 - \frac{T_C(s_{ro}-s_F)}{(h_O-h_f)+(h_{ro}-h_{ri})}$$

$$= 0.4867$$

$$= 48.67\ \%$$

Ideal cycle efficiency(With spray)

$$\text{Ideal cycle efficiency} = 1 - \frac{T_C(s_{ro}-s_s)}{(h_{ro}-h_s)}$$

$$= 0.4217$$

$$= 42.17\ \%$$

Actual cycle efficiency(With spray)

$$\text{Actual cycle efficiency} = \text{Turbine design efficiency} \times \text{Ideal cycle efficiency(with spray)}$$

$$\text{Turbine design efficiency} = \frac{Actual\, efficiency}{Ideal\, efficiency}$$

$$\text{Actual efficiency} = \frac{3600}{8327.147}$$

$$= 43.23\ \%$$

So, Turbine design efficiency = 88.84 %

So, actual cycle efficiency = 37.50%

Actual heat rate

$$\text{Actual heat rate(HR}_s) = \frac{3600}{37.50} \times 100$$

$$= 9600\ \text{kcal/KWH}$$

Appendix D

Ultimate to Proximate conversion

The relationship between ultimate to proximate conversion is given below.

$$
\begin{aligned}
\%C &= 0.97C + 0.7(VM - 0.1A) - M(0.6 - 0.01M) \\
\%H &= 0.036C + 0.086(VM - 0.1A) - 0.0035M^2(1 - 0.02M) \\
\%N &= 2.10 - 0.020\ VM
\end{aligned}
$$

Where	C	=	% of fixed carbon
	A	=	% of ash
	VM	=	% of volatile matter
	M	=	% of moisture

Appendix E

Boiler efficiency at different air supply

Table E.1: Parameter for different excess air supply

Parameter	Excess air										
	0	5	10	15	20	25	30	35	40	45	50
$\%O_2$in Flue gas	0.00	1.00	1.91	2.74	3.50	4.20	4.85	5.44	6.00	6.52	7.00
$\%CO_2$in flue gas	18.47	17.59	16.79	16.06	15.39	14.78	14.21	13.68	13.19	12.74	12.31
Exhaust temp.	125	131	135	141	146	151	157	162	168	173	179
Sp. Humidity	0.02	0.02	0.02	0.02	0.02	0.02	0.02	0.02	0.02	0.02	0.02
DBT	38	38	38	38	38	38	38	38	38	38	38

Table E.2: Heat losses for different excess air supply

Heat losses	Excess air										
	0	5	10	15	20	25	30	35	40	45	50
Dry gas loss	3.75	4.24	4.56	5.05	5.52	6.02	6.54	7.08	7.63	8.21	8.81
Combustible in ash	1.11	1.14	1.15	1.19	1.22	1.25	1.29	1.31	1.36	1.40	1.43
H_2in coal	4.30	4.33	4.34	4.36	4.38	4.39	4.41	4.43	4.45	4.46	4.48
Moisture in fuel	1.120	1.126	1.13	1.134	1.139	1.143	1.148	1.152	1.157	1.160	1.165
Moisture in air	0.11	0.13	0.14	0.16	0.17	0.19	0.21	0.22	0.24	0.26	0.28
Incomplete combus-tion	14.31	10.67	4.65	0.31	0.23	0.22	0.21	0.21	0.20	0.20	0.20
Radiation loss	0.2	0.2	0.2	0.2	0.2	0.2	0.2	0.2	0.2	0.2	0.2
unaccounted loss	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5

Table E.3: Boiler efficiency for different excess air supply

Boiler Eff. at diff air supplied		
Excess air	Total Heat loss	Boiler eff.
0	25.40	74.59
5	22.34	77.66
10	16.67	83.32
15	12.90	87.09
20	13.36	86.63
25	13.91	86.08
30	14.52	85.48
35	15.11	84.88
40	15.75	84.24
45	16.40	83.59
50	17.07	82.92

Bibliography

[1] Annual report by Cenral Electrical Authority of India.

[2] “Energy Efficiency Booklet”, by Bureau of Energy Efficiency.

[3] Energyefficiencyasia.com

[4] Spiraxsarco.com

[5] R. Yadav, “Power Plant Engineering”, Wiley Publication.

