Effects of Air Ingress on Performance of Boiler and Air-Preheater of Thermal Power Station

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Major Project Report

Submitted in partial fulfillment of the requirements

For the Degree of

Master of Technology in Mechanical Engineering

(Thermal Engineering)

By

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May 2015

Declaration

This is to certify that

1. The thesis comprises of my original work towards the degree of Master of Technology in Mechanical Engineering (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.

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Abstract

Ever increasing imbalance between supply and demand of energy (especially electrical) existing not only in our country but worldwide, it is inevitable for developing countries like India to focus on increasing the generation capacity and energy conservation measures. As far as thermal power plants are concerned, there is huge potential of energy saving and increase in energy efficiency by minimizing losses and upgradation of technology. Compared to installation of new power generation unit which requires investment of several crores of rupees per MW set up, energy conservation is very effective and quick solution which requires very little investments. In thermal power plants, overall plant efficiency is dependent on multiple factors like boiler performance, auxiliary power consumptions, coal quality, turbine, condenser and generator efficiency, etc. In the present work, effects of air ingress on performance of boiler and air preheater have been investigated. Air ingress in boiler and air preheater results into the high auxiliary power consumption which ultimately leads to lower power generation capacity and increase in the heat rate. Air preheater leakages must be checked periodically and minimized to prevent the under performance of the unit resulting from overloading of fans. The objective of this work is to quantify air ingress & its effects on the energy efficiency of boiler and air preheater performance. In the present work air ingress sites have been identified by visual inspection during capital overhauling on unit-5. Also air ingress volume between different components of unit-5 of GSECL thermal power station at Wanakbori have been calculated using O_2 mapping. The identified sites of air ingress have been rectified as well as internal leakages within air preheater has been arrested by replacing of various parts. Economic analysis before and after overhauling has been carried out to quantify saving in cost of fuel and fan power, which results into saving of crores of rupees. Further, recent advancements in air preheater technologies are described here which could be implemented to upgrade the existing air preheaters. Exergy analysis which provides deep insight of qualitative energy loss in the systems have been carried out for air preheater and boiler of units 4, 5 and 6. Second law efficiency of air preheater of units 4, 5 and 6 is found to be 55.18%, 57.55% and 51.96% respectively and that of boiler of units 4, 5 and 6 is 44.26%, 45.32% and 42.84% respectively.

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Nomenclature

T	temperature,(°C)
V	velocity, (m/s)
L	length, (m)
R_h	hydraulic radius, (m)
f	friction factor
C_p	specific heat, (kcal/kg°c)
m	mass flow rate, (kg/s)
h	head loss, (mmwc)
$Ntu_{,0}$	modified number of transfer units (-)
C^*	heat capacity rate ratio (C_{min}/C_{max})
Cr^*	total matrix heat capacity rate ratio (C_r/C_{min})
ε	exergy
η	efficiency
s	entropy (kJ/kgK)
h	enthalpy (kJ/kg)
I	irreversibility (kJ/kg)
W	work output (MW)

Subscripts

t	total head loss
b	fuel bed resistance
v	velocity head loss
e	equipment
d	ducts and chimney
a	air
g	gas
w	water
0	reference
gl	gas leaving
ge	gas entering
ae	air entering
al	air leaving
gnl	gas corrected for no leakage
i	in
o	out
I	first law
II	second law
f	fuel
s	steam
tm	thermo-mechanical
ch	chemical

Abbreviations

APH	Air Preheater
TPS	Thermal Power Station
WTPS	Wanakbori Thermal Power Station
GSECL	Gujarat State Electricity Corporation Limited
PAT	Perform Achieve and Trade
BEE	Bureau of Energy Efficiency
CEA	Central Electricity Authority
IGEN	Indo-German Energy Programme
DC	Designated Consumers
ESCerts	Energy Savings Certificate
RAPH	Rotary Air Preheater
ID	Induced Draft
PA	Primary Air
SA	Secondary Air
FD	Forced Draft
ESP	Electro-Static Precipitator
HR	Heat Rate
GHR	Gross Heat Rate
SH	Superheater
DU	Double Undulated
FNC	Flat Notched Crossed
ASU	Air Separation Unit
CPU	Compression and Purification Unit
SCAPH	Steam Coil Air Preheater
DP	Differential Pressure
GSE	Gas Side Efficiency
AL	Air Leakage
FGT	Flue Gas Temperature
ECO	Economiser
I/L	Inlet
O/L	Outlet
CV	Calorific Value
A	Fraction of exergy

Chapter 1

Introduction

This chapter contains brief introduction of boiler and its various accessories such as economizer, air preheater, electrostatic precipitator. A brief introduction on capacities and commissioning of different units of Wanakbori TPS. Moreover, motivation, definition and objective of the present study are also included in this chapter.

1.1 Boiler & Its Accessories

A boiler in coal (fossil fuel) based thermal power plant is a steam generator that includes an economizer, a steam drum, and the furnace with its steam generating tubes and superheater coils. Necessary safety valves are located at suitable points to avoid excessive boiler pressure. The air and flue gas path includes: forced draft (FD) fans, air preheater (APH), boiler furnace, induced draft (ID) fans, fly ash collectors (electrostatic precipitator) and the flue gas stack. [1]

1.1.1 Economizer

The function of an economizer in a steam generating unit is to absorb heat from the flue gases and add this as sensible heat to the feed water before the water enters the evaporative circuit of the boiler.Earlier the economizers were introduced mainly to recover the heat available in flue gas that leaves the boiler and provision of this additional heating surface increased the efficiency of steam generation, saving in fuel consumption, thus the name " Economizer"

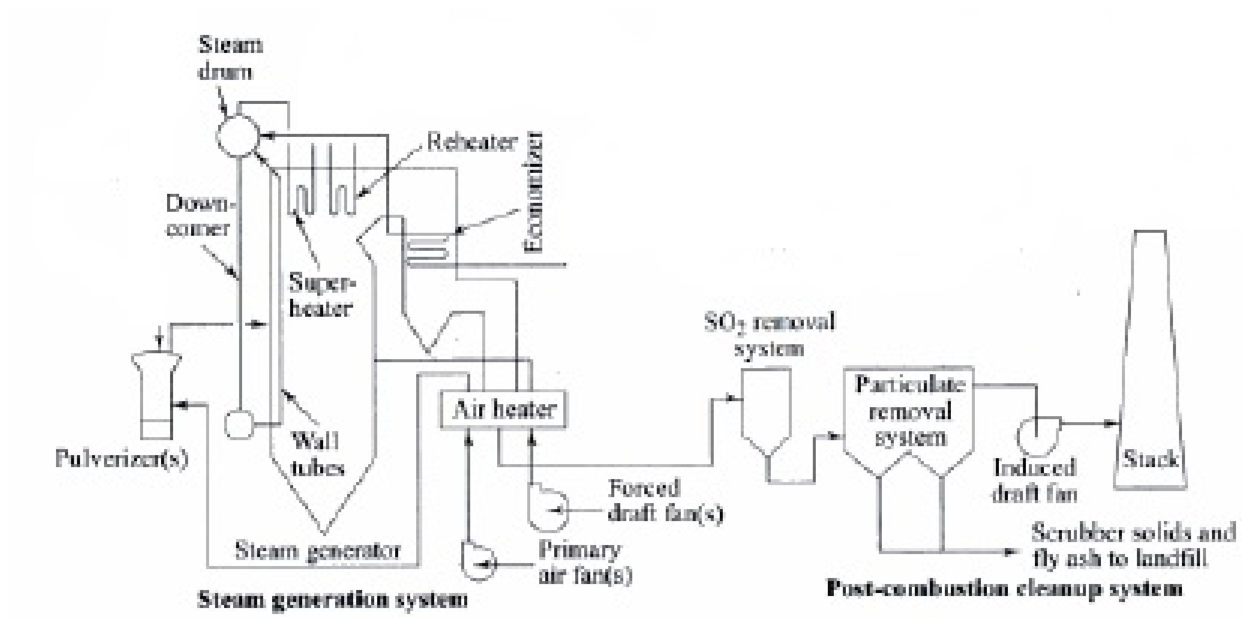


Figure 1.1: Schematic layout of boiler [2]

christened. In the modern boilers used for power generation feed water heaters were used to increase the efficiency of the unit and feed water temperature and hence the relative size of economizer is less than earlier units. This is a good proposition as the heat available in boiler exit flue gas can be economically recovered using air preheater which is essential for pulverised fuel fired boilers. In this a large number of small diameter thin walled tubes are placed between two headers. Feed water enters the tube through one header and leaves through the other. The flue gases flow out side the tubes usually in counter flow.The use of economizer results in saving in coal consumption , increase in steaming rate and high boiler efficiency but needs extra investment and increase in maintenance costs and floor area required for the plant. [1]

1.1.2 Air Preheater

An air preheater is a device designed to heat air before combustion in a boiler with the primary objective of increasing the thermal efficiency of the process. The purpose of the air preheater is to recover the heat from the boiler flue gas which increases the thermal efficiency of the boiler by reducing the useful heat lost in the flue gas. As a consequence, the flue gases are also conveyed to the chimney at a lower temperature allowing simple design of ducts and chimney. Air preheaters are paramount in maintaining a highly efficient power plant. Such systems provide heat recovery to the unit by cooling the flue gas counter-currently with cool incoming pre-combustion air. Cooling of the flue gas transfers the heat that is necessary both for coal drying and overall boiler efficiency. The air preheater is located downstream (flue gas path) of the economizer. A tri sector rotary air preheater which is mainly used for coal with high moisture content is shown in figure 1.3. The materials used in the air preheaters are of carbon steel at high temperature sections and corten steel for the cold end section.[3]

- Types of Air Preheaters

Air preheaters can be classified as Recuperative and Regenerative types based on their operating principle.

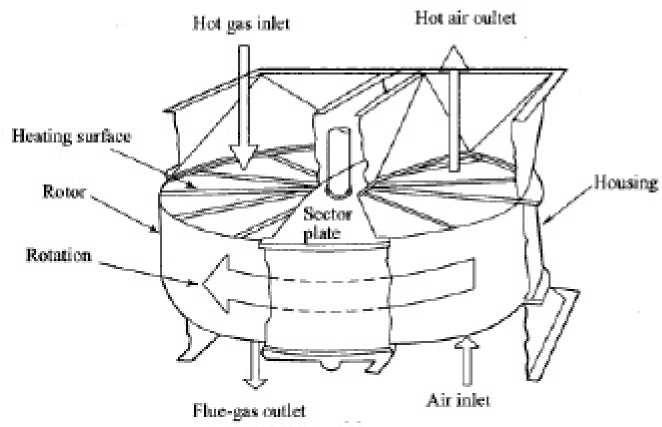


Figure 1.3: Regenerative air preheater[2]

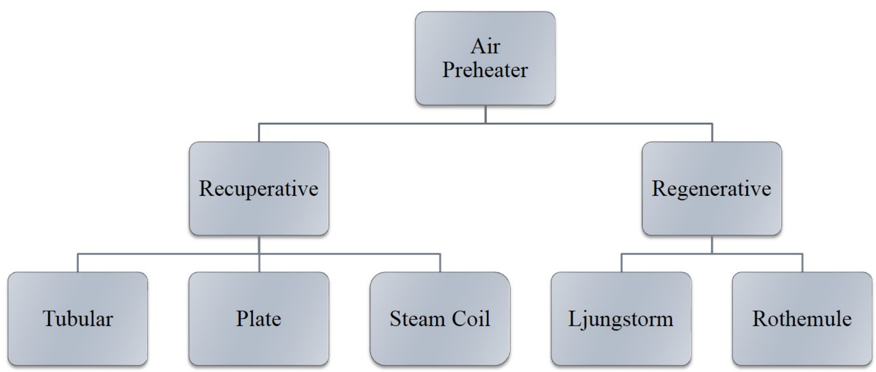


Figure 1.2: Types of Air Preheaters

In Recuperative type heating medium is on one side and air is on the other side of tube or plate and the heat transfer is by conduction through the material which separates the media. These are of static construction and hence there is only nominal leakage through expansion joints, access doors, casings, etc.

In Regenerative type the heating medium flows through a closely packed matrix to raise its temperature and then air is passed through the matrix to pick-up the heat. Either the matrix or the hoods are rotated to achieve this and hence there is slight leakage through sealing arrangements at the moving surfaces.[1]

1.1.3 Electrostatic Precipitator

An electrostatic precipitator (ESP) is a particulate collection device that removes particles from a flowing gas using the force of an induced electrostatic charge. Electrostatic precipitators are highly efficient filtration devices that minimally obstruct the flow of gases through the device, and can easily remove fine particulate matter such as dust and smoke from the air stream. Introduction of ESP in thermal power plants is mainly because of the high ash content in the flue gas because their is 40% ash content in Indian bituminous coal.[1]

1.2 About Wanakbori TPS

Wanakbori Thermal Power Station (WTPS) is in the field of generating reliable and efficient power to the state, Gujarat since last 30 years. It is located near Wanakbori Dam on the bank of Mahi River in Kheda district. It is a coal based power station. There are seven units of 210 MW each with a total installed capacity of 1470 MW. Unit no 8 of 800 MW is proposed and its installation work is under progress. All the above units are of BHEL make. Various units installed at Wanakbori TPS and its commissioning details are mentioned in table 1.1[4]

Table 1.1: WTPS Units Commissioning Details[4]

STAGE	UNIT NO.	CAPACITY (MW)	YEAR OF COMMISSIONING
STAGE I	1	210	23/03/1982
	2	210	15/01/1983
	3	210	15/03/1984
STAGE II	4	210	09/03/1986
	5	210	23/09/1986
	6	210	18/11/1987
STAGE III	7	210	31/12/1998
	8	800	To be commissioned

1.3 Motivation

The Perform Achieve Trade (PAT) Scheme originated in 2001 under Energy Conservation Act, which empowers Indian Government to identify energy incentives industries as Designated Consumers (DCs) and set mandatory energy conservation standards for them. Out of 478 facilities covering 8 sectors identified by the Ministry of Power's Bureau of Energy Efficiency

Table 1.3: Average GHR and Boiler Efficiencies of 210 MW units[5]

Parameters	Average Design (kcal/kWh)	Average Operating (kcal/kWh)	Average deviation (%)	Range of operating (kcal/kWh)
Gross Heat Rate	2408.3	2765.8	14.8	2384 - 3064
Boiler Efficiency	85.8	81.7	4.8	71.0 - 86.0

(BEE), 144 thermal power plants are given target to reduce energy consumption under PAT scheme. Wanakbori thermal power plant is among one of them and its present heat rate & targeted heat rate are mentioned in table 1.2[5]

Table 1.2: Heat Rate Targets[4]

BASELINE HR	TARGET NET HR	REDUCTION
2887 kcal/kWh	2820 kcal/kWh	67 kcal/kWh

To achieve mentioned heat rate reduction, energy audit of various power plant equipments is necessary. Achievement of the PAT Target is essential to avoid buying of ESCerts for compliance to the Act in case of non-achievement. However, all TPS should try to achieve net heat rate beyond the targets so that the ESCerts can be earned out of the energy efficiency savings.The ESCerts will be issued by Bureau of Energy Efficiency (BEE), Ministry of Power, Govt. of India to eligible Designated Consumers (DCs).[5]

In the year 2006, under the Indo-German Energy Programme (IGEN) power plant component is being implemented by Central Electricity Authority (CEA) in association with Bureau of Energy Efficiency (BEE) for performance optimization and efficiency improvements of thermal power plants. The identified mapping studies of 49 units of 210 MW were completed during the period 2007-2009 in 14 Indian states. The findings of the studies on 210 MW plant is as shown in table:

Many major reasons for the high operating gross heat rates are found. Few of them are listed as below:

1. Low combustion efficiency lead to high carbon loss.
2. Poor sealing and heat transfer in air preheaters
3. High air ingress in the boiler and high heat loss due to poor insulation
4. High auxiliary power consumption due to high heat rate and outages and many more.

