Performance assessment and mass energy balance for regenerative type air heaters

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DECLARATION

I Daniël Enslin Swart (Identity Number: 8602285166083) hereby declare that the work contained in this dissertation is my own work. Some of the information contained in this dissertation has been gained from various journal articles; text books etc. and has been referenced accordingly.

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ABSTRACT

The rapid development in South Africa over the last decade and the absence in new-build power plants have put enormous pressure on Eskom to reduce unplanned outages and increase the efficiency and reliability of the power plants.

During the development of this study, Eskom was implementing measures to increase the reliability of their power plants through various measures, from coal quality to maintenance strategies and plant optimisation. One part of the plant that attracted most of the focus was the draught plant, which consists of the air heaters, FD, ID and PA fans. Even though not a full part of the draught plant was the fabric filter plant, but this plant had a direct influence on the overall performance of the draught plant.

The air heaters at Arnot power station was the focal point of the optimisation as it was struggling with ID fan capacity and ID fans operating in stall condition.

An air heater performance test was conducted at Arnot power station to determine the current state of the air heaters after the Arnot capacity increase program was concluded in 2006. The results from the performance test were compared to the original design data specifications to determine the relevant deficiencies which contributed to overall performance issues for the draught plant.

The aim of this dissertation is to develop a model that can be used to determine the performance of the air heaters during different operational condition without the hassle of a full performance test.

It is also aimed to demonstrate that the variable of the air heater within itself is not only the contributing factor of underperformance but that the boiler also plays any important role. It is also intended to demonstrate the importance of proper combustion within the boiler as well as proper maintenance of the air heaters.

The results from the performance test and the air heater simulation model correlated very well with each other. This correlation shows that the model can be used for on load performance assessment by system engineers.

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KEYWORDS

Regenerative

Air heater

Ljungström

Rothemuhle

Performance assessment

Fans

Mass flow

Temperature

Efficiency

Power station

Heat transfer

Testing

Design data

Evaluation

Sealing

Leakage

Mass energy balance

Packs

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Howden Engineering Manager

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Glossary

Diluted gas temperature

The diluted gas temperature is defined as the observed or measured exit gas temperature and include the dilution effect of air leakage through the air heater seals from the air side to the gas side.

Undiluted gas temperature

The undiluted gas temperature is defined as the temperature at which the flue gas would leave the air heater if there was no air leakage from the air side to the gas side

Gas temperature drop

The gas temperature drop is defined as the gas temperature entering the air heater minus the undiluted gas temperature leaving the air heater

Air temperature rise

The air temperature rise is defined as the air temperature leaving the air heater minus the air temperature entering the air heater

Air heater air in-leakage

Air heater air in-leakage is defined as the mass of air leakage to the flue gas side divided by the mass of wet gas entering the air heater, in accordance with ASME PTC 4.3

Air leakage

Amount of air passing from the air side to the gas side, assumed to be passing directly from the air inlet to the gas outlet

Gas side effectiveness

Ratio of the gas temperature drop, to the difference between the air inlet and gas inlet temperature

Undiluted gas exit temperature

The temperature at which the gas would have left the air heater if there were no leakage

Nomenclature

$\dot{M}$	Mass flow
ρ	Density
P_{abs}	Absolute pressure
R	Gas constant
ρ_g	Specific gas density
T	Temperature
Q	Volume flow
P_s	Static pressure
P_{sf}	Fan static pressure rise
ΔP_{sf}	Fan static pressure
V	Velocity
P_v	Dynamic pressure
A	Area
k	Moisture content
η	Effectiveness
Cp_a	Specific heat air
Cp_g	Specific heat gas
O_{2out}	Oxygen % outlet
O_{2i}	Oxygen % inlet

FFP	Fabric filter plant
ID	Induced draught fan
FD	Force draught fan
PA	Primary air fan
$ACIP$	Arnot capacity increase project
MWe	Electrical power generated
$°C$	Degrees Celsius
kW	Kilowatts
kg/s	Kilograms per second
m^3/s	Cubic meters per second

Subscripts

a	Air side
g	Gas side
1	Inlet condition
2	Outlet condition
NL	No leakage

1 Chapter 1: Introduction

1.1 Background

Eskom, a state-owned company is the only power generation utility in South Africa. The utility makes use of various methods for generating power namely; Coal-fired stations, Hydro electrical, Nuclear and gas turbines. Eskom fleet consists of 13 old coal-fired stations and 2 new build coal fired stations.

All of the coal-fired stations are fitted with 2 air heaters per boiler. The stations make use of either a Rothemuhle or Ljungström air heater. That said there are over 150 air heaters operating within the Eskom fleet.

Air heaters perform a critical part of the modern-day power plant, reducing the amount of heat loss from combustion and increasing the overall efficiency of the boiler. The boiler efficiency can be increased by as much as 1% for every 22°C rise in combustion air temperature. The performance of the air heater plays a vital role in the combustion process as heat input and absorption in the boiler are affected by the temperature of combustion air.

There are a few reasons why air heaters are not performing as per design:

- Blockages of the air heater packs due to either boiler tube leaks, insufficient soot blowing or fuel oil carry over during light up. Blockage increases the differential pressure over the air heater packs. The increase in pressure can be seen by the saturation of the ID fans.
- Worn or not the proper setup of seals or wear shoes. The seals are the major culprit to air heater leakage. If the seals do not make proper contact with the heater leakage of air from the high-pressure air side to the low-pressure gas side occurs.
- Worn out packs can also have a big influence on the performance. The packs designed for a lifetime of up to 10 years in normal condition, but due to the high demands the pack need to be inspected regularly and take out to be weighed to determine the loss of mass in the material.
- Soot blowing of the packs goes hand in hand with pack blockages. If soot blowing is not done according to the prescribed schedule blockage occur as described in the first point above. Soot blowing needs to be done at the correct temperature and pressure.

The aim of this dissertation is to evaluate the performance of the air heater as well as develop an on-load assessment tool to assess the efficiency of the air heaters during operation without the necessity of a full performance test.

The assessment tools need to give relevant data to the system engineer that can be used in the process of fault finding or combustion optimization.

This study forms part of the combustion model by Prof C.P. Storm. From the combustion model the performance of the air heater can be evaluated according too different operational and combustion conditions.

1.2 Problem statement

To identify, investigate and evaluate the performance of a regenerative type air heater using thermodynamic mass-energy balance model as well as derive a program that can be used during operational condition to evaluate the efficiency of the air heater.

1.3 Objective

- To develop an excel model to that would incorporate all the parameters for evaluating the performance of the air heater
- To validate the excel model with design data from Arnot power station
- To conduct a full performance test on boiler 3 at Arnot power station and use the results in the excel model
- To evaluate the performance of the air heater
- To derive recommendation from the assessment on problems and issues in the plant\
- To recommend future studies regarding air heater performance

1.4 Method of investigation

- A simulation model of the air heater mass and energy balance was identified as the first objective of this study.
- The mass and energy balance model would then be used to verify the air heaters on load performance.
- Obtain data from performance test of boiler 3 draught plant
- Evaluate current performance with performance from design data and elaborate

1.5 Limits of study

- The simulation is done for Arnot power station but can be used at any other power plant that uses regenerative type air heaters
- The study was only done for the air heaters and not all plant is covered; mills, burners; fans
- The study is done on an as ease boiler and not cleaning was done before hand except for soot blowing of packs.

1.6 Dissertation structure

Chapter 1

Chapter 1 discusses the information regarding the initiation of this study. It gives a description of the current status of the power plants in South Africa. It also summarizes the structure of the dissertation.

Chapter 2

Chapter 2 discusses the surveys that were done on the topic of air heaters with all relevant aspects of this study as well as air heaters as a whole. It also discusses the numerical evaluation that was previously done by Eskom. The survey covers the different types of air heaters, the type of sealing arrangement in the different air heaters, the common leakage paths and the directed and entrained leakage.

Chapter 3

Chapter 3 discusses the plant setup at Arnot power station with the full layout of equipment and air heater arrangement. It also explains the limitations that were pointed out during the ACIP upgrades and ACIP performance assessments.

Chapter 4

Chapter 4 show the method used in formulating the assessment program using thermodynamics and the mass-energy balance.

Chapter 5

Chapter 5 explain the process followed for the plant performance test. It also describes the location of test points and the standard of conducting performance tests in power plants.

Chapter 6

Chapter 6 is the validation of the model with data from the Howden performance assessment tools and the discussion of the results.

Chapter 7

The focus of this chapter is to assess the performance of the air heater by using the excel model that was created. The cycle simulation was done firstly for the design data and secondly for the performance test data in order to compare the performance of current operational condition to that of the design condition.

Chapter 8

Chapter 8 cover the conclusion and recommendations of the study.

Chapter 9

Include the appendix as well as the raw data from the performance test. The combustion model to be used for boiler optimization can also be found in the appendix.

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2 Chapter 2: Literature Survey

2.1 Introduction

The literature survey conducted for this study has been divided into three major topics, i.e. types of regenerative air heaters, regenerative heating elements, sealing arrangement of different air heaters and auxiliary systems for air heaters.