[6] khi.co.jp

[7] watertube boiler/en.wilipedia.org

[8] “Power plant manual” by National Thermal Power Corporation

[9] M. Bhatt, “Effect of water side deposits on the energy performance of the power plant”, Energy Conversion and management 47 (2006) 1247-1263

[10] IS 10391:1998

[11] Quan Zhenhua et al., “Experimental Study of Fouling on Heat Transfer Surface During Forced Convective Heat Transfer”, Chinese Journal of Chemical Engineering, 16(4) 535—540 (2008)

[12] Mohamed A. Antar, Syed M. Zubair , “The impact of fouling on performance evaluation of multi-zone feedwater heaters”, Applied Thermal Engineering 27 (2007) 2505–2513.

[13] XU Zhi-Ming et al., “Costs Due to Utility Fouling in China”, ECI Symposium Series, Volume RP5, 2007.

[14] Ahmet Fertelli and Ertan Buyruk, “ Numerical analysis of heat transfer characteristics for deposit formation shapes around single cylinder ”, Journal of scientific and industrial research, Vol 66, 2007.

[15] A. J. Karabelas, et al., “liquid-side fouling of heat exchangers an integrated r & d approach for conventional and novel designs”, Applied Tiwrmd Engineering Vol. 17. Nos. 8-10, pp. 127-737. 1997.

[16] Dr. Pradeep Jain -DGM, NETRA- R&D, NTPC Limited, "how to increase heat transfer and reduce waterwall tube failure in high pressure boilers"

[17] "RLA Report", by BHEL Tiruchirapalli.

[18] Edwin Lim Chui Seng, "Evalution of High Temperature Boiler Tubes using Itrative Analytical Approach", Faculty of Engineering and Science University Tanku Abdul Rahman, April 2013

[19] Ravindra Kumar P. et al., "Effect of Parameters in Once-Through Boiler for Controlling Reheat Steam Temperature in Supercritical Power Plant", Research Journal of Engineering Sciences, ISSN 2278 – 9472 Vol. 2(1), 27-34, January (2013).

[20] T K Seal et al., "boiler steam temperature control – optimizing efficiency and lifetime", NTPC-Rihand.

[21] Wang Yan-jun et al. " Analysis of De-superheating Spray System Based on Thermal Economic State Equation[J]", Electric Power Science and Engineering, 2007-01.

[22] Yin Jie and Wang Pei-hong, "Influence of different spray desuperheating modes on the thermal economy of unit in power plant", East China Electric Power, 2009-10.

[23] T.M. Paes, "System Identification in Power Plants with Application to LQG Steam Temperature Control Design", DCT, 2006.

[24] David Lindesley, "Power Plant Control and Instrumentation the control of boilers and HRSG systems", IET Control Engineering Series 58.

[25] Power Plant Performance by A. B. Gill.

[26] Chapter 8: Section 3-4, EGR 334 Thermodynamics.

[27] BHEL Manual for WTPS

[28] P. K. Nag, "Basic Engineering Thermodynamic", Tata Mcgraw-Hill Publishing Company limited.

[29] Pradeep Singh Hada, Ibrahim Hussain Shah, "First Law and Second Law Analysis of a Lignite Fired Boiler Used in a 30 MWe Thermal Power Plant", International Journal of Engineering and Innovative Technology (IJEIT) Volume 1, Issue 6, ISSN: 2277-3754, June 2012.

[30] Sarang j gulhane, Prof.Amit kumar thakur, "Exergy Analysis of Boiler In cogeneration Thermal Power Plant", American Journal of Engineering Research (AJER) Volume-02, Issue-10, pp-385-392,2013.

[31] P. Regulagadda et al., “Exergy analysis of a thermal power plant with measured boiler and turbine losses”, Applied Thermal Engineering 30, pg-970–976, 2010.

[32] H Ravi Kulkarni et al., “Energy and Exergy Analysis of Coal Fired Power Plant”, International Journal of Innovative Research in Technology & Science(IJIRTS), ISSN:2321-1156.

[33] Yong Li, Lei Liu, “Exergy Analysis of 300MW Coal-Fired Power Plant”, Energy Procedia 17, pg- 926 – 932, 2012.

[34] Kotas, “The Exergy Method of Thermal Plant Analysis”, Kreiger Publishing Company Malabar.