1.4 Objective of Present Work

Following work needs to be carried out in order to increase overall plant efficiency by reducing gross heat rate.

- Improve thermal performance of airpreheaters by minimizing internal leakages through sealings as well as carry over leakages.
- To achieve higher boiler efficiency by reducing air ingress into boiler and ducting system.
- To reduce loading on induced draft, primary and secondary air fans.
- To minimize number of force outages as well as auxiliary power consumption which lowers the generation capacity of unit.

- To study recent advancements in APH materials and technology which could be implemented.
- To carry out exergy analysis of APH and boiler.

In order to achieve above mentioned performance improvements below mentioned tests are required to be performed:

1. O_2 mapping test of boiler and APH
2. Clean Air Test
3. Chalk Test

Chapter 2

Literature Review

This chapter includes detailed review of literature on pulverised coal fired boiler, Ljungstrom rotary APH, air ingress and exergy analysis. Different experimental, semi-analytical and exergy analysis related work on thermal power plants carried out by various researchers are also discussed.

2.1 Pulverised Coal Fired Boiler

A pulverised coal-fired boiler is an industrial boiler used in most of the thermal power plants in operation nowadays. It generates thermal energy by burning pulverised coal that is blown into the firebox. The basic idea of firing system using pulverised fuel is to use the whole volume of the furnace for the combustion of solid fuels. Coal is ground to the size of a fine grain, mixed with air and burned in the flue gas flow. Coal contains mineral matter which is converted to ash during combustion. The ash is removed as bottom ash and fly ash. The bottom ash is removed at the furnace bottom.

This type of boiler dominates the electric power industry, providing steam to drive large turbines. Pulverized coal provides the thermal energy which produces about 50% of the world's electric supply. Technical specification of pulverised coal fired boiler and its accessories are given in table 2.1.

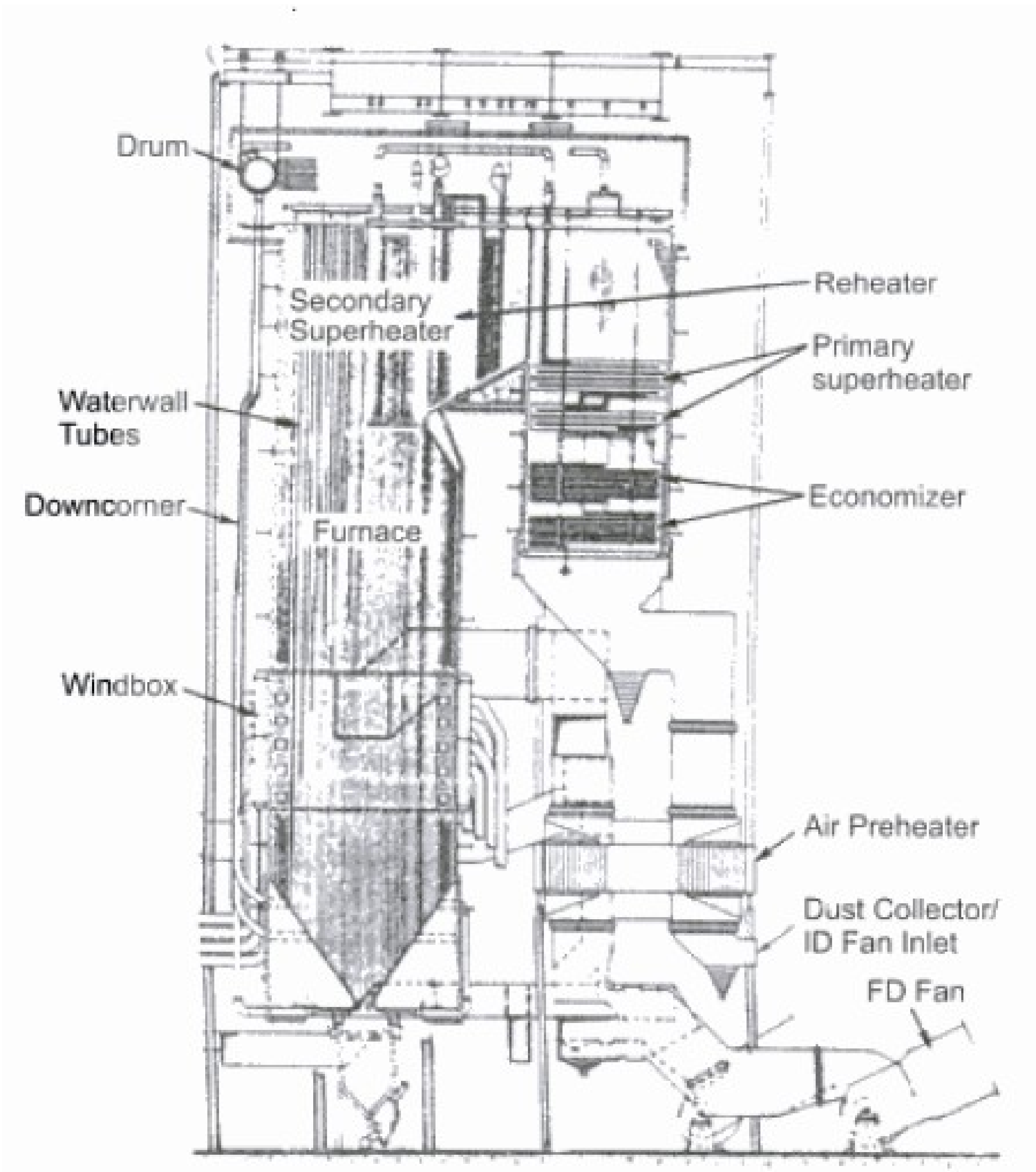


Figure 2.1: General arrangement of pulverised coal fired boiler[6]

Table 2.1: Technical Specifications of Boiler[4, 7]

Boiler Type	Two Pass Natural Circulation Radiant Reheat, Dry Bottom with Direct Tangentially Coal Fired, Pulverised Coal, Balance Draft
Fuel	Indian Bituminous Coal
Size	Upto 210 MW
Furnace Type	Fusion Welded Panels
Boiler dimensions Length of flow passage in the boiler Width Depth	 63.224 m 10.592 m 13.868 m
Low Temperature Superheater	425 ºC
Platen Superheater	515 ºC
Final Superheater	540 ºC
Reheater	300 to 540 ºC
Economizer	245 to 280 ºC
Air Preheater	300 to 336 ºC

The concept of burning coal that has been pulverized into a fine powder from the belief that if the coal is made fine enough, it will burn almost as easily and efficiently as a gas. The feeding rate of coal according to the boiler demand and the amount of air available for drying and transporting the pulverized coal fuel is controlled by computers. Pieces of coal are crushed between balls or cylindrical rollers that move between two tracks or "races." The raw coal is then fed into the pulveriser along with air heated to about 330 °C from the boiler. As the coal gets crushed by the rolling action, the hot air dries it and blows the usable fine coal powder out to be used as fuel. The powdered coal from the pulveriser is directly blown to a burner in the boiler. The burner mixes the powdered coal in the air suspension with additional pre-heated combustion air and forces it out of a nozzle similar in action to fuel being atomized by a fuel injector in modern cars. Under operating conditions, there is enough heat in the combustion zone to ignite all the incoming fuel.

2.1.1 Draught System

The draught system is one of the most essential systems of thermal power plant. The purpose of draught is to supply required quantity of air for combustion and remove the burnt products from the system. To move the air through the fuel bed and to produce a flow of hot gases through the boiler, economizer, preheater and chimney require a difference of pressure equal to that necessary to accelerate the burnt gases to their final velocity and to overcome the pressure losses equivalent to pressure head. This difference of pressure required to maintain

Table 2.2: Basic parameters of 210 MW coal fired thermal power unit[7]

No.	**Rated variable/parameters**	**Unit**	**Value**
1.	Power output (electrical)	MW	210
2.	Boiler efficiency	%	86.0
3.	Power input in coal (energy in coal)	MW	636.6
4.	Power input in steam (energy in steam)	MW	547.5
5.	Coal flow rate	kg/s	41.667
	Total combustion air flow rate	kg/s	269.700
6.	Higher heating value of coal	MJ/kg	15.257
7.	Design coal (permissible variation: ±10%		
	Fixed carbon	%	30.0
	Volatile matter	%	21.0
	Ash	%	41.0
	Moisture	%	08.0
8.	Stoichiometric air to fuel ratio	kg/kg	4.480
9.	Flame temperature	°C	1552
10.	Specific volume of flue gas	m^3/kg	7.980
11.	Feed water flow through water walls	kg/s	204.167
12.	Main steam flow to turbine	kg/s	194.444
13.	Flow through re-heater	kg/s	166.667
14.	O_2in combustion zone	%	3.50
15.	O_2in APH entrance	%	3.53
16.	O_2in APH exit	%	4.20
17.	Leakage rate in boiler up to APH	%	0.17
18.	Leakage rate in APH	%	4.17
19.	Exit flue gas temperature	°C	135
20.	Temperature of air at APH outlet	°C	336
21.	Main steam and re-heat steam temperature	°C	540
22.	Surface areas of heaters		
	Water walls	m^2	2400
	Super heaters	m^2	8200
	Re-heaters	m^2	2600
	Economizer	m^2	5200
	Air preheater	m^2	19,000

the constant flow of air to discharge the gases through the chimney to atmosphere is known as draught.

Draught can be obtained by use of chimney, fan, steam or air jet or combination of these. When the draught is produced with the help of chimney only, it is known as natural draught and when the draught is produced by any other meanss except chimney it is known as artificial draught.

2.1.1.1 Losses in the air-gas loop system

The total draught required to produce the current of air and to discharge the hot gases to the atmosphere is the arithmetic sum of all draught losses in the series circuit.

The total draught losses are given by

$$h_t = h_v + h_b + h_e + h_d$$

where

h_t = Total draught loss in mm of water head

h_v = Velocity head in mm of water head

h_b = Fuel bed resistance equivalent to mm of water head

h_e = Head loss in the equipments

h_d = Head loss in ducts and chimney

The details of loss are given below:

1. Fuel Bed Resistance (h_b) : The fuel bed resistance depends on fuel size, bed thickness and combustion rate.

2. Head Loss in Equipments (h_e) : The manufacturers generally supply data for equipment resistance like air heater, economizer, boiler passes, super heaters, etc.

3. Velocity Head Loss (h_v) : The velocity head loss is always equal to $V^2/2g$ where V is the velocity at exit of the chimney. The draught system is designed to give minimum $V^2/2g$ loss but it must be sufficient to diffuse and mix with the surrounding atmospheric air.

4. Head Loss in Ducts and Chimney (h_d) : The draught loss due to friction in air and gas ducts and chimney is given by Fanning equation as

$$h_d = f.(\frac{L}{4R_h})(\frac{V^2}{2g)}) \text{ in metres of fluid flowing}$$

where $R_h$is hydraulic radius (cross-sectional area/wetted perimeter) and $'f$' is the friction factor of the duct through which air or gas flows. It depends on the smoothness of the duct and Reynold number of the fluid flowing. The value of $'f'$may be taken as 0.005 for steel.

2.1.1.2 Balanced Draught

It is always preferable to use a combination of forced draught and induced draught instead of forced or induced draught alone. The balanced draught is a combination of forced and induced draught. The forced draught overcomes the resistance of the fuel bed therefore sufficient air is supplied to the fuel bed for proper and complete combustion. The induced draught fan removes the gases from the furnace maintaining the pressure in the furnace just below the atmosphere. This helps to prevent the blow-off of flames when the doors are opened as the leakage of air is inwards.[7]

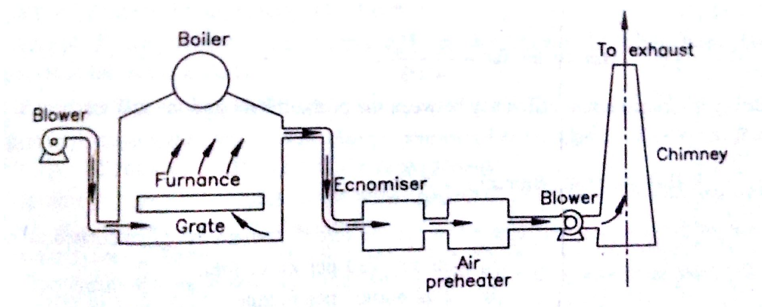


Figure 2.2: Balanced draught[7]

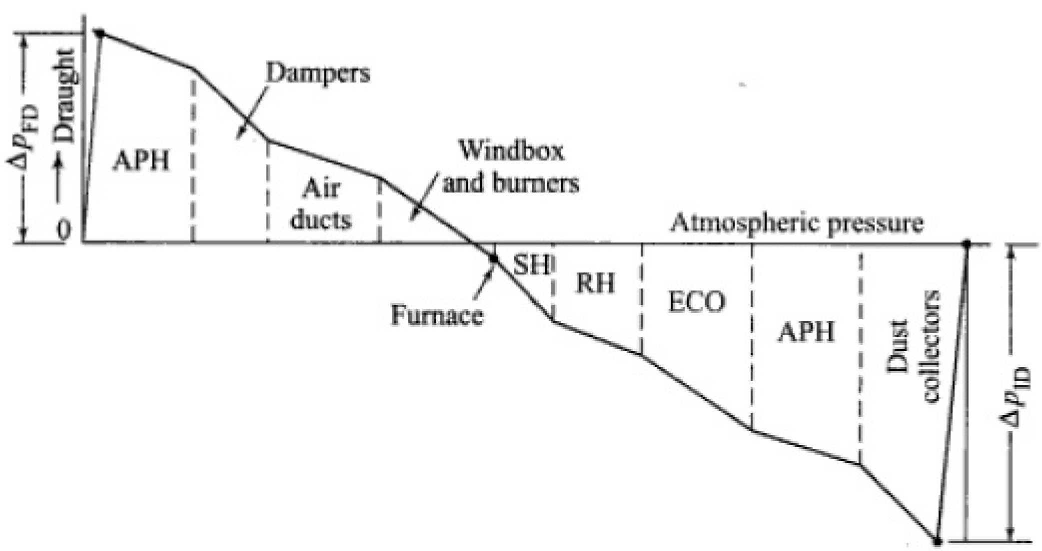


Figure 2.3: Pressure distribution through the system when the balanced draught is used[6]

Advantages of balanced draught system over natural draught are listed below :

1. The rate of combustion is high as the available draught is more. The better distribution and mixing of air and fuel is possible therefore the quantity of air required per kg of

fuel is less. This further reduces the mass of the flue gases formed and heat carried by exhaust gases.