The purpose of the types of regenerative air heater study was to illustrate the different in rotating packs and rotating hoods arrangement.

The regenerative heating elements study illustrates the different heating profiles and material.

The literature survey regarding the sealing arrangement was done to show the types of seals and the difference of leakage between air and air, air and gas, and gas and gas.

2.2 Air heater types

2.2.1 Description of air heater

Air heaters are found in most steam generating plants to heat up the combustion air and to increase the overall thermal efficiency of the combustion process. In most of the applications flue gas from the combustion of coal serves as the heat or thermal energy source. The air heater is seen as a heat trap which collects the heat from the flue gas stream as it rotates. It then transfers the heat to the cold air coming in from the FD fan to the boiler. This process of increasing the temperature of the combustion air can increase the overall boiler efficiency by up to 10%. The hot air from the air heater not only increase the efficiency of the boiler but is also use to dry and transport the pulverized coal to the boiler. (Kitto, 2005)

Drobnic et al (2006) stated that the air heater has a very important influence on the overall efficiency of the boiler and due to the compact design of the air heater and the high thermal performance they are commonly found in coal-fired boilers. This high thermal performance assists in minimising the heat loss from flue gas, also known as dry flue gas losses.

All air heaters have a general operational design fault know as air heater leakage. Air heater leakage occurs when high-pressure air flows through a sealing system and leaks into a low-pressure system. The quantity of leakage is very dependent on the sealing setup and pressure differential between the air and the gas streams.

2.2.2 Recuperative heat exchanger

In a typical recuperative heat exchanger, heat is transferred continuously through stationary heat transfer tubes that separate the hot gas from the cold air. Recuperative heat exchangers have the advantage of operating with very small amount of leakage due to the nature of the setup and zero rotational parts. An expansion joint between the tube sheet and heater casing provides an air/gas seal to prevent leakages. In certain power stations, the primary air is heated in the tubular air heater by means of hot flue gas. The flue gas is transferring heat from within the tubes to the cold combustion air on the outside of the tubes. These air heaters are mostly used in a power station as primary air heaters, feeding air to the mills via the PA fans. Figure 1 shows the typical arrangement of a tubular air heater. (Kitto, 2005:20-7)

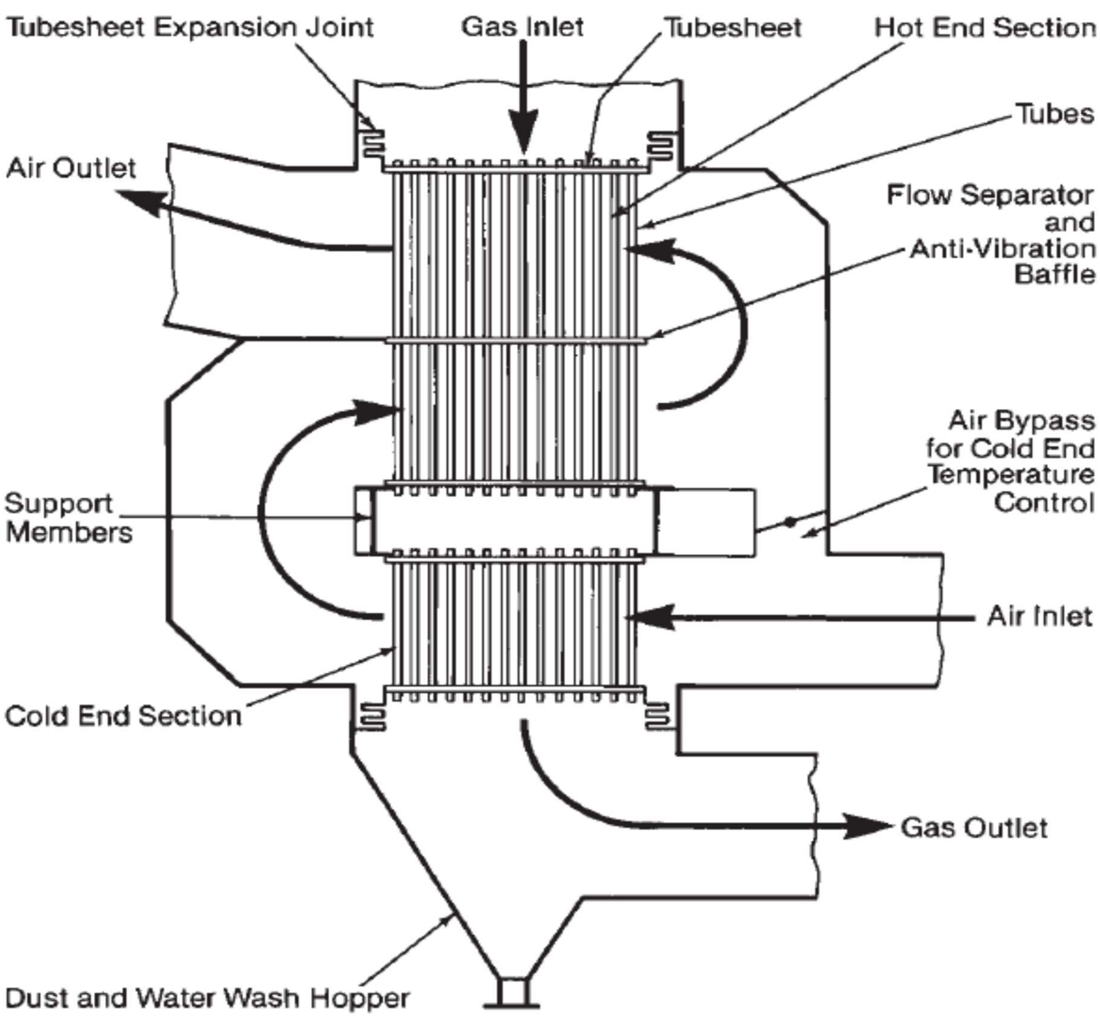

Figure 1 : Vertical tube air heater(Kitto 2005)

2.2.3 Regenerative heater exchangers

A regenerative type heat exchanger transfer heat by convection as it storage medium is continuously exposed to either hot flue gas or cold air. A variety of materials can be used for the storage of the heat; the transfer of heat is achieved by the rotating shaft. In the power industry, they make use of a tightly packed pair of plates to transfer the heat. As the air heater rotate it exposes the pack to either the air stream or the gas stream. Heat is absorbed in the gas stream and heat transfer is taking place in the air stream. This type of heat exchanger can either have a rotating pack assembly or a rotating hood. The difference will be discussed in the headings below. (Kitto, 2005:20-8)

2.2.4 Ljungström air heaters

Throughout the history of boilers, there has been much advancement in the area of operational improvement and reduction in fuel cost. But few of those inventions were as successful as the Ljungström air heater invented by Fredrik Ljungström, technical director at Aktiebolaget Ljungström Angturbin (ALA). The first Ljungström air heater that was installed in a commercial boiler saved as much as 25% of the fuel consumption. In a modern-day boiler, the Ljungström air heater contributes to about 20% of the total heat transfer area in the boiler process, but only contributes to 2% of the total investment. (ASME, 1995:3)

The Ljungström air heater is a remarkable invention in many ways, till 1994 about 20000 air heaters have been supplied to various parts of the world. It has been estimated that the total operating hours for all Ljungström air heaters to be at 1,500,000,000. (ASME, 1995:3)

The Ljungström air heater is a regenerative type heat exchanger that makes use of slow rotating packs that is used for heat transfer. The arrangement of the air and gas duct is in such a way the half of the heater is in the gas stream and the other half in the air stream which supply combustion air to the boiler. The hot flue gas heats the part of the rotor as the gas flows over the element packs. The rotor then turns the heated section into the path of the air to heat up the combustion air. The rotor is divided up into sections that pass through seals to prevent flue gas from entering into the combustion air.

The rotor of the Ljungström air heater is very large and can vary between 8m to 13.5m in larger applications with a debt of up to 2.5m and a total weight of 1100tons. The flue gas normally enters the heater at 340°C and cooled down to 145°C. This temperature may vary in accordance with the type of fuel used in the boiler. The gas flow rate through the heater are of the order of 2 million m^3/h through both sides of the heater, and the temperature effectiveness, the difference between the temperature drop and the maximum available temperature difference, is about 85%. One advantage of the Ljungström is that the flue gas temperature can be reduced to sulphuric acid dew point temperature without affecting the heat transverse performance. But with recuperative type heaters in which the heat flows through a separation wall, the efficiency drops off very rapidly as the temperature drop below dew point as the deposit form on the heat transfer area. (ASME, 1995:4)

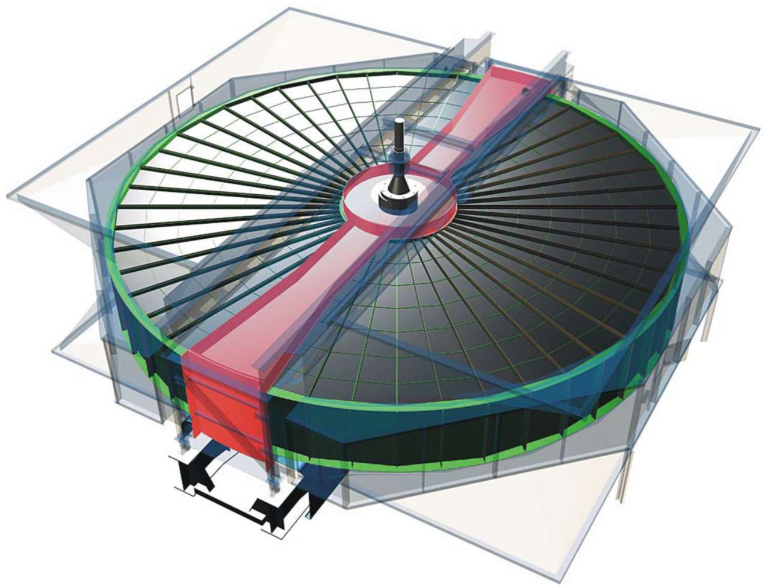

Figure 2 : Ljungström air heater (Howden,2014)

In the vertical shaft arrangement as shown in figure 1, the flue gas enters the air heater from the top or the hot end side and the cold air from the bottom or the cold end side. This is also called contraflow heater arrangement. During operational conditions, the air heater experiences a temperature differential between the hot and cold end casing the rotor to distort, or cap as it is known by. This form of capping creates a gap between the stator and the rotor allowing air to leak into the flue gas stream. Leakage will be further explained in this chapter. The cleaning of the air heater takes place by installing a soot blowing system. The system makes use of super-heated steam to clean the packs

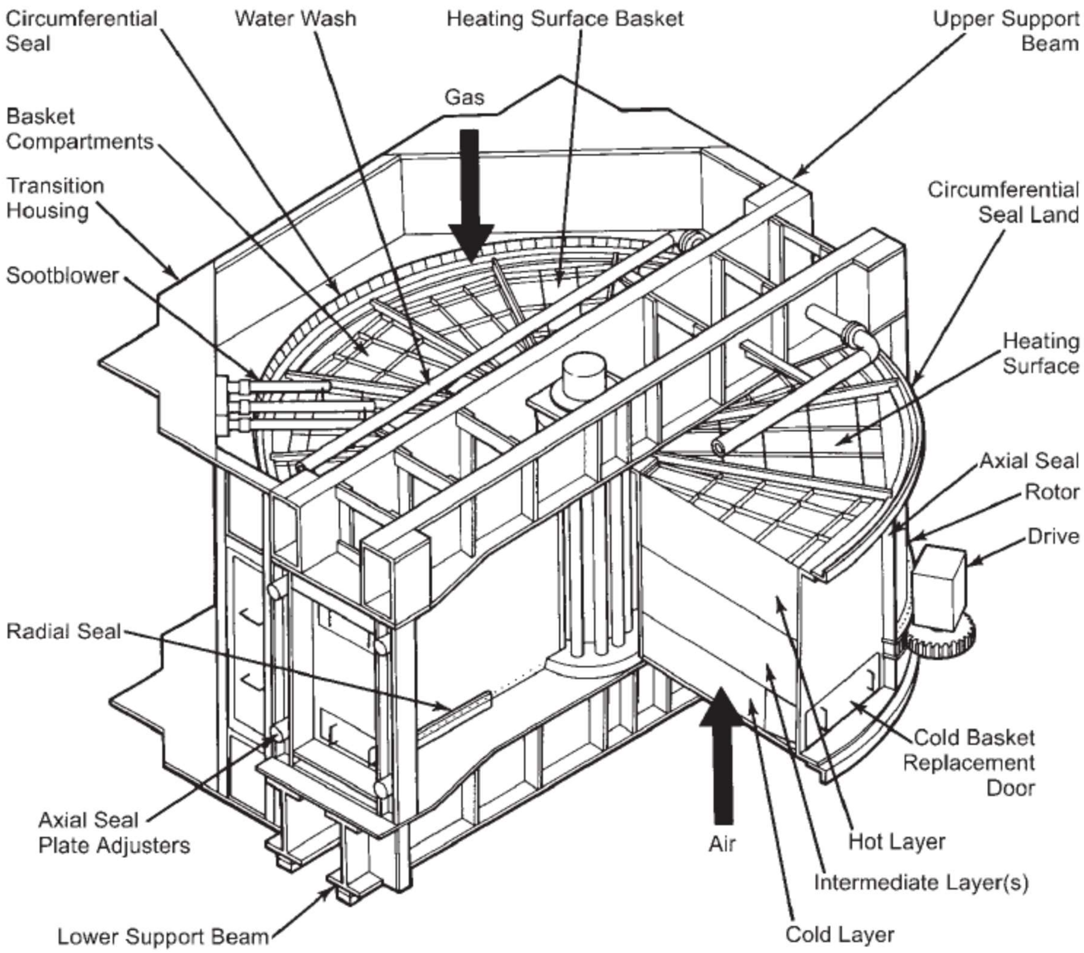

Figure 3 : Vertical shaft Ljungström air heater exploded view (Kitto,2005)

2.2.5 Rothemuhle air heater

The Rothemuhle air heater uses stationary element packs and rotating hoods (Fig 3). The stationary element packs are contained in a cylindrical shell called the stator. On both sides of the stator rotates a symmetrical double wing section called the hoods. These hoods rotate synchronously via a common vertical shaft. The shaft is supported by two bearings located in the stator bearing compartment and is slowly turned by means of a pin rack connected to the bottom hood via a pinion gear. Heat is transferred as flow streams through the heating surfaces in a cross-flow configuration, one flow stream inside the hood and the other on the outside of the hood. (Kitto, 2005)

Rothemuhle air heaters distort in the same manner as Ljungström air heaters. The air heater tends to cap towards the cold end in a saucer like manner. Special sealing systems are mounted on the interface between the rotating hoods and stator to reduce leakages.

Sealing between the rotating hoods and the stationary duct is maintained by a ring of spring backed cast iron wear shoes. (Kitto, 2005)

The rotor of the Rothemuhle air heater is very large and can vary between 8m to 13.5m in larger applications with a debt of up to 2.5m and a total weight of 1100tons. The flue gas normally enters the heater at 340°C and cooled down to 145°C. This temperature may vary in accordance with the type of fuel used in the boiler. (ASME, 1995:4)

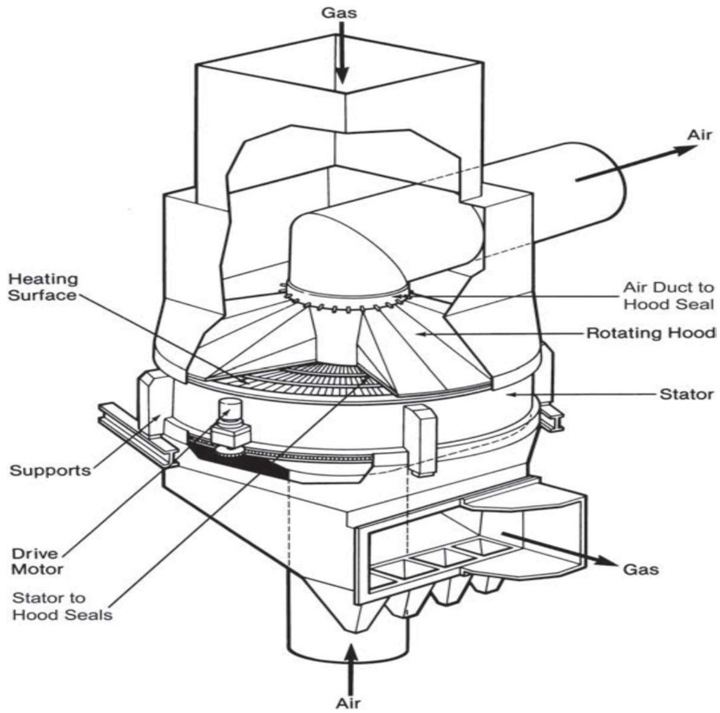

Figure 4 : Rothemuhle air heater (Kitto,2005)

2.2.6 Regenerative heating elements

Regenerative air heater heating elements are a compact arrangement of plates formed to a specific pattern. Each element pack consists of hundreds of flat, corrugated or undiluted plate profiles. The rolled form corrugations and undiluted patterns separate the plates to allow for flow through the pack as well as to increase the heat transfer area of each pack by creating flow turbulence. The steel plates, 3mm thick, are spaced 5 to 10mm apart. Closely spaced and highly profiled heating elements exhibit a very high heat transfer rate, high-pressure drop, and very high fouling potential while widely spaced heating elements where every other plate is flat, exhibits a much lower heat transfer rate, low-pressure drop, and reduced fouling potential. A combination of plate profile, material and thickness are selected for maximum heat transfer, low-pressure drop, cleaning ability, and high corrosion resistance. (Kitto, 2005)