2. The air flow can be regulated according to the requirement by changing the draught pressure.

3. Low grade fuel can be used in combustion chamber as the intensity of artificial draught is high.

4. The efficiency of the artificial draught is nearly 7 % whereas the efficiency of the chimney draught is hardly 1%.

5. The fuel consumption per kW due to artificial draught is 15% less than the natural draught.

6. It prevents formation of smoke as complete combustion is possible even with less excess air.

The major disadvantage of the artificial draught is the high capital cost required and high running and maintenance costs of the fans used.

2.2 Ljungstrom Air Preheater

Throughout the history of boilers there have been many advancements in order to obtain a better performance and lower fuel consumption. However, few inventions have been as successful in saving fuel as the Ljungstrom Air Preheater invented by Fredrik Ljungstrom, then Technical Director at Aktiebolaget Ljungstrom Angturbin. The first installation in a commercial boiler saved as much as 25% of the fuel consumption. In a modern utility boiler the Ljungstrom Air Preheater provides up to 20% of the total heat transfer in the boiler process, but the Ljungstrom Air Preheater only represents 2% of the investment.[3]

The use of a Ljungstrom Air Preheater in a modern boiler plant saves a considerable quantity of fuel - so much that the cost of the preheater is generally recovered after only a few months. It has been estimated that the total world-wide fuel savings resulting from all Ljungstrom Air Preheaters which have been in service is equivalent to 4,500,000,000 tonnes (4,960,000,000 tons) of oil. An estimate shows that the Ljungstrom Air Preheaters in operation annually saves about $30 Billion US. The distribution of thermal power capacity in which Ljungstrom Air Preheaters are installed over the world is shown in the table below. The estimated distribution of annual saving in $US can also be found in the table 2.3 : [3]

Table 2.3: An Estimation showing Ljungstrom APH installed across the world[3]

Continent	Installed Capacity in GW	Percent of global installed capacity	Saving in Billion US$/year
Africa	65	5	1.6
North America	471	30	6.6
South America	35	2	0.7
Asia	370	24	7.7
CIS	216	14	3.8
Europe	365	23	7.6
Oceania	38	2	0.5
TOTAL	**1560**	**100**	**28.5**

2.2.1 Special Features and Characteristics

The Ljungstrom Air Preheater is a regenerative heat exchanger, and comprises a slowly rotating rotor filled with heat transfer plates. The hot and cold gas ducts are arranged so that half of the rotor is in the flue gas duct and the other half is in the primary air duct which supplies combustion air to the furnace. The hot flue gases heat the part of the rotor in their path, and as the rotor rotates, the hot section moves into the path of the combustion air and preheats it. The rotor is divided into a number of sections which pass through seals in order to prevent the flue gases and the combustion air from mixing.

Ljungstrom Air Preheaters are used primarily in boiler plants for preheating the combustion air. In recent years, their use has expanded to include energy recovery in combination with removal of oxides of sulphur and nitrogen from flue gases sometimes using catalyst coated heating element plates. An advantage of the Ljungstrom method over other methods is that the temperature of the flue gases can be reduced below the sulphuric acid dew point without causing problems. In a recuperative heat exchanger, in which the heat flows through a separating wall, the efficiency falls off rapidly if the temperature is reduced below the dew point, as deposits form on the heat transfer surfaces and act as a thermal barrier. On the other hand, deposits on the surfaces of a Ljungstrom Air Preheater have no measurable effect on heat transfer performance. A Ljungstrom Air Preheater is also relatively insensitive to corrosion. Any parts that do corrode away can be easily replaced at a modest cost, and leakage of air to the flue gas side is not affected by corrosion of the heat transfer plates.[3]

2.2.2 Construction Details

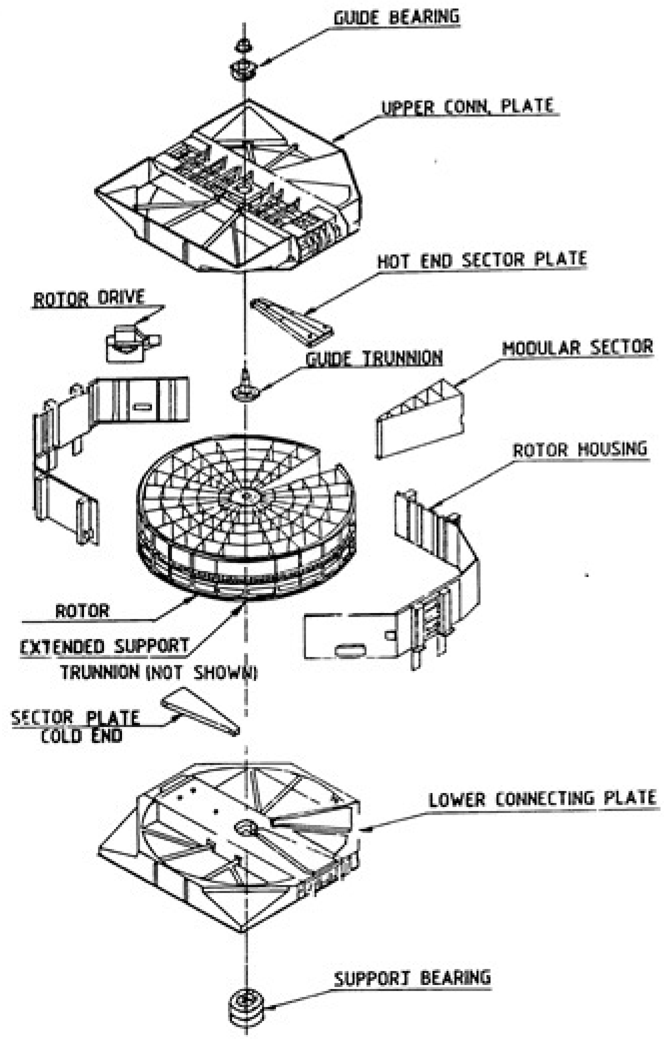


Figure 2.4: Main APH components[1]

An exploded view of Air Preheater is shown in figure 2.4 which shows main APH components such as rotor assembly and housing, hot and cold end connecting plate assembly, sealing system i.e., radial, axial and bypass seals, guide and support bearing assembly, lubrication oil circulation system, main drive assembly and air line components, etc.

2.2.3 Working Principle

As the name implies the tri-sector preheater design has three sections. Use for flue gas. One for primary air (used for drying and transport of coal through mill to the burner) another for secondary air (additional air for combustion around the burners). These helps in avoiding wastage of heat pick up by air due to primary air flow and also help in selecting different temperatures for primary air and secondary air. Whatever is not utilized in primary air can be picked up by secondary air stream. Thousands of these high efficiency elements are spaced compactly arranged with in 12 sectors shaped compartments for heater size from 24.2 to 27 inches, and 24 sector shaped compartments for heater size from 28 to 33 of radially divided cylindrical shell called the rotor. The housing surrounding the rotor is provided with duct connections at both ends and is adequately sealed by radial and axial sealing members forming an air passage through one half of the preheater and a gas passage through the other. As the rotor slowly revolves the mass of the elements alternatively through the air and gas passage, heat is absorbed by the element surfaces passing through the hot gas stream. These are the same surfaces are carried through the air stream. They released the stored up heat thus increasing the temperature of the incoming combustion or process air.

The Ljungstrom air preheater is more widely used than any other type of heat exchanger for comparable service. The reasons for this world wide acceptance are its proven performance and reliability, effective leakage control, and its adaptability to most, any fuel burning process. It is both designed and built to operate over extend periods with durable, uninterrupted service. Simplicity of design also makes it easy and economical to maintain while in operation and at scheduled outages.[1]

2.2.4 Advantages of Air Preheaters

1. Combustion stability is improved by use of hot air.
2. Intensified and improved combustion.
3. Burning poor quality fuel efficiently.
4. High heat transfer rate in the furnace and hence lesser heat transfer area requirement.
5. Less unburnt fuel particle in flue gas thus complete combustion is achieved.
6. Intensified combustion permits faster load variation.
7. In the case of pulverised coal combustion, hot air can be used for drying the coal as well as for transporting the pulverised coal to burners.
8. This being a non-pressure part will not warrant shut-down of units due to corrosion of heat transfer surface which is inherent with lowering of flue gas temperature.[1]

2.3 Air Ingress

In balanced draft furnace, the FD fan / PA fan / SA fan pump the air in to the furnace. The flue gas produced is drawn through the boiler by the ID fan. Hence the furnace and downstream the furnace the boiler is under negative pressure. Thus if some leakage spots are there, the ambient air is drawn through such openings.

By virtue of boiler configuration, openings are to be made in the boiler enclosures / waterwall enclosure. If a seal is improperly designed or improperly erected the seal may fail and develop leakages. It is possible some of the seals are not taking care of thermal expansion or the service conditions and thus leakage may develop. One step further the repairer has not put back the seals as per design.[4]

Various air ingress zones are shown in figure 2.5 and effects are listed below :

1. High unburnt carbon in fly ash

   It is a general practice to trim the air flow based on O_2 indication from flue gas. When the air leakage is present the O_2 indicated by the on-line O_2 meter would mislead the operator. The furnace runs in to substochiometeric condition. This ultimately leads to increased to unburnt.

2. Increased fuel consumption

   The air ingress downstream the flue path leads to increased heat loss in the chimney. To compensate for the heat loss one has to feed more fuel.

3. Overloaded ID fan

   When we experience that the ID fan is falling short of capacity, we tend to invest in new ID fan / we start with fan vendors for increasing the fan capacity. In many cases the second ID fan is opted with higher capacity with the assumption that the existing ID fan is short of capacity. In some cases even the second ID fan may also prove useless as the leakage persists and the furnace still goes with positive pressure. The boiler operating expenses increase due to additional power consumption.

4. Back fire in furnace

   The back fire is continuously experienced. The ID damper is at full open position. The operator has no option except to continue with the problem. The unsafe situation persists. The Insulation of the boiler is spoilt on this account. Soot is seen around the furnace access doors.

5. Secondary combustion in superheater zones

   The furnace begins to starve when the air ingress is more from roof seal box / Con-

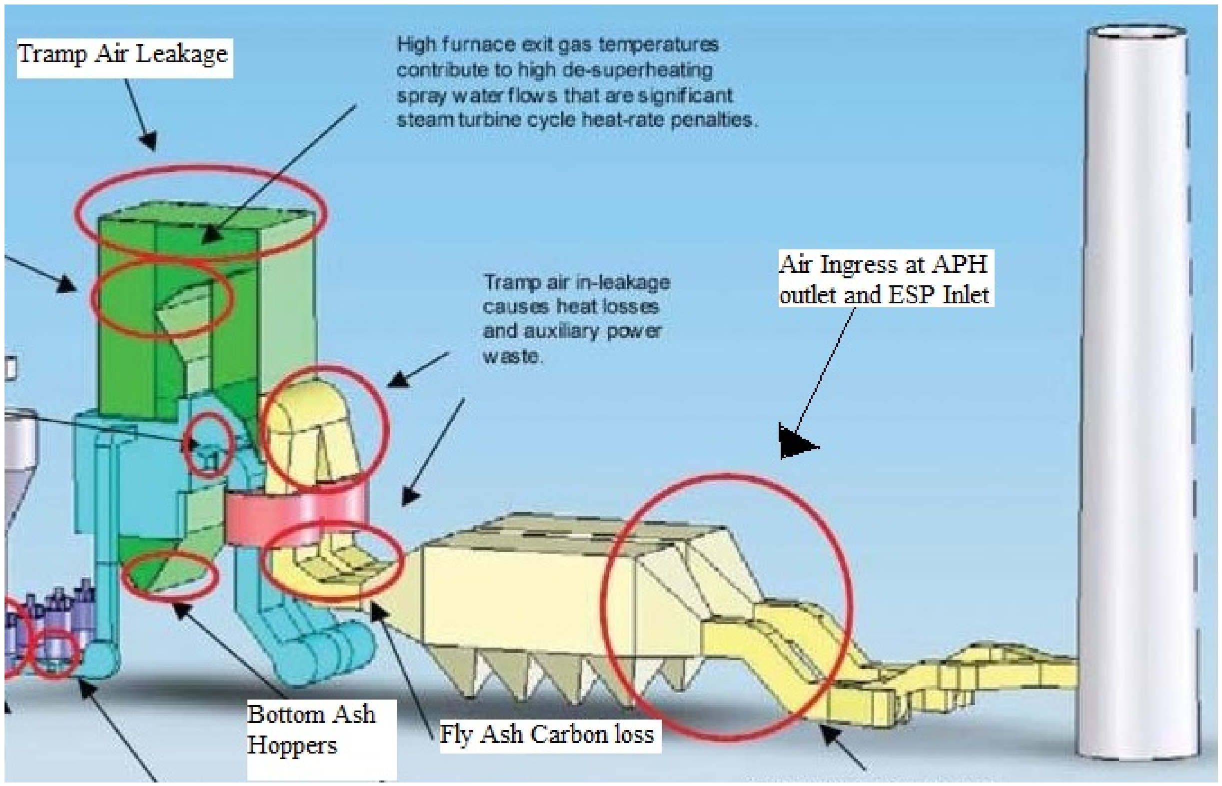


Figure 2.5: Air Ingress Zones[8]

vection SH seal box. The leakage air allows secondary combustion of volatiles. The SH temperature becomes uncontrollable. Particularly at the time of load variation, the fuel feeder rpm is regulated by the operator and he finds the SH steam temperature rises faster than the pressure.

6. Secondary combustion in Boiler bank hoppers

   The unburnt fuels burn at the boiler bank ash hoppers, since the furnace is at substochiometeric conditions when there is air ingress is present downstream. The unburnt fuel travels downstream instead of completely burning in the furnace. The ash hoppers get distorted due to secondary combustion.

7. Ash blockages in boiler bank Baffles

   When the ash is not fully burnt in the furnace, the flow ability of ash comes down. The ash particles now contain fuel particles which may be fibrous / irregular in nature. Thus the fuel and ash settle at every possible location, where the surfaces are less inclined or flat. The ash does not flow freely and thus ash accumulates at baffles. Whatever the draft is set the ash does not flow due to nature of the accumulations.

8. Ash blockages in Hoppers

   Carryover of fuel particles to ash hoppers would lead to combustion in ash hoppers. Lumps form due to static combustion. We try to poke the ash drain pipes but situations repeat often. The combustion is not complete at the furnace and hence the troubles.

9. Excess Desuperheater spray

   In most of the designs the furnace is designed to be hotter as the combustion is to take place here. The furnace dimensions are so chosen, to achieve the necessary residence time for the fuel particle to burn fully. Starvation occurs when the air can bypass the furnace and enter the flue gas downstream. No one can assess the amount of air ingress at leaky zones. Under such conditions the combustion zone shifts to SH section. Simultaneous combustion and heat transfer at SH section leads to excess Steam temperature. To our luck if excess capacity is available in the spray control valve, we tend to spray more. More spray will lead to solids added to SH section. The solids leave behind in SH leading to deposit related failures. More the spray the turbine blade deposition is experienced.