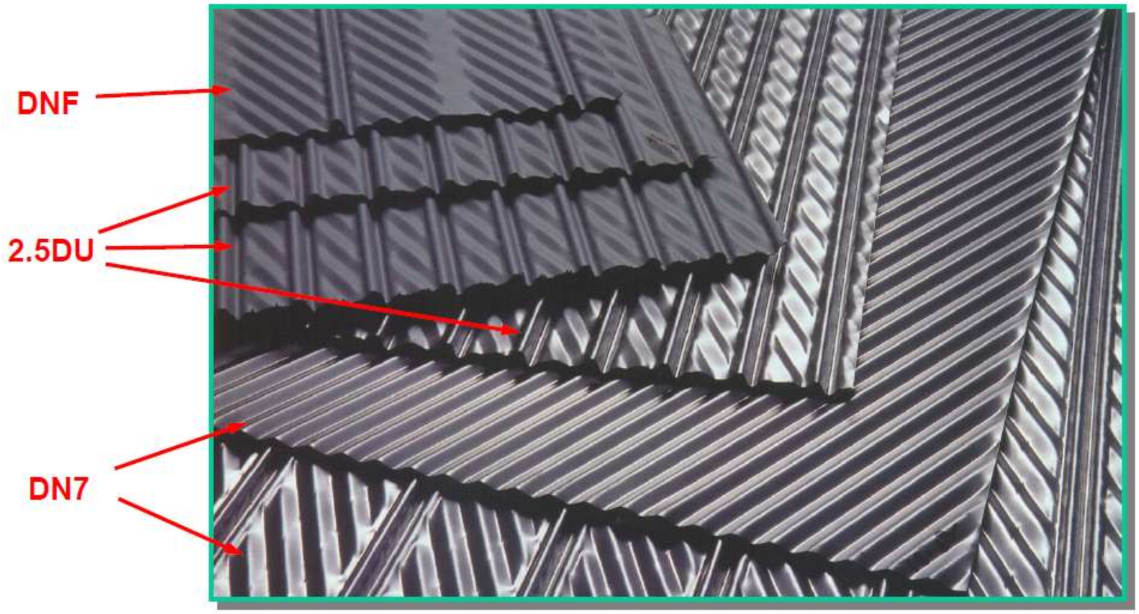

Figure 5 : Element pack profiles in use (Howden system engineers training)

The heating elements are stacked and either bundled in a self-containing basket or only by means of 2 flat bar straps to keep the plate pairs in place. The pack layers differ from the hot and cold end. The packs on the cold end side have a more open profile to increase the flow of air and to decrease blockages that might occur. The packs on the cold end are also smaller than on the hot end due to acid dew point that can destroy a cold end pack. The smaller size allows for packs to be changed during outages with minimal cost.

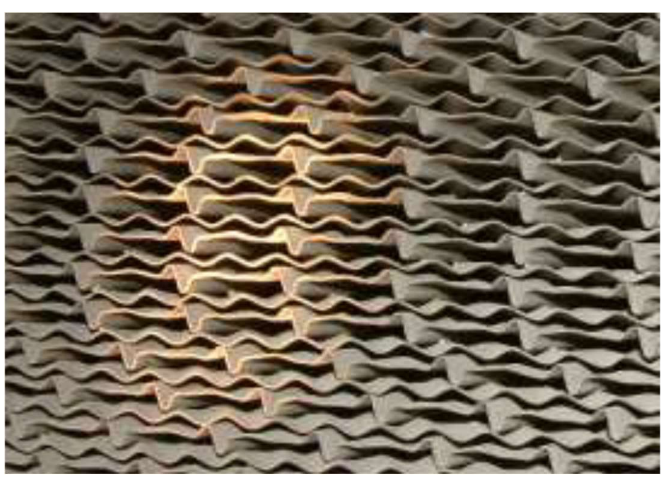

Figure 6 : Pack plate profile 2.5DU (Howden,2014)

2.2.7 Sealing arrangement

To reduce the amount of air leaking into the gas stream, high to low pressure and consequently, reduce the performance of the air heater each of the types of air heaters needs to implement a sealing system to reduce the leakage paths.

Rothemuhle air heaters make use of cast iron wear shoes that is mounted on a sealing frame that is in contact with the stator to seal the air and gas paths. Shoes are mounted on the sealing frame of the air heater hoods. The sealing frame makes use of spring pins to carry the load of the frame and shoes. The spring pins are also used to set the tension on the shoes when the air heater is on load to allow for a perfect contact seal. The cast iron wear shoes are 30mm thick, allowing it to bed in during operation and still be able to operate for up to 3 years.

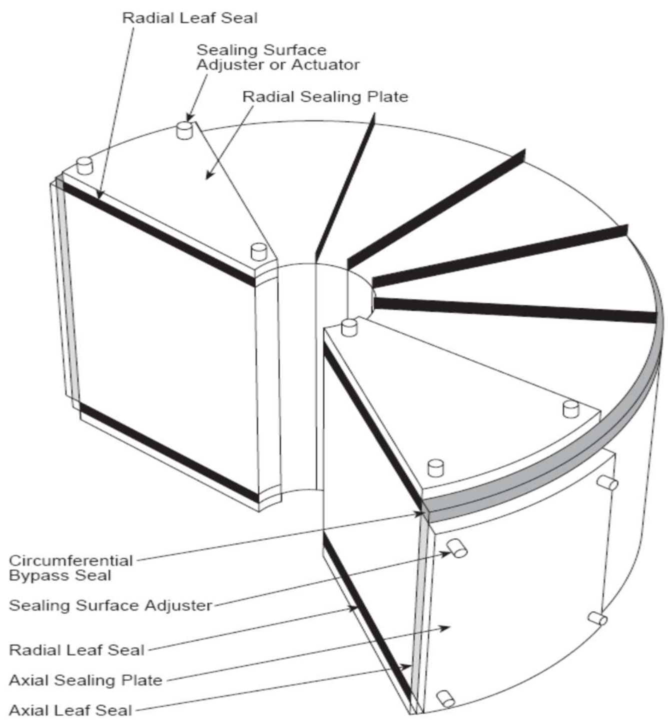

Figure 7 : Ljungström air heater sealing arrangement (Kitto,2005)

Figure 8 : Cast iron wear shoes installed (Howden,2014)

Ljungström air heaters make use of sealing strips to seal the rotating packs from the air/gas paths. These strips are manufactured from 3mm thick mild steel plate that is bolted onto the radial and axial plates of the rotor. The radial seals are responsible for preventing leakage from the flue gas side to the air side as the hoods are rotating. The circumferential seals are located at the edge of the air heater to prevent any bypass leakage around the air heater. On some air heaters, the sector plate can be adjusted during operation thus allowing for the sealing to be change during different boiler load.

Some companies have started to install either brush type seal or positive contact seal on the Ljungström air heaters.

Brush type seals consist of thousands of filaments that form part of a high-integrity seal and provides a very high degree of abrasion resistance and bend recovery under continues operation. (SEALEZE)

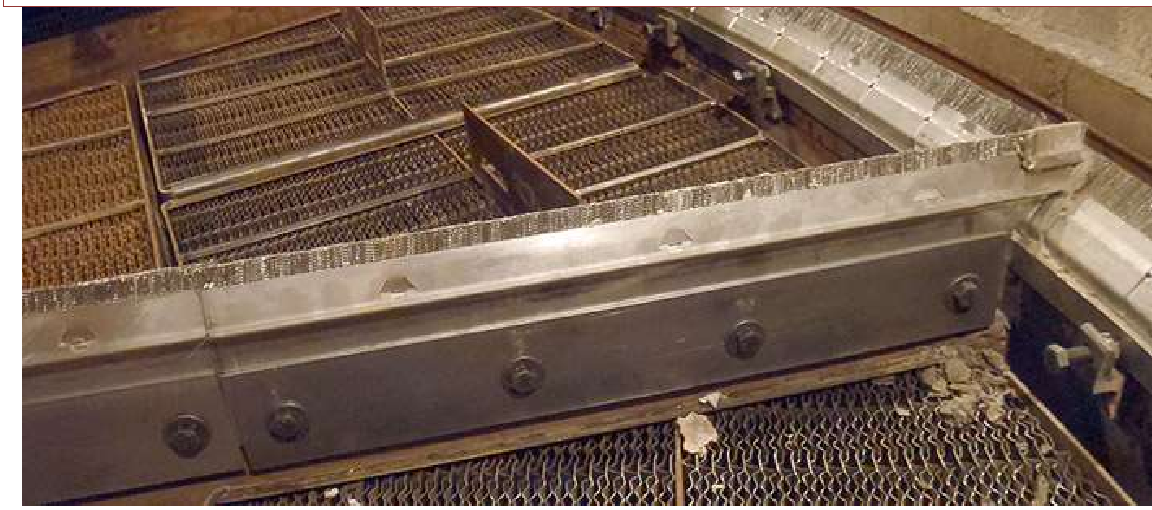

Figure 9 : Brush type seals for reducing leakage

Both brush type and positive contact seals have not been used for very long time and concrete evidence of the performance cannot be evaluated.

2.2.8 Leakages

Leakage can be described as the loss of combustion air into the flue gas stream at the air heater. Leakage is expressed as a percentage of the flue gas inlet flow. Leakage is undesirable because it is a representation of the fan power that was lost transporting air which did not partake in the combustion process and starve the fan power due to high volume flow. Leakage also reduces the overall thermal performance of the air heater. (Kitto, 2005)

Recuperative type heat exchangers start off with close to zero leakage, but with operation leakage can increase. With sufficient maintenance leakage can be kept below 3%. (Kitto, 2005)

For regenerative air heaters leakage can be described as either direct or entrained leakage. Direct leakage occurs when high-pressure air leaks into the low-pressure flue gas stream through gaps between the rotor and sector plate as the heater is rotating or gaps between the cast iron wear shoes and the stator face as the hoods are rotating. (Kitto, 2005)

Entrained leakage occurs as the packs or hoods are rotating and some air or gas is left inside the pack cavity. The leakage is directly proportional to the size of the cavity meaning larger air heaters have higher entrained leakage.

The design leakage for regenerative air heaters varies between 5% and 12% but can increase exponentially due to deterioration of the sealing system. Air heater leakage can either be calculated as the difference in the flow between the inlet and outlet of the air stream or the gas stream but it is difficult to get very accurate flow measurement due to the size of the ducting and the internal support structure, or a more accurate calculation is based on the oxygen content measured in the flue gas stream at the inlet and outlet of the air heater. (Kitto, 2005)

Skiepko and Shah (2005) presented a methodology for the evaluation of air heater leakage and to determine quantitatively what the influence on air heater performance and heat transfer are. The results show that it can drastically reduce the effectiveness of the air heater and that the correlations between leakage and effectiveness almost linearly is to the quantity of leakage.

Table 1 : Advantages and disadvantages of air heater types (Kitto, 2005)

Table 1
Advantages and Disadvantages of Air Heater Types

Type	Advantage	Disadvantage
Recuperative	Low leakage No moving parts	Large and heavy Difficult to replace surface
Regenerative	Compact Easy to replace surface	Leakage High maintenance Fire potential

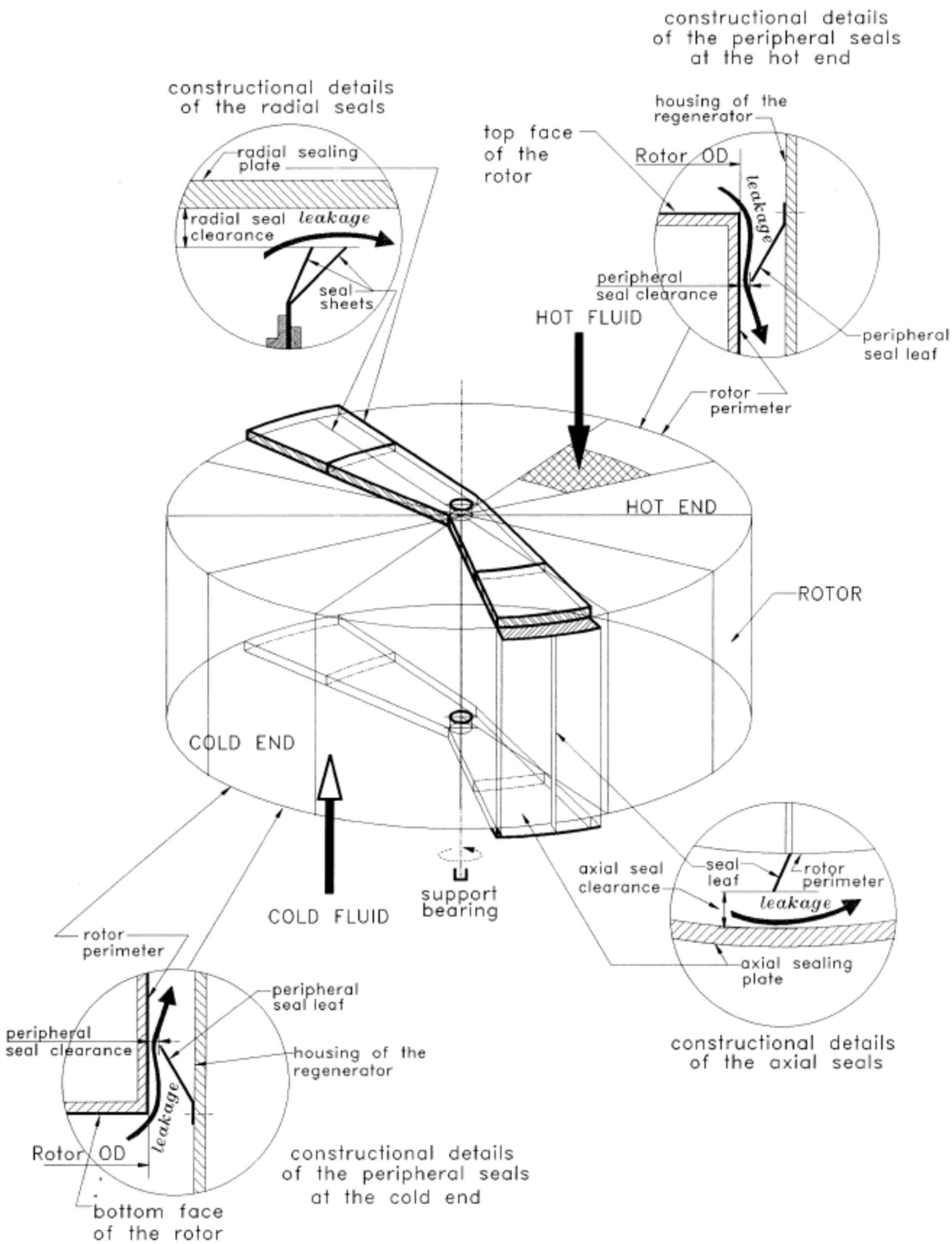

Figure 10 : Leakage path Ljungström air heater(Kitto, 2005)

2.2.9 Pressure drop

In regenerative air heaters, the main cause for pressure drop is due to frictional flow through created by the element packs. Typical values for pressure drop under original condition would be 1 kPa. The pressure drop is calculated as the static pressure difference between the hot and cold end, air side and gas side.

2.3 Operational conditions

Most air heaters within the Eskom fleet of power station experience the same maintenance and operational conditions. Corrosion, erosion, leakage fouling, and fires are all common operational condition that the air heater experience during the lifetime. Air heaters in high ash environments require more maintenance than plants in cleaner environments. Due to the high ash content of the coal at most of the stations, the corrosion rate on the pack are very high and the 10-year life expectancy cannot always be achieved.

One of the chemicals found in coal is sulphur and the air heater cold end side is most likely to experience the corrosion. Sulphur dioxide that is produced during combustion is converted to sulphur trioxide and when it is combined with water forms sulphuric acid. The vapour from the sulphuric acid condenses on the surface of the packs below the dew point temperature, below 120 °C. (Kitto, 2005)

Fouling occur when the pack surface is blocked by ash particles which are contained within the flue gas stream. It is most common at the cold end due to the acid build up on the packs. Fouling increases, the pressure drop over the air heater which leads to higher power consumption on the ID and FD fans.

Particles in the flue gas stream can also cause erosion damage to the air heater and its surrounding structure. The velocity of the flue gas stream is very high at the hot end side and most of the erosion damage also occur on this side of the air heater.

Air heater fires are a very rare occurrence but if not contain quickly can destroy a complete air heater within minutes. The fire is caused by fuel oil carry over during start up. As the fuel oil is left on the packs and the packs are heated causing the fuel oil to start burning. As soon as the pack material is on fire, the oxygen flow through the air heater needs to be cut off.

2.4 Common air leakage areas in the boiler plant

The most common areas of air leakage into the boiler are through duct leakage, boiler skin casing and sealing trough at the bottom of the boiler. The effect of leakage at different parts of the boiler can be classified as followed; combustion area, post-combustion area or back pass of the boiler and air heater. (Siddhartha, 2007)

Air ingress in the combustion area can have the following effects:

- Reduced heat transfer efficiency due to lower flue gas temperature in the combustion zone.
- Reduction in flame temperature resulting in a reduction in radiant heat transfer.
- Increase in volume of air passing through the boiler.

Air ingress post-combustion area:

- Higher velocities that can cause higher erosion rates in economizer and air heater inlet ducting.
- Increase demand on induced draught fans leading to increasing auxiliary power consumptions of the fan motor.
- Failure in operational logics due to false readings of flue gas oxygen content.

Air ingress through air heater:

- Less air participating in combustion due to bypass leakage.
- Dilution of flue gas reducing temperatures.
- Increase demand on FD and ID fans.