10. Clinkers formation in furnace

    The furnace temperatures are controlled by incorporating necessary heat transfer surface and by admitting required excess air to cool down the gas below the ash melting temperatures. The excess air can not be given in the furnace when the ID draws the leakage air downstream. Refractory furnaces get coated ash deposits. Honey combing of ash accumulations is seen in some agro waste fired boilers. Refractory roof tops eventually collapse due to increased weight.

11. High furnace temperatures and refractory walls cave in
    The excess air when not given in the furnace, the furnace temperature exceeds the design gas temperatures. When the fuel does not have much of ash, the furnace temperatures go up. The refractory walls expand unusually leading to furnace walls caving in.

12. Furnace doors failures
    Furnace doors in balanced draft furnaces are refractory lined to thickness of not more than 250 mm. The doors get cooled by the outside ambient air present around the door. When the furnace is under positive pressure the ambient air is not present near the vicinity of the doors. Then the doors bulge. The manhole frame and manhole distort due to heat. We think the doors fail due to material defect.

13. Fly ash nuisance around the boiler
    The fly ash poses a great nuisance not only harming the eyes but also lungs. The boiler house becomes shabby. The industrial standards go down in front of your customers. The costs for cleaning the boilers go up. During maintenance, hours are to be allocated only for cleaning. The boiler downtime increases due to this.

14. Injuries to personnel
    Some of the boilers are provided with penthouses where, the ash accumulates to high level. It poses safety hazard for any one who enters the penthouse. The hot ash lying underneath could cause hot burns as well.[4]

2.4 Exergy

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

The main objective of the implementation of an exergy-based approach is to find appropriate trade-offs between fuel use and investment cost or environmental impact, in order

Table 2.4: Comparison between Energy and Exergy[9]

SR No.	**ENERGY**	**EXERGY**
1	It is subjected to the law of conservartion	It may be looked upon as law of degradation of energy
2	It is function of the states of the systems under consideration	It is the function of state of the system and of surroundings
3	It may be calculated on the basis of any assumed reference state	In this case reference state is imposed by the environment
4	Energy in - Energy out = 0	Exergy in - Exergy out = Exergy destroyed
5	It does not depends on pressure for ideal gases	It depends on pressure

to improve a process. Exergy analysis of the systems, which analyses the processes and functioning of systems, is based on the second law of thermodynamics.

In this analysis, the effectiveness of the second law which states the exact functionality of a system and depicts the irreversible factors which result in exergy loss and efficiency decrease is mentioned. Therefore, solutions to reduce exergy loss will be identied for optimization of engineering installations. The great advantage of exergy calculations over energy calculations is that exergy calculations pin point exactly where the real losses in processes appear. Furthermore the exergy content stream is a real evaluation of energy it indicates the fraction of energy that really can be used.[9]

- Comparison of Energy and Exergy

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 are required 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.

In this work, exergy analysis of boiler and APH can be carried out by measuring:

1. Exergy of fuel supplied

2. Exergy of water supplied

Table 2.5: Comparing Experimental Results with Theoretical Results[10]

Test no.	A	B	C	$Ntu_{,0}$	C^*	Cr^*	Experimental efficiency	Analytical efficiency	Numerical efficiency
1	9.5	1.3	1.3	13.3	0.99	349.01	0.635	0.96	0.95
2	14.5	1.3	1.3	13.3	0.99	399.05	0.625	0.96	0.95
3	9.5	4.5	1.3	16.7	0.29	259.77	0.975	0.999	0.999
4	14.5	4.5	1.3	16.	0.29	396.5	0.965	0.999	0.999
5	9.5	1.3	4.5	4.8	0.28	261.75	0.835	0.99	0.99
6	14.5	1.3	4.5	4.8	0.28	399.5	0.84	0.99	0.99
7	9.5	4.5	4.5	6.4	0.99	75.6	0.58	0.88	0.87
8	14.5	4.5	4.5	6.4	0.99	115.41	0.597	0.88	0.87

3. Exergy of air supplied

4. Exergy available with steam

5. Exergy available with flue gas

2.5 Recent Work on Boiler and APH

2.5.1 Experimental Work

In the study on Experimental and Sensitivity Analysis of RAPH by N. Ghodsipour and M. Sadrameli[10], a rotary regenerator was simulated by solving a developed mathematical model and optimized with the experimental design method. In this way, the effect of dimensionless parameters on the effectiveness of rotary heat exchangers was investigated. There are only three main parameters affecting the regenerator efficiency. These are A = rotational speed, B = hot air velocity and C = cold air velocity. Statistical calculations were performed and ANOVA table for each of the parameters and their interactions were prepared and F values were compared. By comparing F values, it is cleared that B and C parameters and their interactions have significant effects on the efficiency of the regenerator. Finally the mathematical model is used to obtain the optimized parameters for the regenerator. Comaprison of reults shown in table 2.5

Study on the direct leakage of RAPH with multiple seals was studied both experimentally and numerically by Mingkun Cai et al.[11], in which they have tested different seal structures from single to triple seals cases, which are used to control the direct leakage. To describe the effect of the gap width and inlet pressure on the orifice coefficient and in terms of the air leakage engineering modifying factor, some improved relationships were obtained, which are superior over the traditional ones because of their better accuracy. The main factors that affect the air leakage are the air flow expansion, inlet velocity at the seal gap entrance, and the flow boundary conditions on the seal plate surface. The first upstream seal accounts for most of the total pressure drop on the entire multiple-seal system. The simulation results on the multiple-seal system show that the system of triple seals is better in controlling the leakage than that of the double seals when the seal gap is small; however, when the seal gap increases up to a critical value, the difference between the double- and triple-seal becomes negligible.

Experimental work performed by M.S. Bhatt[7] to quantify effect of air ingress on the energy performance of coal fired thermal power plants. Ingress of air in boilers leads to drops in energy efficiency. This paper presents the effects of air ingress in the combustion zone, post combustion zone and air preheater on the energy efficiency and loading capacity of a coal fired thermal power plant operating on fuel with high ash (35–45%). The optimal O_2 in the flue gas for a pulverized coal fired system is 3.5% (corresponding to 20% excess air). The operating values are in the range of 4.2–6.0% in membrane type boilers and up to 10% in refractory type boilers (after sustained periods of operation). The leakage rate of boilers (up to the entrance of the APH) is designed at 0.2% while the average operating values are 7.25% for membrane type enclosures and 33.61% for refractory enclosures. The leakage rate of the APH is designed at 5.0% while the operating values range from 13.66% to 20.13% for rotary and tubular APHs. When the O_2 in the combustion zone varies from 3.5% to 8.0%, efficiency drops of 2.0% points are experienced in the boiler and turbine separately, and the gross overall efficiency drop is ˜3.0% points. The units do not experience any capacity drop up to an O_2 in the flue gas of 6.0% before the APH. At an O_2 in the flue gas (before APH) of 7.2%, a mild limitation on the unit capacity of around 2–3% is experienced. When O_2 in the flue gas (before APH) reaches a level of 9.0%, 20% capacity drop of the unit is experienced due to which the plant load cannot be raised higher than 80%. Beyond the level of 9.0% (rare occurrence), the unit is quite difficult to operate and has to be taken off for overhaul.

Analysis of performance of Ljungstrom Air preheater elements carried out by Vullloju et al.,[12] the performance of Ljungstrom Air Preheater is depended on the heat transfer element profiles. New profiles are being designed in such a way that these new profile elements must be improving the efficiency of Air Preheater. In this context, it is felt necessary to develop new element profiles with lesser pressure drops for efficient heat transfer with less power consumption to improve overall efficiency of thermal power plants. In this research paper, two types of element profile (Flat Notched Crossed & Double Undulated elements) are tested using cold flow studies with the help of wind tunnel and compared their performance at different Reynolds numbers. Figure 2.6 shows pressure drop increases with increase in velocity. From

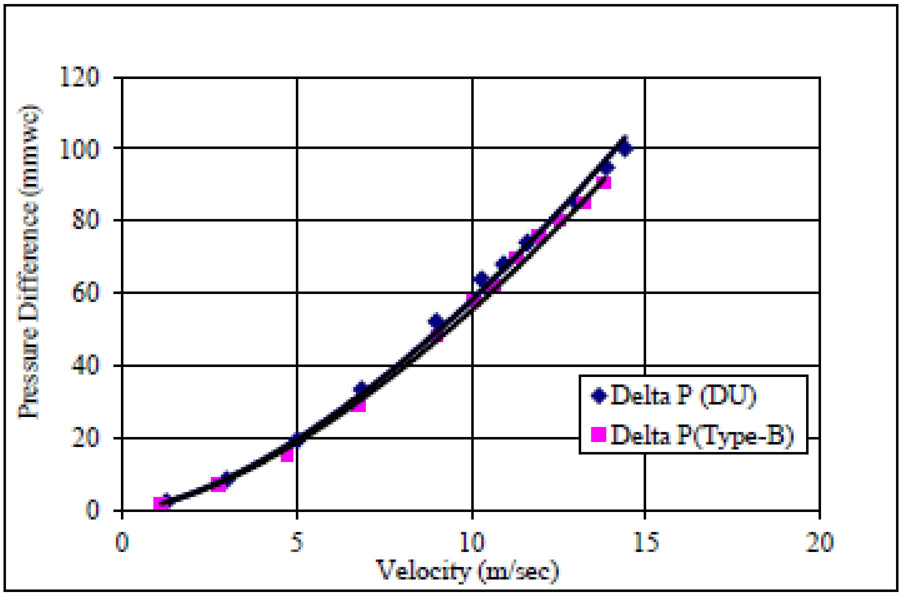


Figure 2.6: Comparison of DU and FNC elements[12]

the tests carried out in the Wind Tunnel test facility to find out the performance of both DU and FNC elements, the test result reveals that;

1) It has been concluded that hydraulic diameter does not effect on the performance of DU and FNC elements as the difference between hydraulic diameter of DU and FNC is less.

2) The pair height for FNC elements is less than the pair height for DU elements. So, Flat Notched Crossed elements are occupied less volume than Double Undulated elements in a given number of pairs of elements. Thus, the size and weight of Air Preheater can be decreased using FNC elements instead of DU elements.

3) It is concluded that heat transfer co-efficient FNC elements is higher than DU element as The residual time for FNC elements is more than the residual time for DU elements .

4) The fluid pumping power is directly proportional to the pressure drop in the fluid across element. The fluid pumping power is less for FNC elements compared to DU elements.

From the above points, it has been concluded that the performance of FNC elements is more than DU elements.

The work on performance evaluation and optimization of air preheater in thermal power plant completed by G. Shruti et al.,[13] focuses on the performance of Regenerative type air pre heater. The performances were evaluated before and after radial sector plate clearance adjustments with air preheater tests, and improvement is seen along with air preheater optimization. By reducing the area available for leakage from the air to the gas side between the rotor and the air preheater housing by adjusting the radial sector plate reduces the air leakage and max efficiency can be obtained. And increase in x ratio indicates maximum heat recovery in the Air preheater.

2.5.2 Analytical Method

Hong Yue Wang et al.,[14] performed the study on heat transfer model of tri-sectional RAPH based on the semi-analytical method. Even though tri-sectional thermal regenerators are applied in power plant all over the world due to its compactness and better performance, low temperature erosion is still one of the biggest hurdles to be overcome. To prevent the erosion problem at the cold end, the temperature distribution in the regenerator must be determined. In the present investigation, a semi-analytical method is employed to investigate the three-dimensional heat transfer of a tri-sectional rotary air preheater. Special attention was placed on the temperature change of fluids and temperature distribution in the matrix of the preheater. The simulation results indicate that the semi-analytical approach is more precise and has better convergence than that of the standard numerical iterative method. The numerical analysis and experimental results show a significant dependence on the reduced length and reduced period of the rotary air preheater. The effects of dimensionless parameters on the temperature distribution of tri-sectional air preheater were investigated.

1. The semi-analytical method yields a stable, rigorous solution to the heat transfer model of tri-sectional air preheater. It can be used to simulate the cases which were previously beyond computation.

2. The semi-analytical method allows a considerable reduction in the amount of computational effort required to calculate accurate solution to the model. The characteristics of precision and convergence are better than the standard numerical method and correlates well with limited experimental data.

3. The dimensionless parameters of reduced length Λ and reduced period Π were employed to show the temperature distribution of tri-sectional air preheater in three-dimensions. Furthermore, theoretical simulation results can be used to guarantee that the tri-sectional air preheater operates safely and economically.

2.5.3 Exergy Analysis

Hayato Hagi et al.[15], carried out exergy analysis on oxy-pulverized coal power plant by means of the mapping exergy destruction sources. Operating conditions and current state-of-the-art design are considered in order to set a base-case for the identification of the exergy destruction. The study reveals the location and the magnitude of the losses. Main losses occur in the boiler, steam generation, turbines, distillation unit, compression steps of the Air separation unit (ASU) and Compression and purification unit (CPU), gas quality control system, and in the regenerative heater. Compression heat integration, bypass regenerative heater with improved heat exchange, preheating of the oxygen flow with the bypass flow surplus heat and reheat of the cold depolluted flue gas in a regenerative heater are implemented. Important reduction in exergy destruction is reported and the exergy efficiency of the integrated power plant increases from 36.4% to 39.6%, which corresponds to an overall exergy destruction diminution of 16%.

Exergy analysis on irreversibility of RAPH carried out by Hong Yue Wang et al.[16], were employed to measure the effect of irreversibility in RAPH on the efficiency of thermal power plant. A major disadvantage of the RAPH is that there is an unavoidable leakage due to carry over and pressure difference. There are gas streams involved in the heat transfer and mixing processes. There are also irreversibilities, or exergy destruction, due to mixing, pressure losses and temperature gradients. Therefore, the purpose of this study is based from the second law of thermodynamics, which is to build up the relationship between the efficiency, and the efficiency of the thermal power plant are examined by changing a number of parameters of RAPH. Furthermore, some conclusions are reached and recommendations are made so as to give insight on designing some optimal parameters.

Pradeepsingh Hada and Ibrahim Hussain Shah [17], 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 for boiler was calculated and found as 93 MW.

Mali Sanjay D and Dr. Mehta N S [18], presented easy method of exergy analysis for thermal power plant. The results indicate that at each point of power plant exergy destruction is much more than energy loss at that points. It has been observed that 47.43% exergy loss occur in the combustor which shows combustor is not fully adiabatic and combustion may not be complete. The major exergy destruction occurs in APH which needs to be carefully inspected. HP and IP turbine have best performance and LP turbine have poor performance.

R Mahamud et al. [19], through exergy analysis and efficiency improvement techniques identifies areas where most of the exergy is lost and discuss potential of the lost energy for improvement of the plant energy efficiency. It shows that the boiler of a subcritical power generation plant is the major source of exergy loss. Only negligible amounts of useful waste energy can be recovered through implementing some heat recovery system. In order to achieve significant improvement of energy efficiency the boiler and turbine systems need to be altered, which require further techno-economic study.