2.5 Current status of air heater assessment

In 1996, a collaborating study between Eskom and Wits was conducted and the intent of the study was to evaluate and improve the thermal performance of air heaters. To date, the focus of the study was the effects of fouling and erosion on the thermal performance. The reason for this was because all of the air heater used was designed in Europe where the ash content in the coal was much lower than in South Africa.

Caby (1996) made use of a single blow transient testing technique to estimate the different heat transfer coefficient for the different type of heating elements used within Eskom. The pressure drop for the air heater was also measured. The test done by Caby (1996) were carried out at the Eskom thermal test station at RT&D in Johannesburg. The results of the test were compared to results for Rothemuhle in Germany. The results for the heat transfer coefficient was within 6% of the German test data and the pressure drop within 3%.

The results from Caby (1996) show that even with the high ash burden of the south African coal that thermal performance is very close to the German test data.

2.5.1 Numerical evaluation

A basis of three-dimensional numerical modelling and fluid mechanics was used to base the theoretical study of air heater performance. The model for the study made use of flue gas through the air heater as well as the heat transfer to evaluate the temperature distribution in the air heater. The model does take into account the effect on leakage through the air heater on the flue gas side, using undiluted gas temperatures. (Drobnic et al 2006)

The weakness identified by the study was the unavoidable leakage from the high-pressure gas side to the low-pressure air side as well as the entrapped leakage during the rotation of the air heater hoods or packs. The model that was developed simulated the operational condition of the air heater, including the leakage at different seal setting conditions. The model indicated that with very high leakage rates the effectiveness of the air heater can decrease by as much as 10%. With the higher leakage rates, the boiler would require more air resulting in higher flow rates through the air heater and more flue gas that need to be removed from the boiler. Due to the higher demand for air the FD and ID fans need to consume more power, thus resulting in a typical consumption of up to 2% more power resulting in less output by the unit. The increase in absorbed power is a decrease in plant efficiency. With this, it is very important to keep seal setting as optimal as possible to reduce any unnecessary leakage. (Drobnic et al 2006)

One path of air and gas leakage that was identified in the study that plays a big role in the efficiency of the air heater was bypass leakage around the periphery of the air heater. This leakage plays no part in any heat transfer as it passes the heater without being in contact with any heating surface. (Drobnic et al 2006)

To verify these finding a CFD model was run for the radial seal setting. The mass flow rate of the leakage through the radial seals was determined by the pressure difference between air and gas streams. The results from the model indicate high concentrations on the edges of the rotating matrix, with higher velocities and lower temperatures due to leakage for the high-pressure air stream. (Drobnic et al 2006)

When the clearances on the seal setting were increased, more air leaked into the gas stream resulting in the decrease of flue gas temperatures. The model confirmed that an increase in leakage at the cold end would have a significant effect on the flue gas exit temperature. (Drobnic et al 2006)

Another numerical study that was conducted using the experimental correlation of the nusselt number and friction factor show that with leakage as small as 5% can have a reduction in the thermal efficiency of the air heater. (Drobnic et al 2006)

The rate at which leakage occur is dependent on the rotational matrix, seal setting and pressure drop for high-pressure air stream to low pressure gas stream. One numerical study found that regardless of the leakage distribution in the air heater the transfer of heat to the air stream stay constant. The leakage distribution only affects the pressure drop on the flue gas side. (Skiepko, T. & Shah, R.K. 1999)

The study presented a methodology to evaluate the leakage and to quantitatively determine how the leakage of the air heater influence the overall air heater performance. The study also indicated that leakage cause a significant drop in air heater effectiveness. The drop-in effectiveness shows a direct relationship to the direct leakage measured. (Skiepko, T. & Shah, R.K. 2005)

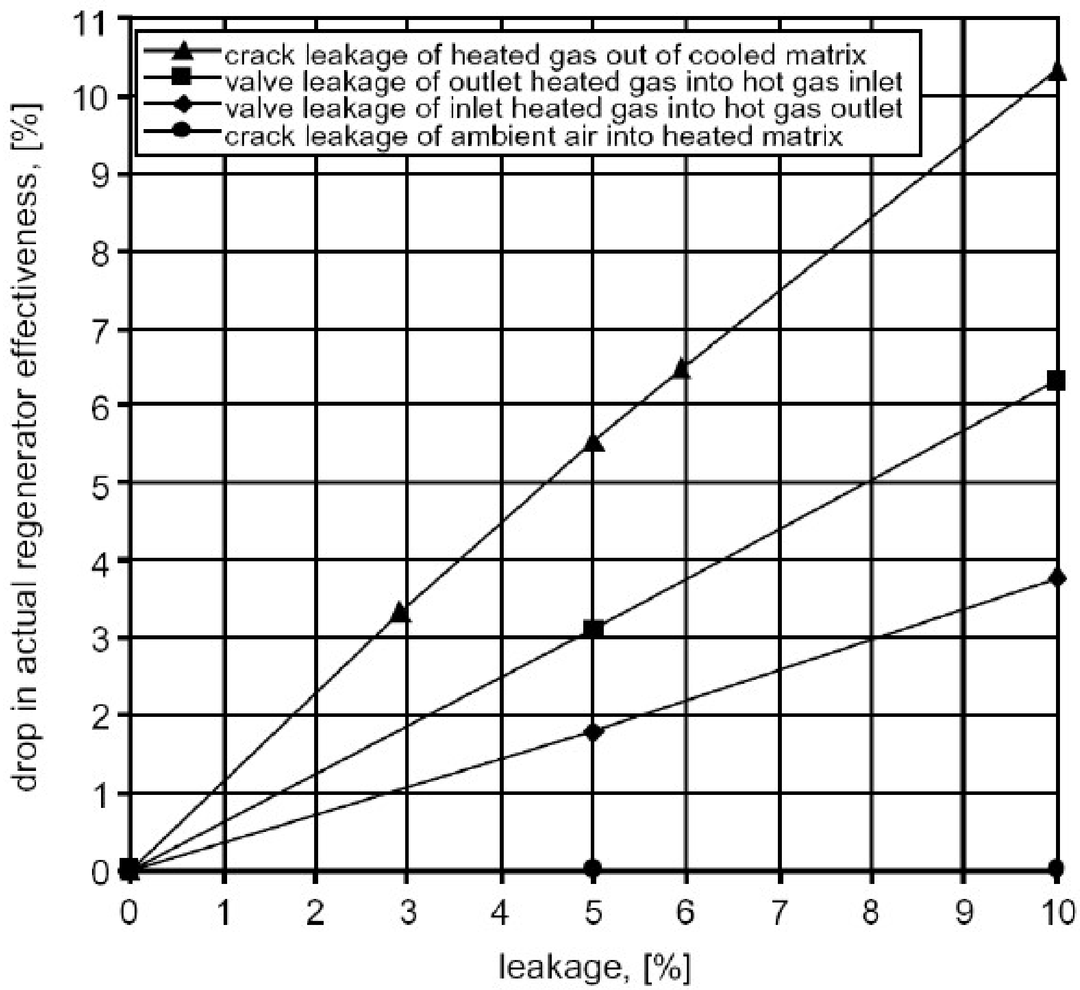

Figure 11 : The effect of leakage on air heater effectiveness (Skiepko, T. & Shah, R.K. 2005)

The study also found that for every 1% of leakage occurring on the cold end side the total drop in air heater effectiveness was 7%. (Skiepko, T. & Shah, R.K. 2005)

2.6 Online monitoring

Online monitoring systems were omitted from most of the older station in South Africa due to the demand on the grid in the early 1980's. Demand was low and the high leakage rate did not affect the energy output of the unit. Since the increase in demand, leakage has started to become the focal point for maintenance.

In order to establish an accurate indication of the performance of the air heater, knowledge of the leakage rate is very important. This continues monitoring of the air heater leakage could be incorporated into the DCS system.

To determine leakage without a DCS incorporation the leakage has to be measured by means of a performance test. Flue gas samples need to be taken before and after the air heater and then analysed by measuring the percentage of oxygen in the samples. This method is very time-consuming and special measuring equipment needs to be used. This method also has some problem that might occur. The condition of the flue gas changes as the load varies and the stratification of flue gas changes. If the mills in service experience any problems during the test, the result might be a misinterpretation of the leakage. There is also a person that take the measurements and errors from incorrect reading can influence any results.

All power plant uses O_2 probe at the exit of the economizer to the inlet of air heater. This is to measure the excess oxygen after combustion and to control the amount of air that the FD fans need to supply. If a second O_2 probe can be installed after the air heater on the flue gas side, the amount of leakage can be calculated by means of the following equation $L = \left(\frac{O_{2out} - O_{2in}}{20.9 - O_{2out}}\right) \times Factor$

This can only be used to estimate the leakage as it was said above that the stratification of flue gas in the ducting changes with load and to gather the correct information an infinite amount of O_2 probes would have to be installed in the ducting.

2.7 Conclusion

A through literature survey was conducted on the types of air heaters and their subsystems. As well as a discussion on the operational conditions. Numerical evaluation model was also discussed.