2.5.4 Recent Advancements in Air Preheater Technology

2.5.4.1 VN Sealing System

By fitting the advanced Howden VN sealing system, the significant increase in lekage over a period of time can be reduced or eliminated. Modern air preheaters and Howden retrofitted air preheaters achieve 5 to 8% air leakage with a negligible leakage rise following 3 to 5 years operation. Mant heaters currently have little provision for on-load adjustment of seals. On the latest designs of heaters, sector sealing plates and axial seals are provided with automatic adjustment, via gap sensors linked to seal actuators. Leakage rates immediately after commissioning can be low, but high levels of maintenance are required. The unique Howden VN sealing system does not use moving sector plates and associated sensors, actuators and control systems.[20]

Howden experience showed that moving sector plates were a prime cause of the increasing leakage due to; wear on the rotor seals and deterioration of seals between the adjustable sector plates and the casing. The solution was to use fixed sector sealing plates and to compensate for the increased gaps between rotor and sector plate by doubling the number of seals on the rotor. In addition the sealing strips on the rotor were changed to a single leaf design that had been proven to be an improvement over the previous channel design. A further benefit of VN sealing is that annual maintenance hours are reduced by more than 50% over the more traditional designs.

Benefits of upgrading air preheaters to latest Howdem technology include:

- Reduced draught fan power and consequent increase in saleable power output.
- Lower gas velocities permitting increased precipitator efficiency and reduced dust emission from the stack.
- Reduction of the temperature dilution effect of air leakage, reducing corrosion attack downstream of the air preheater.
- Reduction of leakage, making more air available at the coal mills to ensure that sufficient pulverised fuel can be transported to the burners and thus any shortfall in station MW output is recovered.

2.5.4.2 Enamelled Elements

Vitreous enamel coating produces a combination of both corrosion protection and good cleanability to heat transfer elements and should be specified where high fuel sulphur contents (above 0.5%) and/or high moisture with high dew points. It is a must for most oil fired operations.[20]

Enamel coatings exhibit the following properties:

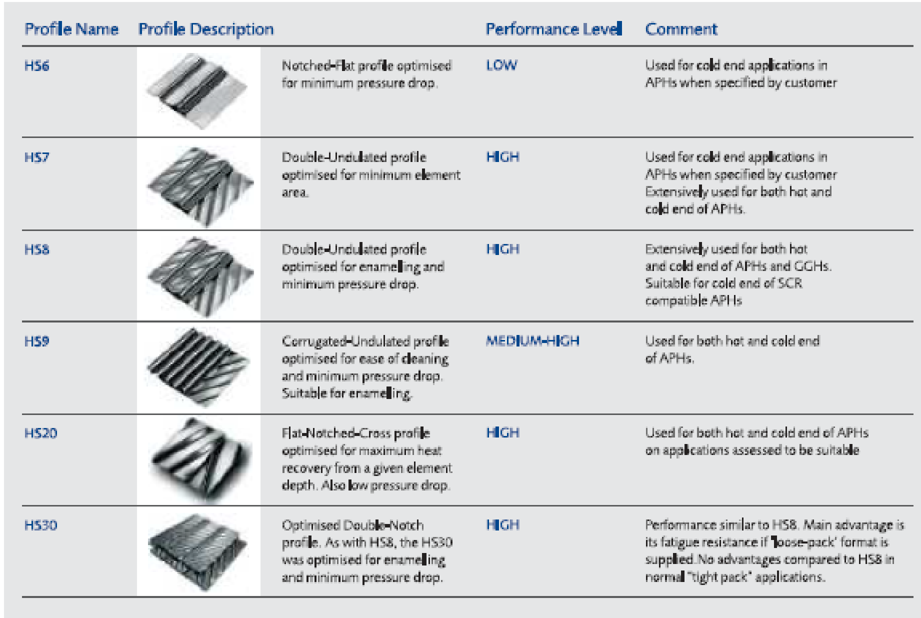
- Extreme corrosion resistance.
- Excellent bonding to steel substrate.
- High mechanical and thermal shock resistance.
- Abrasive and wear resistance.
- Smooth low friction surface.
- Non toxic and environmnetally safe.

The quality and life of the enamel coating is dependent on a number of critical factors, including:

- Base material
- Enamel constituents
- Enamel thickness
- Enamel porosity

Howden elements use specially decarbonised enamelling steels, in combination with enamels with a minimum amount of additives, in order to give maximum adhesion at the steel/enamel boundary and to minimise the production of gas pores during baking.

Because heat transfer elements are regularly subjected to high-energy excitation forces from the sootblowers used to clean the elements, it is essential that these elements are packed sufficiently tightly in their baskets to minimise the element vibration and consequent fatigue damage. When enamelled elements are used, it is equally important that the packs are not over-compressed as this can cause localised damage to the enamel coating that can lead to subsequent loosening of the elements during heater operation.



Profile Name	Profile Description		Performance Level	Comment
HS6		Notched-Flat profile optimised for minimum pressure drop.	LOW	Used for cold end applications in APHs when specified by customer
HS7		Double-Undulated profile optimised for minimum element area.	HIGH	Used for cold end applications in APHs when specified by customer Extensively used for both hot and cold end of APHs.
HS8		Double-Undulated profile optimised for enamelling and minimum pressure drop.	HIGH	Extensively used for both hot and cold end of APHs and GGHs. Suitable for cold end of SCR compatible APHs
HS9		Corrugated-Undulated profile optimised for ease of cleaning and minimum pressure drop. Suitable for enamelling.	MEDIUM-HIGH	Used for both hot and cold end of APHs.
HS20		Flat-Notched-Cross profile optimised for maximum heat recovery from a given element depth. Also low pressure drop.	HIGH	Used for both hot and cold end of APHs on applications assessed to be suitable
HS30		Optimised Double-Notch profile. As with HS8, the HS30 was optimised for enamelling and minimum pressure drop.	HIGH	Performance similar to HS8. Main advantage is its fatigue resistance if 'loose-pack' format is supplied. No advantages compared to HS8 in normal "tight pack" applications.

Figure 2.7: Howden Element Range from Notched-Flat to Double-Notched[20]

2.5.4.3 Leakage Solutions

Full contact seals have a proven track record of reducing air heater leakage by approximately 50% compared with the original equipment type radial seals Figure 2.8 seals commonly found today on most air heaters. An example of a high performance full contact radial seal (DURAMAX™) is shown in Figure 2.9. In comparison with an original design seal, which is really a rigid "proximity" air dam, the full contact seal is constructed with a spring bellows that allows the seal to maintain a continuous, but flexible contact with the sealing plate at all times, effectively eliminating the main path for radial seal leakage. The high performance circumferential seals (DURAFLEX™) shown in Figure 2.10 have an interlocking/self reinforcing structural design which allows the seals to be set in close proximity to the rotor sealing surface without being damaged, thus minimizing the gaps and leakage openings in comparison to original style seals (figure 2.11). [21]The following photographs illustrate the differences in these seals.

2.6 Conclusion : Literature Review

After studying the existing literature on boiler and APHs used in power plants and exergy analysis work carried on different power plant components, the following conclusion can be drawn:

- Experimental, analytical and numerical study made on RAPH by varying its rotational

Figure 2.8: Original Radial Seal[21]

Figure 2.9: Continous Contact DURAMEX Seal[21]

Figure 2.10: DURAFLEX Circumferential Seal[21]

Figure 2.11: Original Circumferential Seal[21]

speed, hot air velocity and cold air velocity shows that maximum efficiency will be obtained between 2-5 rpm.

- Simulation results with multiple seals shows that triple seals are better than double seals when the seal gap is small but when the seal gap increases the difference between them becomes negligible.

- When O_2 in the flue gas before APH reaches up to 6% the unit do not experience capacity drop but if it reaches up to 9% then the unit experiences 20% drop in generating capacity i.e., plant load cannot be raised than 80%.

- Performance of FNC element is better than DU elements based on comparison of size, weight, heat transfer co-efficient and pumping power.

- By adjusting radial sector plates reduces area available for leakage from air side to gas side which minimizes leakage and maximize efficiency.

- A semi-analytical approach employed to solve three dimensional heat transfer problem of RAPH shows that the semi-analytical approach is more precise and has better convergence than that of the standard numerical iterative method.

- Identification of exergy destruction shows that the overall exergy efficiency of integrated power plant can be increased from 36.4% to 39.6%.

- Reduction of the total quantity of air leakage in APH does not guarantee the improvement in the efficiency of the power plant, but the reduction of air leakage at the hot end will directly enhance the efficiency of the thermal power plant while the total air leakage is held constant.

Chapter 3

Performance Evaluation of Air Preheater

This chapter includes operation and maintenance practices of APH. Energy Audit of APH which includes O_2 mapping by analysis of flue gas and air entering and leaving the APH. Procedure, operating conditions and instruments required for the test are discussed in detail in this chapter. Various indices which provides proper idea about the effectiveness of APH performance and conditions of internal leakages in the APH are also including.

3.1 Operation and Maintenance of APH

During operation the degree of fouling is reflected on the pressure drop across the air heater for a particular loading condition. On-load cleaning if provided can be done at suitable intervals to keep the fouling to the admissible level. Otherwise it is essential to plan a shut-down of the unit for off-load cleaning. The cleaning of air heater to keep down the fouling is essential as this is the primary cause for all the air heater problems i.e. plugging corrosion, erosion and fire. Use of low sulphur oil, air by-pass, gas by-pass, steam coil air heaters, hot air recirculation if any provided should be resorted to at low loads and starting to reduce the fouling and corrosion. Low excess air will also reduce the corrosion. During shut-down at regular intervals the air heater has to be inspected for the rate of fouling and corrosion. From the rate of corrosion the length of life may be predicted and replacement can be planned accordingly, on an outage.

During annual overhauls air heaters should be thoroughly cleaned and water-washed. During other shut-down it can be cleaned depending upon the fouling. Where erosion is experienced (mostly) in P.F. fired boilers cleaning of air heater at all possible shut-downs will yield good results. Leakage in air heaters can be checked during operation by analysing flue gas for CO, drop across the air heater. This leakage is through seal and entrainment of air heaters. Seal design and adjustments provided now offer drop of CO, to around 0.5%. This may rise upto 1.0% for 12 months operation. The CO, drop is checked before shutting down for overhaul and during the overhaul the seals are to be adjusted or replaced in the case of regenerative air heater. Since regenerative air heaters are of rotary type, it is necessary to have a preventive maintenance programme, to check the drives, bearings, cleaning devices, oil circulation system etc.[1]

3.2 Energy Audit of APH

3.2.1 O_2 Mapping procedure

There's a lot of stratification in flue gas at air heater outlet ducts and sometimes at air heater inlet ducts due to bends in gas duct & skin air ingress. So a representative value of flue gas composition ($O_2/CO_2/CO$) and temperature is to be obtained by grid sampling of the flue gas at multiple points in a plane perpendicular to the flow at APH inlet and outlet, using a portable flue gas analyzer & digital thermometer. Single point measurements using Orsat provide only relative values and can be misleading. A grid traverse using portable gas analyzers is essential for correct assessment of air heater performance.

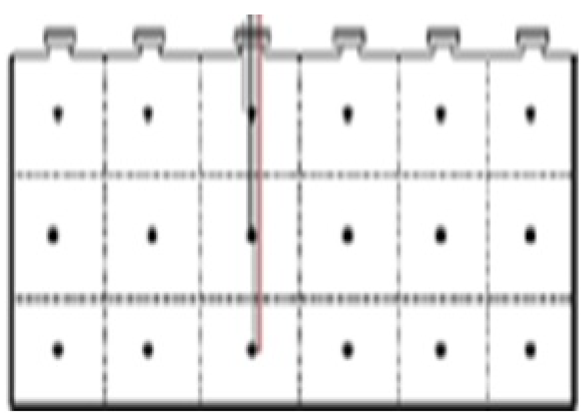


Figure 3.1: Grid Sampling Traverse for correct assessment[22]

A typical cross section of the flue gas duct with an 18-point grid is shown in figure 3.1 . Each dot indicates a sampling point for the measurement of gas composition and temperature. As shown, fixed multiple point sampling probes are installed with thermocouples forming a grid covering the complete cross-section of the duct. Alternatively, single tube probes can be used to perform a traverse across the section of the duct. Gas duct is divided into equal cross-sectional areas and gas samples drawn from center by point by point traverse.

Marking /etching is done on the sampling tubes at D/6, D/2 & 5D/6, if D is the duct depth. The probe is inserted in each port and samples are at different depth as per marking. Temperatures of flue gas are also measured at the same location using a similar tube temperature probe.[22]

3.2.2 Measuring instruments used for O_2 mapping

a) Flue Gas Analyser

Flue Gas Analyzer is a device that monitors the composition of the flue gas of a boiler heating unit to determine if the mixture of air and fuel is at the proper ratio for maximum heat output. Flue gas analyser is a portable instrument meant for measuring oxygen, carbon monoxide and carbon dioxide in the flue. Typical flue gas analyser used for O_2 mapping is shown in figure 3.2. Its resolution is 0.1 and accuracy is 0.2%.

Figure 3.2: Flue Gas Analyser

b) Digital Thermometer

Digital thermometers of range upto 1200 °C are used for measuring the temperature of flue gas leaving the furnace of the boiler. A digitherm is as shown in figure 3.3.

Figure 3.3: Digital Thermometer

c)Digital Manometer

A digital manometer as shown in figure 3.4 is used for measuring the pressure inside various components of boilers such as economizer, APH, ducts,etc. Its range is ±0 to 3500 mmWC. Its accuracy is 0.2% and resolution is 1 mmWC.

3.2.3 Operating conditions of Test runs & duration

Test runs are conducted at an easily repeatable level at defined baseline condition at full load. The operating conditions for each test run are as follows.

- No furnace or air heater soot blowing is done during the test.
- Unit operation is kept steady for at least 60 minutes prior to the test.
- Steam coil air heater (SCAPH) steam supply is kept isolated.
- No mill change over is done during the test.
- All air and side damper position should be checked during the test period.
- The test is abandoned in case of any oil support during the test period.

Figure 3.4: Digital Manometer

- Regenerative heater should be in service with normal drip cascading

The test run duration will be the time required to complete two transverse for temperature and gas analysis. Two separate test crews sample the gas/air inlet and outlet ducts simulation.

3.2.4 Traverse location

1. Gas Side
   The gas inlet traverse plane is located as close as possible to the air heater inlet. This is done to ensure that any air ingress from the intervening duct/expansion joints is not included in air preheater performance assessment. The gas outlet traverse plane is located at a suitable distance downstream the air preheater to allow mixing of the flow to reduce temperature and O_2 stratification.

2. Air Side
   The air inlet traverse plane is located after air heating coils and as close as possible

to the air heater inlet. Since the entering air temperature is uniform, a single probe with 2 temperature measurement points in each duct is used for measurement. The air outlet traverse plane is located at suitable distance downstream the air heater to allow mixing of the flow to reduce the gas stratification. A grid of six measurement points in each duct is used to measure the temperature.

3.3 Performance Indices

Air heater is the last heat recovery equipment that transfers heat from the exiting flue gas to the incoming combustion air. Losses due to deterioration in air heater performance cannot be recovered and result in deterioration of ESP & boiler efficiency. A 22 ºC increase in exit gas temperatures leads to loss of almost 1% in boiler efficiency and 25 kcal/kWh in unit heat rate. For a 210 MW unit, deterioration of 1% boiler efficiency results in an annual loss of Rs. 4 to 5 crores. APH performance deterioration also results in increase in gas flows and temperatures leading to increased loading of ID fans. Many a times, this becomes a limiting factor in running the unit at rated load, particularly during summer months. So, keeping APH in good condition is extremely important for Utilities in view of its impact on heat rate and unit availability both. APH Performance assessment and analysis is a complex subject. The performance depends on gas and air flows through air heater, gas and air inlet temperatures, physical condition of baskets including fouling and conditions of APH seals etc. Some parameters like gas and air flows cannot be accurately measured and hence the impact of changes also can't be established. Flue gas & air temperature at APH outlet do indicate performance but in a limited way. It needs to be integrated with other indices like thermal efficiency, X ratio, leakage, air temp rise, gas temp drop, differential pressures (DP) etc. to establish whether deterioration in performance is because of deterioration in condition of APH internals or changes in extraneous factors like number of mills, air & gas flows, temperatures or PA header pressure etc.[22]

3.3.1 Air Leakage

Air heater leakage is the weight of air passing from the air side to the gas side of the air heater. This index is an indicator of the condition of the air heater's seals and increases with wear of seals. The increase in air heater leakage increases power requirements of the forced draft and induced draft fans, increasing unit net heat rate and at times limiting unit capacity. Air heater leakage is expressed as a percentage of gas flow entering the air heater.

$$AL = (\frac{(O_2gl - O_2ge)}{(21 - O_2gl})) * 0.9 * 100$$

where,

O_2ge = percent O_2 in gas entering air heater

O_2gl = percent O_2 in gas leaving air heater

Air heater leakage dilutes the flue gas and lowers the as measured exit gas temperatures. Gas outlet temperature corrected to no leakage condition is calculated using the following formula.

$$T_{gnl} = \frac{AL * C_{pa} * (T_{gl} - T_{ae})}{100 * C_{pg}} + T_{gl}$$

where,

T_{gnl}= gas outlet temperature corrected for no leakage

C_{pa}= mean specific heat between T_{ae} and T_{gl} = 1.023 kJ / kgK

C_{pg}= mean specific heat between T_{gl} and T_{gnl} = 1.109 kJ / kgK

T_{ae}= temperature of air entering air heater

T_{gl}= temperature of gas leaving air heater

3.3.2 Gas Side Efficiency

Air heater gas side efficiency is defined as the ratio of the temperature drop, corrected for leakage, to the temperature head, expressed as a percentage.Temperature drop is obtained by subtracting the corrected gas outlet temperature from the gas inlet temperature. Temperature head is obtained by subtracting air inlet temperature from the gas inlet temperature. The corrected gas outlet temperature is defined as outlet gas temperature calculated for 'no air heater leakage' and is given by following equation.

$$GSE = \frac{(T_{ge} - T_{gnl}) * 100}{T_{ge} - T_{ae}}$$

where,

T_{ae} = temperature of air entering air heater

T_{ge} = temperature of gas entering air heater

T_{gnl} = gas outlet temperature corrected for no leakage

Gas side efficiency is an indicator of the internal condition of the air heater. With wear of the baskets, ash pluggage etc., the air heater gas side efficiency decreases, accompanied by an increase in exit gas temperature and a decrease in air heater air outlet temperature.

3.3.3 X-Ratio

Air heater X-ratio is the ratio of heat capacity of air passing through the air heater to the heat capacity of flue gas passing through the air heater

$$X - ratio = \frac{T_{ge} - T_{gl}}{T_{al} - T_{ae}} = \frac{M_{al} * C_{pa}}{M_{ge} * C_{pg}}$$

$$X - ratio = \frac{Gas\ Side\ Efficiency}{Air\ Side\ Efficiency}$$

Air Side Efficiency (SA & PA): Ratio of air temperature gain across the air heater corrected for no leakage to the temperature head.

$$Air\ Side\ Efficiency = [\frac{(T_{al} - T_{ae})}{(T_{ge} - T_{ae})}] * 100$$

X-Ratio depends on the moisture in coal, air infiltration in the boiler, air & gas mass flow rates through the air heater and specific heats of air & flue gas.

Chapter 4

Exergy Analysis of Boiler and APH

Useful concepts of exergy analysis of boiler are discussed. Equations proposed by Shieh and Fan for finding exergy of fuel, air, water, steam and flue gas are mentioned in this chapter. Second Law efficiency and irreversibility of coal fired boiler are discussed here.

4.1 Exergy Analysis

4.1.1 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 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 [23].

Therefore, at steady state

- Exergy in - Exergy out = Exergy destroyed

4.1.2 Second Law Efficiency & Irreversibility

A common measure of energy use efficiency is the first law efficiency $\eta_{\mathbf{I}}$. 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 assignnig a quality index to energy[23].

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 efficienct 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[23].

$$\eta_{\mathrm{II}} = \frac{Minimum\ exergy\ intake\ to\ perform\ the\ given\ task}{Actual\ exergy\ intake\ to\ perform\ the\ given\ task}$$

$$\eta_{\mathrm{II}} = \frac{A_{min}}{A}$$

Where A is the availability or exergy.

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

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.

Mathematically,

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

4.1.3 Exergy Formulation for Thermal Power Plant

The thermodynamic 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[24].

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

$$\sum \dot{m}_i = \sum \dot{m}_o$$

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

$$\sum E_t + Q = \sum E_o + \dot{W}$$

The balance of entropy for a control volume system is

$$\sum \dot{S} + \sum \frac{Q}{T} + \dot{s}_{gen} = \sum \dot{s} + \sum \frac{Q}{T}$$

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

$$\sum \dot{E}x_t + \sum (1 - \frac{T}{T_k}) Q_k = \sum \dot{E}x_o + \dot{W} + \dot{E}x_d$$

where stream exergy rate is

$$\dot{E}_x = \dot{\text{m}}(e_x)$$

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

The exergy balance explained 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

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

When combustion occurs in the boiler, coal is burned to form carbon dioxide, water vapour 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

$$5.08C + 2H_2 + 6.08(O_2 + 3.76N_2) = 5.08CO_2 + 2H_2O + 22.86N_2$$

The exrgy of coal for the mass of carbon in coal is written as

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

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

$$\eta_{energy} = \frac{W_{output}}{m_f\, x\, CV}$$

$$\eta_{exergy} = \frac{W_{output}}{E_{x,coal}}$$

The formulas to find out exergy of these various components is given by Pradeepsingh Hada and Ibrahim Hussain Shah [17] are:

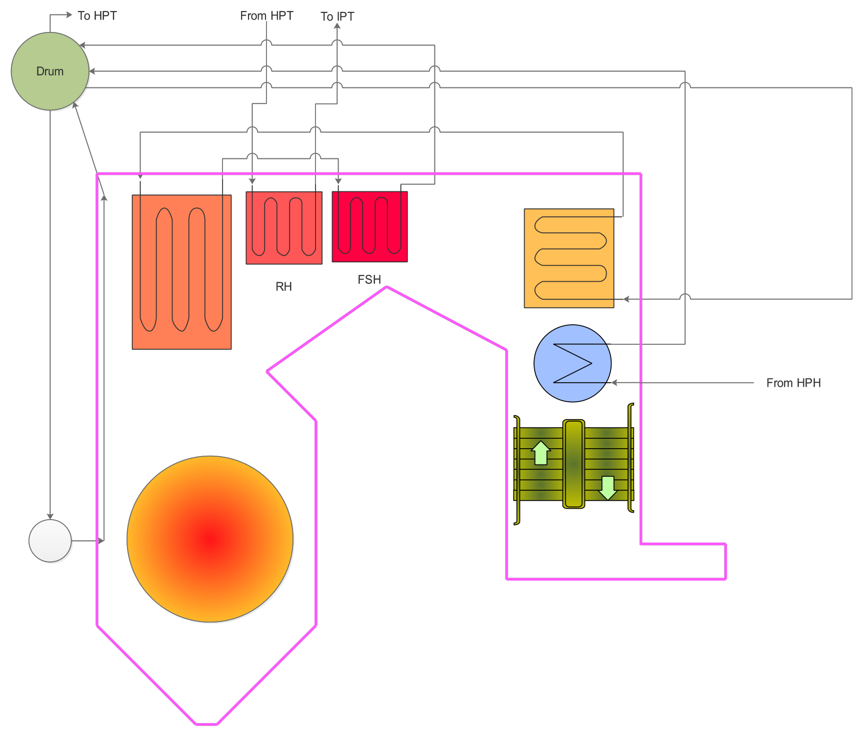


Figure 4.1: Boiler

4.1.3.1 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 given in [17]. Specific exergy of fuel is,

$$\varepsilon_f = 34183.16(C) + 21.95(N) + 11659.9(H) + 18242.90(S) + 13265.90(O) \text{ kJ/kg}$$

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

4.1.3.2 Exergy of Water Supplied

Specific exergy of water is given by,

$$\varepsilon_w = (C_p)_w[(T_w - T_o) - T_o ln(\frac{T_w}{T_o})] \text{ kJ/kg}$$

where,

T_w = Temperature of feed water

To = Reference temperature

$(C_p)_w$ = Specific heat of water at constant temperature

4.1.3.3 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. Specific exergy of air is given by,

$$\varepsilon_a = (C_p)_a[(T_a - T_o) - T_o ln(\frac{T_a}{T_o})] \text{ kJ/kg}$$

where,
T_a =Temperature of air
To =Reference temperature
$(C_p)_a$ = Specific heat of air at constant temperature

4.1.3.4 Exergy Available with Steam

Specific exergy of steam is given by,

$$\varepsilon_s = (h - h_o) - T_o(S - S_o) \text{ kJ/kg}$$

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

4.1.3.5 Exergy Available with Flue Gas

Specific exergy of flue gas is given by,

$$\varepsilon_g = (C_p)_g[(T_g - T_o) - T_o ln(\frac{T_g}{T_o})] \text{ kJ/kg}$$

where,
T_g = Temperature of flue gas
To = Reference temperature
$(C_p)_g$ = Specific heat of flue gas at constant temperature

4.2 Exergy Analysis of Air Preheater

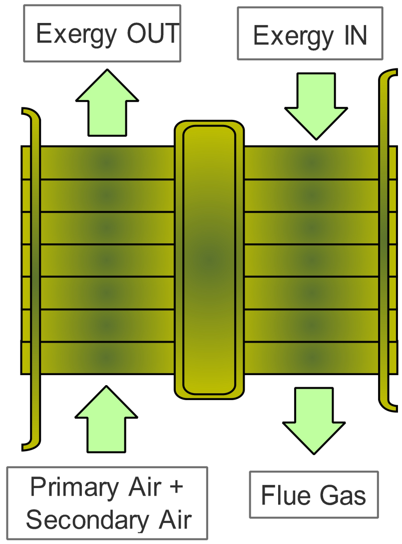


Figure 4.2: Exergy In and Out of Air Preheater

Figure 4.2 shows flow of exergy in and out of air preheater. Following are the assumptions made while exergy analysis of air preheater.

- Cycling working medium fluid behaves as a perfect gas, with constant specific heat i.e., For air 1.005 kJ/kgK and flue gas 1.2 kJ/kgK
- Assuming steady flow
- Neglecting K.E. and P.E. changes
- Neglecting air preheater leakages
- Air preheater is adiabatic overall
- A symmetric balanced APH
- No significant pressure drop across APH

Table 4.1: Air Preheater measured parameters

Parameters	Designed	Unit - 4	Unit - 5	Unit - 6
PA Mass Flow Rate (kg/s)	28.86	28.86	28.86	28.86
SA Mass Flow Rate (kg/s)	165.11	165.11	165.11	165.11
FG Mass Flow Rate (kg/s)	299	295	295	295
PA Temp In (K)	317	318	318	318
PA Temp Out (K)	612	548	558	562
SA Temp In (K)	317	311	311	311
SA Temp Out (K)	597	567	573	575
FG Temp In (K)	635	612	615	623
FG Temp Out (K)	423	410	416	393

Table 4.2: Second Law Efficiency and Irreversibilities of APH

	Designed	Unit - 4	Unit - 5	Unit - 6
Exergy in FG In (kW)	29372.36	26311.62	26276.71	29511.51
Exergy in PA Out (kW)	2832.294	1935.539	2068.231	2122.096
Exergy in SA Out (kW)	14950.03	12582.91	13054.51	13212.93
Irreversibility (kW)	11590.04	11793.18	11153.97	14176.49
Second Law Efficiency (%)	60.54	55.18	57.55	51.96