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3 Chapter 3: Plant/process description

3.1 Introduction

This study was based on Eskom Arnot Power station in Mpumalanga but the program can be used at any other power station in the Eskom fleet.

This chapter describes the plant/process of Arnot Power station. It also focuses on the limitations that were identified during the ACIP upgrades.

3.2 Design futures of Arnot Power Station

Arnot power station consists of six units. All units were retrofitted to increase the power output from the originally designed 350 MWe to 400 MWe per unit in the two-phase Arnot Capacity Increase Project (ACIP). In the first phase, the unit capacity was increased from 350 to 370 MWe. This was accomplished by the replacement of the generators, transformers, and their accessories. In phase II the unit capacity was increased from 370 to 400 MWe through general plant modifications in the turbine plant, boiler plant, control and instrumentation and outside plant.

Each boiler is fitted with two air heaters. Combustion air is supplied by two forced draught fans located on the basement level. The forced draught fans take suction at the top of the boiler house (184 feet level). The air is supplied to the boiler via the air heater where the air is heated to over 200°C.The primary air fans take suction from the air heater hot air discharge ducting and supply the air to the mills. The induced draught fans remove all the flue gas through the air heater and fabric filter plant.

3.3 Plant description

3.3.1 Milling plant

Boilers 2-6 are equipped with 6 vertical spindle mills of Loesche GmbH design and manufacture. Each mill is provided with a single primary air fan. These mills were upgraded during ACIP and their throughput was raised from 35 tons per hour to 41 tons per hours. However, sustainable maximum output remains at 38 tons per hour. The mill loads from the boiler test shown in table 4. For the performance test that was done after ACIP, 5 mills were in service with design coal being used. The current test that was done was also for 5 mills as there needs to be one mill on standby during operation.

Each mill has a PA fan that takes suction from a common duct after the air heater air side hot end. The PA fan blows air through the mill and up to the burners.

3.3.2 Burners

Each boiler is equipped with tilting tangential firing system which consists of 8 corner firing burner boxes. Four are located at the corners of the boiler and four are located on the furnace front and rear walls. The burners are directed tangentially on a circle in the middle of the boiler. The burner rows are equipped with a tilting mechanism to shift the fire vortex up and down within the boiler depending on pressure and load. The velocity at the burners is between 25 – 30 m/s for the primary air and 46 – 50 m/s for secondary air. For optimized combustion, the primary flow needs to be decreased to between 20 – 25 m/s to have a more stable load on the PA fans and more stable flow. The secondary air can thus be increased to 50 – 55 m/s.

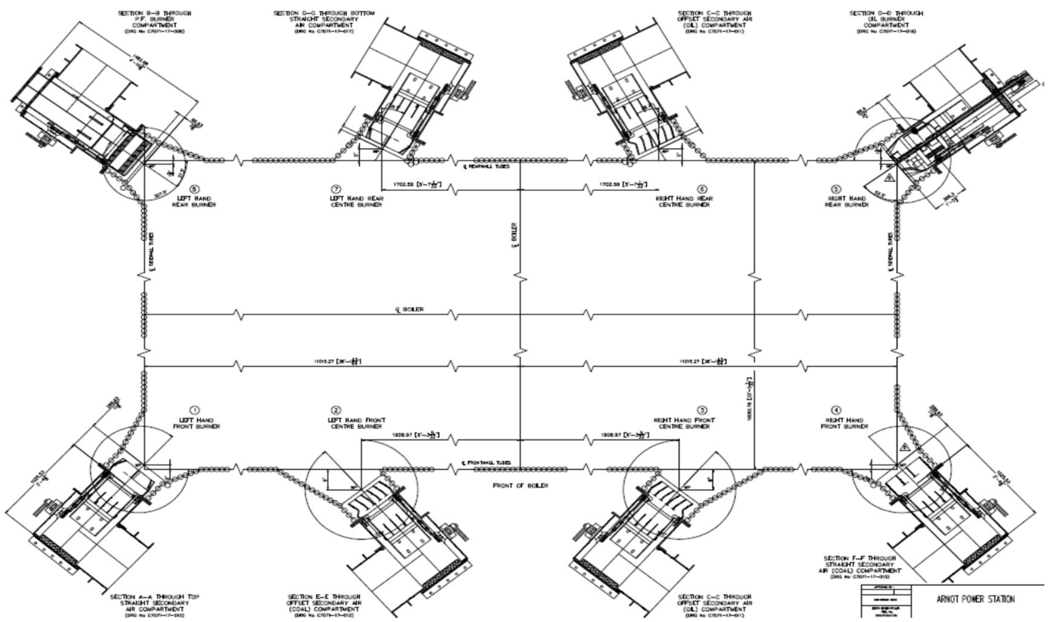

Figure 12 : Burner arrangement

3.3.3 Heating surfaces

The boilers at Arnot power station are drum type, dual pass natural circulating boilers with heating surfaces contained in three areas of the boiler:

1. The furnace or main combustion area has vertical rifle tubes forming the skin casing of the furnace.
2. Penthouse area containing the superheater and reheater tubes bundles
3. The convection path containing superheater and economiser bundle tubes.

The front enclosure of the furnace consists of the third to fifth stage superheaters as well as the third stage reheater. The rear enclosure of the rear gas path consists of stage one and two superheaters, stage one and two reheater as well as the economiser bank.