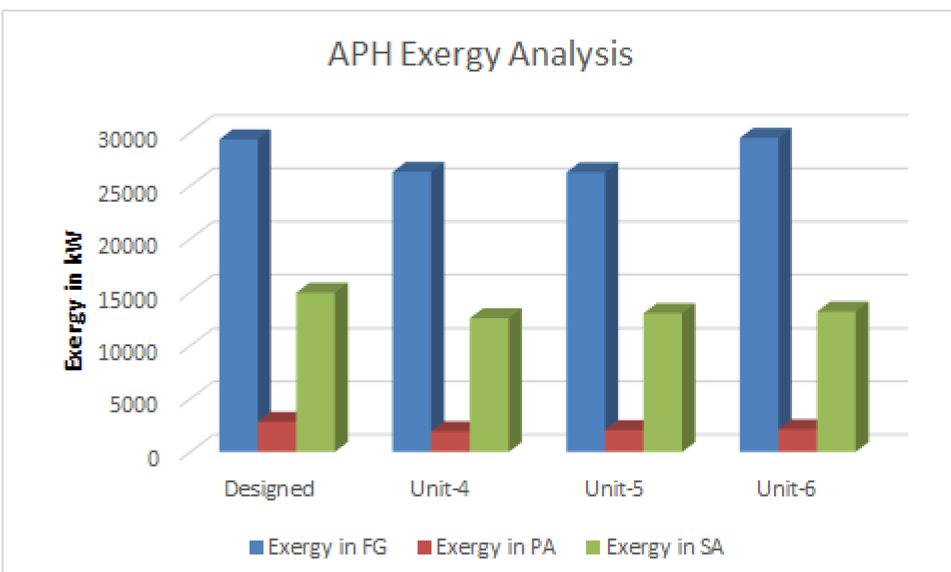


Figure 4.3: APH Exergy Analysis

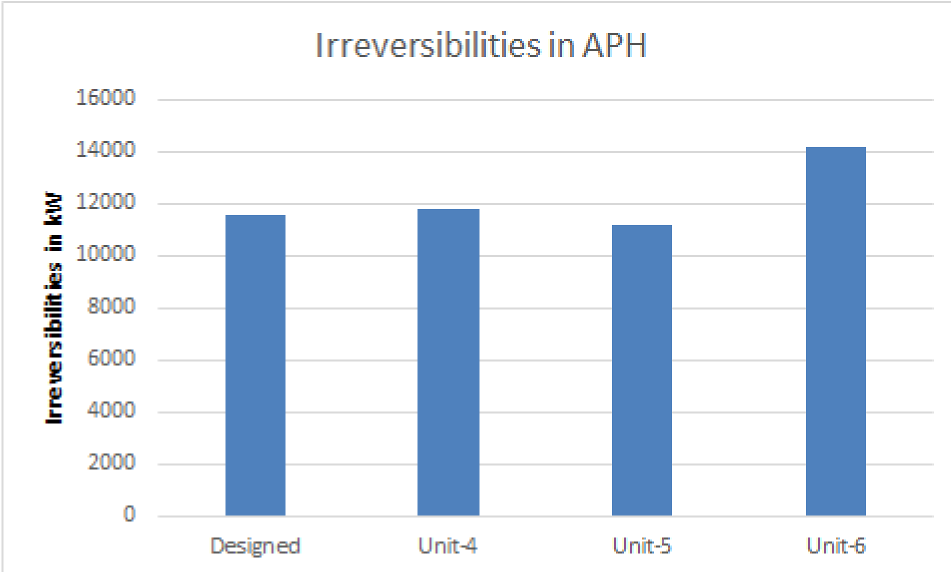


Figure 4.4: APH Irreversibilities

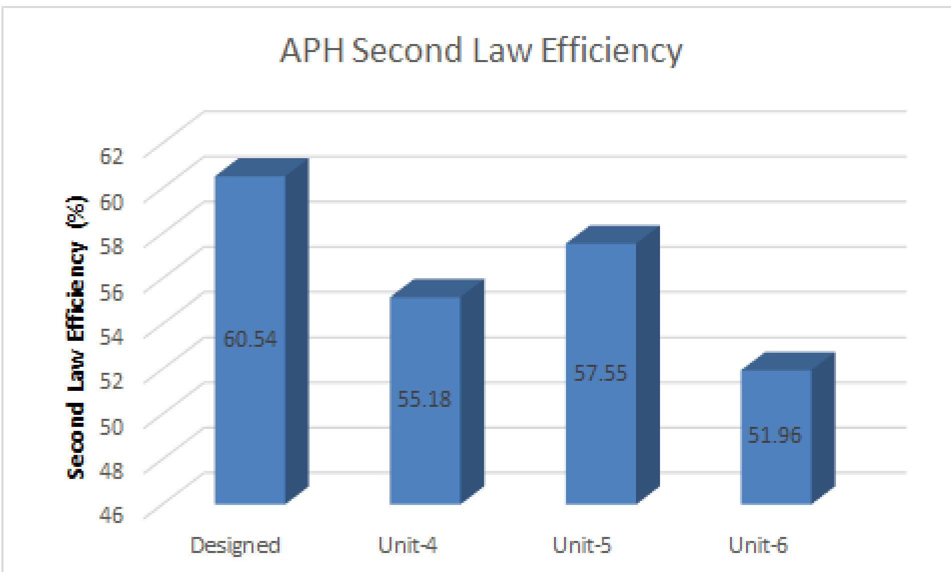


Figure 4.5: APH Second Law Efficiency

4.3 Exergy Analysis of Boiler

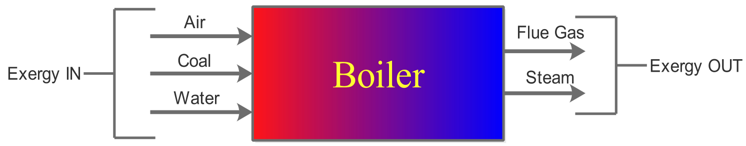


Figure 4.6: Exergy In and Out of Boiler

Following are the assumptions made for exergy analysis of boiler.

- Irreversibity due to streamwise conduction in the walls of boiler is neglected. The following readings are taken at full load i.e., 210MW for calculating second law efficiency and irreversibility of boiler.

Proximate analysis of coal used in units 4, 5 and 6 have been carried out in chemical laboratory, GSECL wanakbori and designed values of coal used in 210 MW plant have been taken from Technical Data Book published by GSECL wanakbori have been given in the following table 4.3:

Table 4.3: Proximate Analysis of Coal

	Designed	**Unit - 4**	**Unit - 5**	**Unit - 6**
% M	8	8	9	9
% A	32	32	32	30
% VM	15.25	17.2	15.48	17.3
% C	39	38	39	40
% S	0.64	0.51	0.5	0.55

Following formulas have been used for conversion of proximate analysis to ultimate analysis:

1. % C = 0.97C + 0.7(VM - 0.1A) - M(0.6 - 0.01M)

2. % H = 0.036C + 0.086(VM - 0.1A) - 0.035M^2(1-0.02M)

3. % N = 2.10 - 0.020VM

4. % O = 100 - (%C + %H + %N + %S + %A)

Ultimate analysis of coal carried out based on above formulas for conversion from proximate analysis have been given in the following table 4.4

Table 4.4: Ultimate Analysis of Coal

Constituents	**Designed**	**Unit - 4**	**Unit - 5**	**Unit - 6**
C (%)	42.11	42.65	42.84	43.63
H_2(%)	2.25	2.95	2.97	3.02
O_2(%)	6.76	5.53	5.68	5.77
S (%)	0.64	0.51	0.5	0.55
N (%)	1.80	1.84	1.83	1.83

Table 4.5 consist of mass flow rates values of different quantities such as coal flow, water flow, air flow, steam flow and flue gas flow entering and leaving the boiler.

Table 4.5: Mass Flow Rates of different quantities at inlet and exit of boiler

	Mass Flow Rate (kg/s)			
	Designed	**Unit - 4**	**Unit - 5**	**Unit - 6**
Coal	33.33	38.88	37.52	38.61
Water	194.44	186.31	183.89	178.33
Air	244.44	195.83	191.66	194.44
Steam	194.44	182.12	180.55	176.38
Flue Gas	277.78	171.38	174.44	236.11

Temperature readings of different quantities entering and leaving the boiler are taken from control room of respective units and shown in table 4.6 .

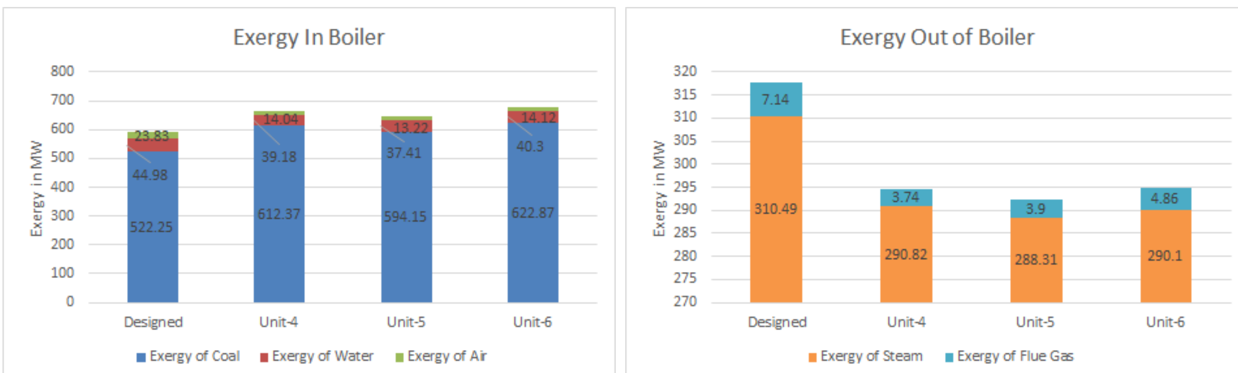
Table 4.6: Temperature of different quantities at inlet and exit of boiler

		Temperature (K)			
		Designed	**Unit - 4**	**Unit - 5**	**Unit - 6**
Water		518	506	502	515
Air	**PA**	611	558	556	561
	SA	603	573	572	573
Steam		808	808	808	808
Flue Gas		419	408.51	410	405

Exergy analysis of boiler has been carried out using the conditions i.e., temperature, mass flow rates and constituents of various quantities entering into the boiler and leaving out of the boiler. Specific exergy and exergy values of different quantities calculated are as shown in table 4.7

Table 4.7: Exergy Analysis of Boiler

	Units	Designed	Unit - 4	Unit - 5	Unit - 6
Exergy of Coal	**kJ/kg**	15669.10	15750.13	15835.48	16132.42
	MW	522.25	612.37	594.15	622.87
Exergy of Water	**kJ/kg**	231.30	210.30	203.45	225.98
	MW	44.98	39.18	37.41	40.30
Exergy of Air	**kJ/kg**	97.48	71.71	68.96	72.63
	MW	23.83	14.04	13.22	14.12
Exergy of Steam	**kJ/kg**	1596.87	1596.87	1596.87	1596.87
	MW	310.49	290.82	288.31	290.10
Exergy of Flue Gas	**kJ/kg**	25.69	21.81	22.35	20.58
	MW	7.14	3.74	3.90	4.86



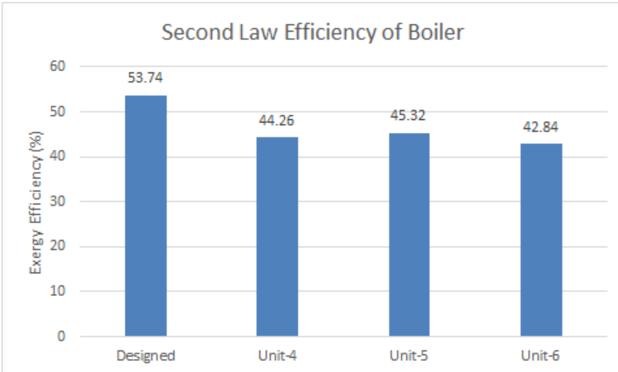
(a) Exergy Into Boiler (b) Exergy Out of Boiler

Figure 4.7: Boiler Exergy Analysis

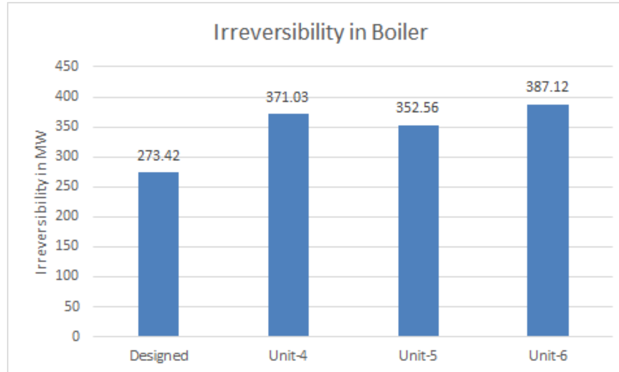
From the exergy values calculated in table 4.7 second law efficiency and irreversibility of different units of GSECL wanakbori have been calculated as shown in table 4.8.

Table 4.8: Second Law Efficiency and Irreversibility of Boiler

	Designed	Unit - 4	Unit - 5	Unit - 6
Second Law Efficiency (%)	53.74	44.26	45.32	42.84
Irreversibility (MW)	273.42	371.03	352.56	387.12



(a) Second Law Efficiency of Boiler



(b) Irreversibility in Boiler

Figure 4.8: Second Law Efficiency & Irreversibility of Boiler

Conclusion:

- The second law efficiencies of units 4, 5 and 6 comes out to be 44.26%, 45.32% and 42.84% while the designed second law efficiency for each of the units is 53.74%.
- Irreversibilities of units 4, 5 and 6 is found to be around 80 to 100 MW more than the designed irreversibility of 273.42 MW.

Chapter 5

Results and Discussions

Results of performance tests carried out to measure the performance of APH and air ingress in to boiler of unit-5 of WTPS are discussed in this chapter. Also, comparison between the results based on O_2 mapping before and after the overhauling is shown. Good operations and maintenance practices which drives improvement in the performance of the system are also listed.

5.1 Methodology adopted for O_2 mapping

Performance test of Flue Gas Path System by measuring oxygen at different location of flue gas path is conducted for measuring air ingress. Following calibrated instruments has been used to measure gas composition and other parameters :

1. Flue gas analyser for gas composition measurement by volume basis
2. Manometer for draft measurement
3. Temperature indicator for temperature measurement

Measurement is done on a flue gas duct by Grid Method (Shallow, Middle and Deep of the duct). Measurement points (tapping position) as per standard code are made before actual test being conducted. In each flue gas duct four points have been made from where sample has been taken. To calculate leakage percentage, path wise (Left and Right, Pass A to D) and segment wise calculation has been done for various locations and also overall air leakage calculation is also done by taking the average of all the values of different path.

Table 5.1: APH Parameters of Unit-5 Before Overhauling

Parameters	**Unit**	**Inlet A**	**Inlet B**	**Outlet A**	**Outlet B**
O_2	%	2.85	3.24	6.2	6.68
CO	PPM	1357	274	1325	392
NO	PPM	134	156	117	130
CO_2	%	16	16	14	13
SO_2	PPM	564	557	123	136
NO_x	PPM	141	164	485	504
Draft	mmWC	-59	-62	-140	-136
T_a	°C	33	33	270	283
T_g	°C	325	310	128	142

5.1.1 APH Performance Test of Unit-5

Measurement of various parameters related to thermal performance of APH and flue gas analysis before and after overhauling is as shown in table 5.1 and 5.2.

Observations:

- Average measured percentage air ingress in APH is 21% against the permissible value of 12.0
- Average flue gas corrected temperature for no leakage is 156°C which is well within permissible value of 156.7°C.
- Pressure drop measured across the APH-A and APH-B on gas side is 81mmWC and 74mmWC respectively against permissible value of 70mmWC.

The overhauling of the APH includes the following :

- Cleaning of APH hopper of debris and ash.
- Replacement of radial and axial seals.
- Cleaning of cold end baskets and replacement of damaged baskets.
- Water cleaning of APH when the heater water temperature is at 80-90°C to remove sulphur deposits.

Table 5.2: APH Parameters of Unit-5 After Overhauling

Parameters	Unit	Inlet A	Inlet B	Outlet A	Outlet B
O_2	%	3.01	3.05	5.39	5.06
CO	PPM	1244	357	1205	381
NO	PPM	226	225	246	239
CO_2	%	15	15	13	14
SO_2	PPM	577	484	118	156
NO_x	PPM	237	163	147	256
Draft	mmWC	-60	-62	-144	-135
T_a	°C	30	30	280	295
T_g	°C	311	322	124	136

Results:

1. There is a significant reduction in air ingress into the APH from 21% to 12.5% (i.e., 40 % reduction).
2. Gas side efficiency is improved from 56% to 61% after overhauling.
3. There is also improvement in air temperature rise at APH inlet and gas temperature drop at APH outlet.
4. Over and above the APH performance is restored well up to permissible limits. Comparison of various indices before and after overhauling is shown in table 5.3.

5.1.2 Air Ingress measurement of unit-5

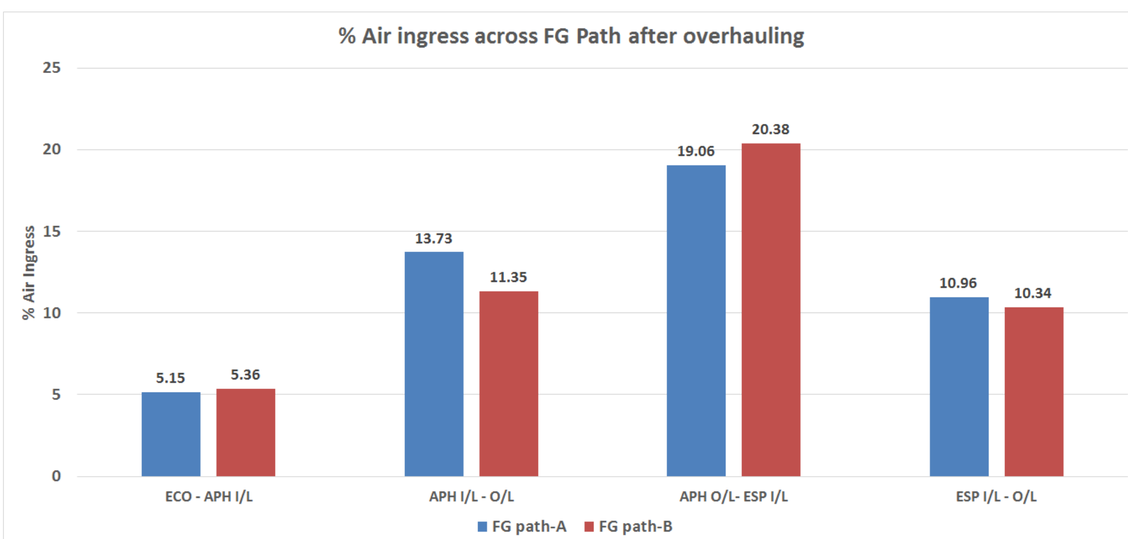


Figure 5.2: Air Ingress across FG Path after overhaul

Table 5.3: APH Performance Indices Calculated Before and After Overhauling

APH Performance comparison				**Before**			**After**		
Parameters	**Unit**	**Designed**	**Permissible**	**APH A**	**APH B**	**AVG**	**APH A**	**APH B**	**AVG**
AL	%	8	12	20.37	21.62	21	13.72	11.34	12.5
T_{gnl}	°C	146.8	156.7	148	165	156	137	147	142
GSE	%	62.6	58.7	61	52	56	62	60	61
X-ratio	-	0.83	0.76	0.75	0.57	0.67	0.7	0.66	0.68
Air Temp Rise	°C	284	220	237	250	243.5	250	265	257.5
Gas Temp Drop	°C	204	200	197	168	182.5	187	186	186.5

Table 5.4: Air Ingress Measurement Before Overhauling

Parameters	Average Readings of Unit-5 Before Overhaul													
	-	**ECO Outlet**	**APH Inlet**		**APH Outlet**		**ESP Inlet**				**ESP Outlet**			
	Unit	ECO	APH A	APH B	A	B	A	B	C	D	A	B	C	D
O_2	%	1.98	2.85	3.24	6.2	6.86	8.23	8.9	9.63	9.41	10.92	9.48	10.29	11.37
Draft	mm WC	-17	-62	-64	-144	-148	-194	-194	-202	-202	-225	-226	-228	-227
FGT	°C	594.5	319	327	151	139	130	140	139	133	125	133	128	122
BETWEEN		**ECO Outlet - APH Inlet**			**APH Inlet-Outlet**		**APH Outlet - ESP Inlet**				**ESP Inlet - Outlet**			
-	-	A		B	A	B	A	B	C	D	A	B	C	D
Air Ingress	%	4.315		6.390	20.38	23.05	14.31	20.10	21.93	19.80	24.02	4.53	5.55	18.32
Average Air Ingress	%	5.36			21.71		19.035				13.105			
Pressure Drop	mm WC	45		81	82	84	50	50	54	54	31	32	26	25
Temperature Drop	°C	275.5		267.5	168	188	21	11	0	6	5	7	11	11

Table 5.5: Air Ingress Measurement After Overhauling

Para-meters	**Average Readings of Unit-5 After Overhaul**													
	-	**ECO Outlet**	**APH Inlet**		**APH Outlet**		**ESP Inlet**				**ESP Outlet**			
	Unit	ECO	APH A	APH B	A	B	A	B	C	D	A	B	C	D
O_2	%	1.21	3.01	3.05	5.39	5.06	8.3	7.93	7.8	8.2	10.12	8.85	9.28	9.4
Draft	mm WC	-22	-60	-62	-144	-135	-164	-165	-163	-164	-178	-176	-168	-174
FGT	°C	442	311	322	127	136	119	124	134	125	118	123	130	117
BETWEEN		**ECO Outlet - APH Inlet**			**APH Inlet-Outlet**		**APH Outlet - ESP Inlet**				**ESP Inlet - Outlet**			
-	-	A		B	A	B	A	B	C	D	A	B	C	D
Air Ingress	%	5.15		5.36	13.73	11.35	20.62	17.5	18.68	22.08	15.06	6.82	11.37	9.31
Average Air Ingress	%	5.26			12.54		19.718				10.64			
Pressure Drop	mm WC	38		40	84	73	20	21	28	29	14	11	5	10
Tempera-ture Drop	°C	131		120	184	186	8	3	2	11	1	1	4	8

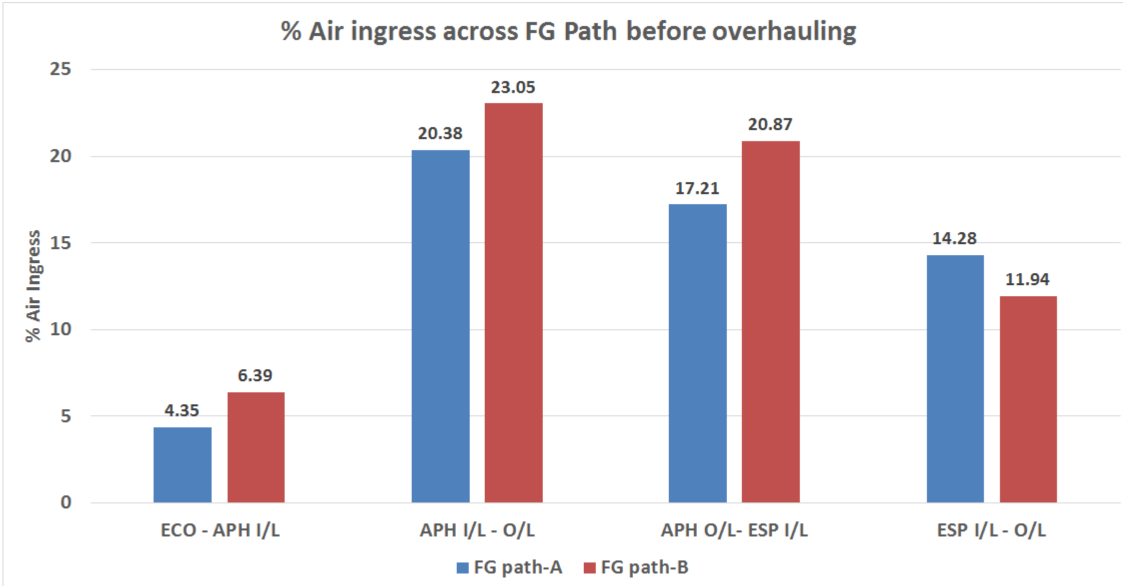


Figure 5.1: Air Ingress across FG Path before overhaul

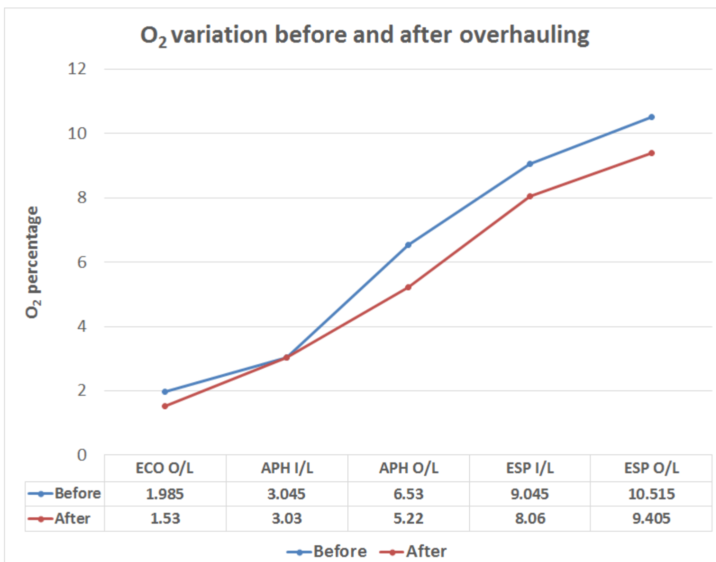


Figure 5.3: O_2 Variations before and after overhaul

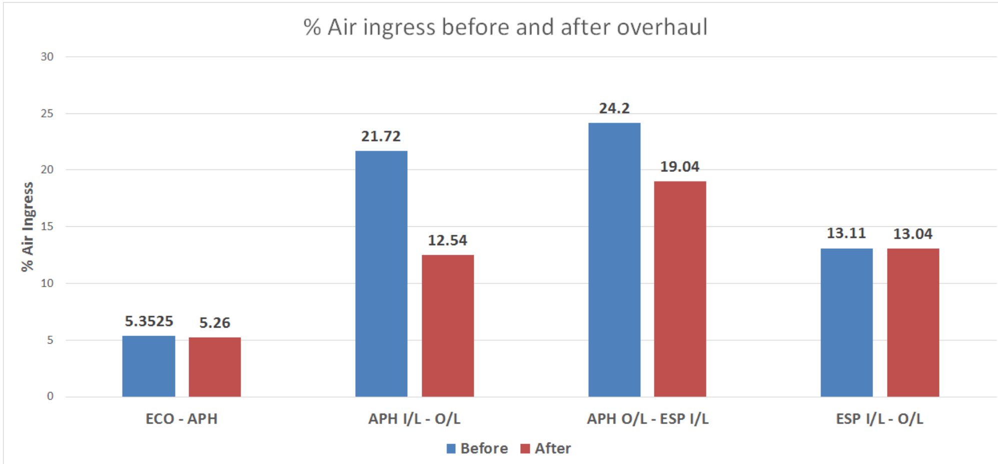


Figure 5.4: Air Ingress comparison before and after overhaul

Conclusions:

- It is observed that Flue Gas Path is suffering from heavy air ingress before overhaul.
- O_2 percentage at Economizer Inlet is 1.98% and jumped to 11.37% at ESP Outlet.
- It is also observed that air ingress is very predominant between APH Outlet and ESP Inlet
- APH itself is suffering from huge ingress and oxygen percentage jumps from 3.24% to 6.86%.
- There is marginal Air Ingress between Economizer Inlet and Economizer Outlet.
- Temperature Drop from APH outlet to ESP outlet i.e. 151°C to 122°C, which is very high and concludes heavy air ingress.

5.1.3 Economical gain after overhauling

After overhauling economical analysis is carried out using the readings taken from control room as shown in table 5.6.

Table 5.6: Comparison of Auxiliary Consumptions

Sr. No.	**Parameters**	**Unit**	**Before**		**After**	
			APH-A	**APH-B**	**APH-A**	**APH-B**
1	Unit Load	MW	210		210	
2	Fuel Consumption	T/hr	137		134	
3	PA fan current	amps	128	130	124	124
4	FD fan current	amps	33	29	30	31
5	ID fan current	amps	135	128	108	110
6	Total consumption of fans	amps	296	287	262	265
7	Total fan loading	amps	583		527	

Profit Gained After Overhauling:

A) Cost saving per hour :

1. Fuel saving = 137 - 134 = 3 T/hr
2. Estimated coal cost = Rs 4500 per MT
3. Cost saving per hour = 4500 x 3 = Rs 13500
4. Cost of fuel saving per year = 13500 x 24 x 365 = Rs. 11,82,60,000

B) Cost benefit due to fan loading :

1. Total fan loading before overhauling = 583 amps
2. Total fan loading after overhauling = 527 amps
3. Total saving in current = 56 amps
4. Power saved = ($\sqrt{3}$x V x I x PF)/1000
   = (1.73 x 6600 x 56 x 0.86)/1000
   = 549.9 kWh
5. Energy saved per year = 549.9 x 24 x 365
   = 48,17,124
6. Power cost = Rs 2.60 per unit
7. Saving per year = 4817124 x 2.60
   = Rs 1,25,24,522.4

C) Total savings per year = 11,82,60,000 + 1,25,24,522.4
   = Rs 13.07 crores

Conclusion :

1. Thermal Performance of Air Preheater is improved.
2. Air ingress from economiser to air preheater outlet reduces but even after patching and repairing work on ducts between APH and Economiser their is no significant reduction in air ingress.
3. After overhauling work of ESP have been carried out along with replacement of worn parts its air ingress reduces from 13.11% to 10.64%.
4. Load on the fans are reduced thus power consumption reduced and hence operating cost also reduces.

Chapter 6

Conclusion & Future Scope

This chapter includes conclusions drawn from the performance evalution of air preheater and air ingress measurement of unit-5, the results of exergy analysis of air preheater and boiler of units 4, 5 and 6 of GSECL wanakbori are also listed here. Future scope of this work is also pointed out.

6.1 Conclusion

In the present work, performance evaluation using O_2mapping of APH of unit-5 (210MW) of GSECL wanakbori have been carried out. Following conclusions are drawn from the same:

- Air leakage percentage of unit-5 comes down to 12.5% after replacement of seals, cleaning & repairing work compared to 21% before overhaul.
- After capital overhauling, fuel consumption of unit-5 is reduced which results in saving of Rs.11.8 crores per annum. It not only brought down the coal consumption but also reduced emission.
- Consumption of PA, FD and ID fan is reduced by 56 amps for unit-5, which will result in saving of Rs.1.25 crores annually.

Exergy analysis of boilers and air preheaters of units 4, 5 and 6 of GSECL wanakbori have been carried out. Following conclusions have been drawn from the analysis:

- Second law efficiencies of boilers of units 4, 5 & 6 are found to be 44.24%, 45.32% & 42.84% while designed second law efficiency is 53.74%.

- Irreversibility of different units have been calculated and found to be 80 to 100 MW more than the designed values of irreversibility of each units.

- Second law efficiency of APH of units 4, 5 and 6 of GSECL wanakbori is 55.18%, 57.55% and 51.96% respectively while the designed value of second law efficiency is 60.54%. As the capital overhauling of unit-6 have not been carried out since couple of years, it resulted into more irreversibility compared to other two units and lower second law efficiency accordingly.

6.2 Future Scope of Present Work

- Since the heating elements profiles used in existing air preheater has been designed nearly three decades ago. Computational Fluid Dynamic can be used for the performance comparison by designing various element profiles and also by comparing performance of elements by changing their materials.

- Ljungstrom Rotary Air Preheater consist of the conventional radial, axial, circumferential and bypass seals. Impact on performance of air preheater can be studied by replacing that seals with either flexible seals or by installation of seals made up of advance materials.

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