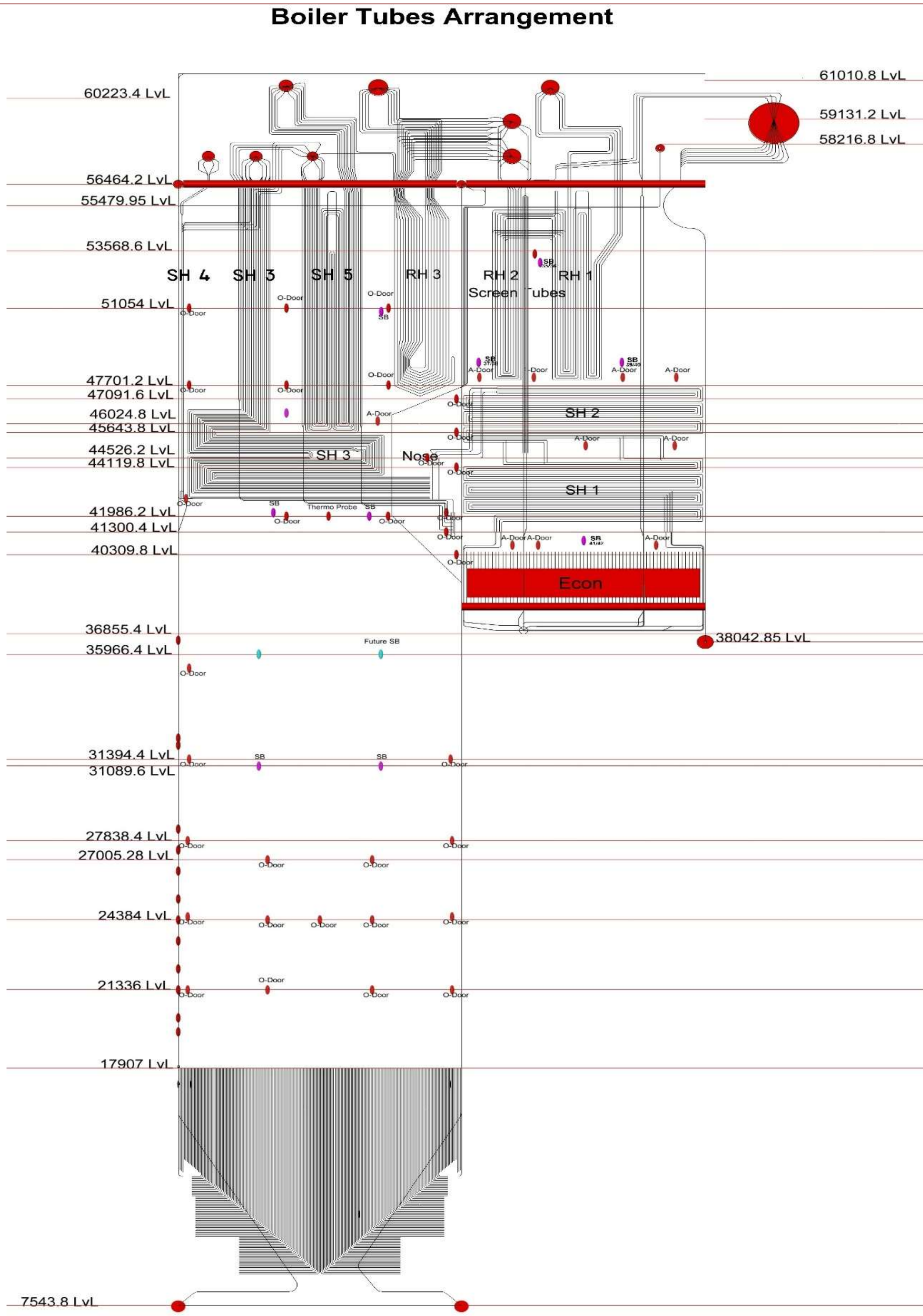

Figure 13 : Indication of heating surfaces in the boiler

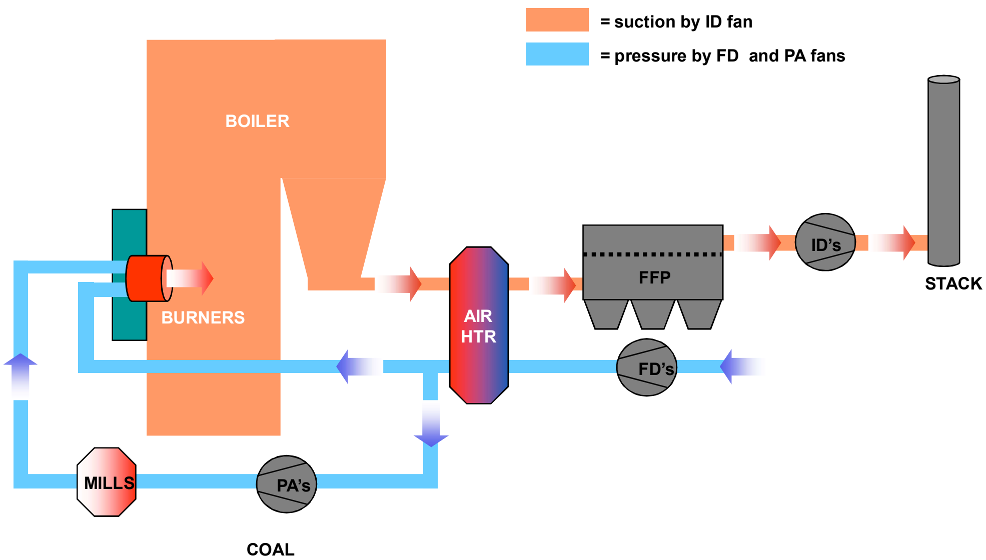

Figure 14 : Arnot air and gas layout

3.3.4 Arnot air heaters

Boiler 1 is fitted with a Rothemuhle type air heater and boiler 2-6 is fitted with horizontal shaft Ljungström type. All of the air heaters was manufactured and installed by Howden Power (Pty) Ltd during the construction of the station in 1960.

Flue gas exits the economizer and continues to the air heater gas side hot end. The flue gas passes through the element packs composed of multiple metal plates. The flue gas typically enters the air heater at 325°C at a mass flow rate of 210kg/s. The flue gas moves through the air heater element packs and raises the temperature of the metal plates. As the air heater rotates the hot plates move into the cold air stream, the cold air is heated up to temperatures of 255°C.

The heating surface elements are made from steel sheet which is formed in notches and undulations. The notches run parallel with the rotor axis and space the plates the correct distance apart. The undulations run at 30° to the notches and impart high turbulence to the gas and air passing through the preheater.

The element packs at Arnot is double undulating (DU) heat transfer plates. This element profile increases the flowing of air through the packs and promotes the absorption of heat. The 2.78DU profile at this depth offers an optimum combination of heat transfer efficiency, pressure drop and ability to maintain a clean flow path by soot blowing. Moving to a denser element profile could improve heat transfer but the risk of permanent fouling could escalate. A less dense profile will reduce pressure drop but thermal performance will be compromised. Newer pack profiles include the HC11 profile with even better heat transfer and lower pressure drops. The new HC11 packs are also much easier to clean due to the large spacing between the pack plate. (Howden Power, 2015)

Two sector plates span horizontally across the open ends of the rotor sealing off the air stream from the gas stream. The plates, which are located between the centre section of the air and gas duct connection and the rotor, consist of three parts, a centre section and two outer sections which are hinged to the centre section. The plates are adjusted to seal against the flexible radial seals attached to the rotor and against the inner and outer circumferential sealing surface on the rotor.

Primary seals, which utilise the pressure differential of the air and gas streams, are fitted along the gas side edge of the sector sealing plate to prevent leakage of air through the passage between the sector sealing plate and the centre section of the transition ducting.

3.4 Current plant limitations according to ACIP

3.4.1 Introduction

During the period from 2005 till 2011 Arnot power station has increased its capacity from 350MWe to 400MWe. The upgrades were done on the turbine and generator. The boiler plant and boiler auxiliaries were left as installed. The limitations of the ACIP final report will be discussed below.

3.4.2 ID fans

The original Arnot ID fans were designed to suit the existing boiler fitted with electrostatic precipitator gas cleaning system. Fan drive was via a hydraulic-controlled variable speed coupling. In the late 1990's the gas cleaning system on boiler 4-6 was upgraded to a fabric filter which introduced a significant increase in the gas path system pressure drop. However, unlike other FFP designs, there are no introductions of any attemperating air to cool the incoming gas stream. (Howden, 2009)

With the introduction of an FFP system, the increase in system resistance necessitated the installation of completely new ID fan rotating assemblies into the existing casings, including upgraded motor to cope with the increase in power requirements.

Before the commence of the upgrade of the ID fans a fan performance test was performed as the baseline to the upgrades. The duty point for 350MWe was plotted on the performance curve.

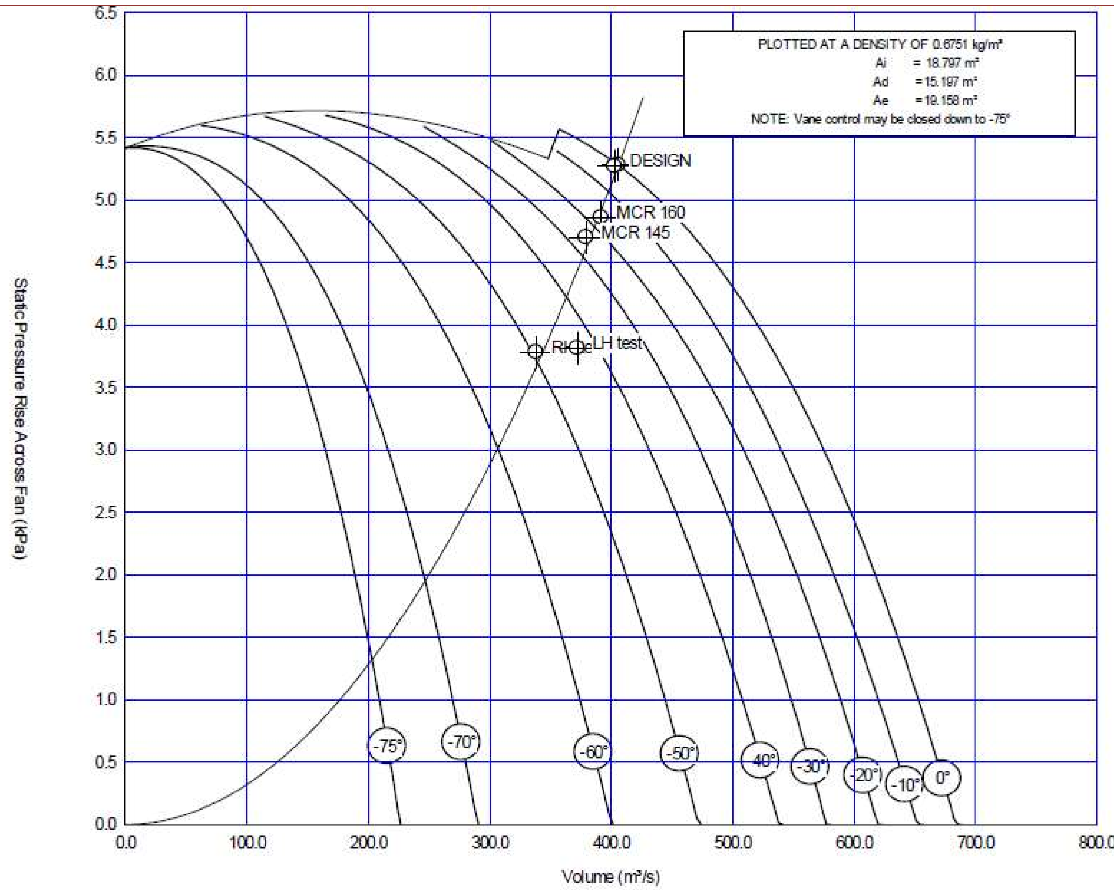

Figure 15 : ID fan performance curve 350MWe (Howden,2004)

The 350MWe tested fan pressures are significantly lower that the MCR pressure specified when the fans were installed. The tested flows are also somewhat reduced. Nevertheless, there is a comfortable margin in terms of both flow and pressure rise in the event of excursion in boiler gas conditions.

In the extrapolation of the tested 350MWe duties to the 400MWe conditions, the ID fan remains within the performance envelope within reasonable margins on both pressure and flow. However, if the FFP pressure drop were to increase towards the end of the bag life cycle and the air heater leakage rate as well as pressure drop were to increase, the duty point would be beyond the upper limit of the fan capacity. (Howden, 2009)

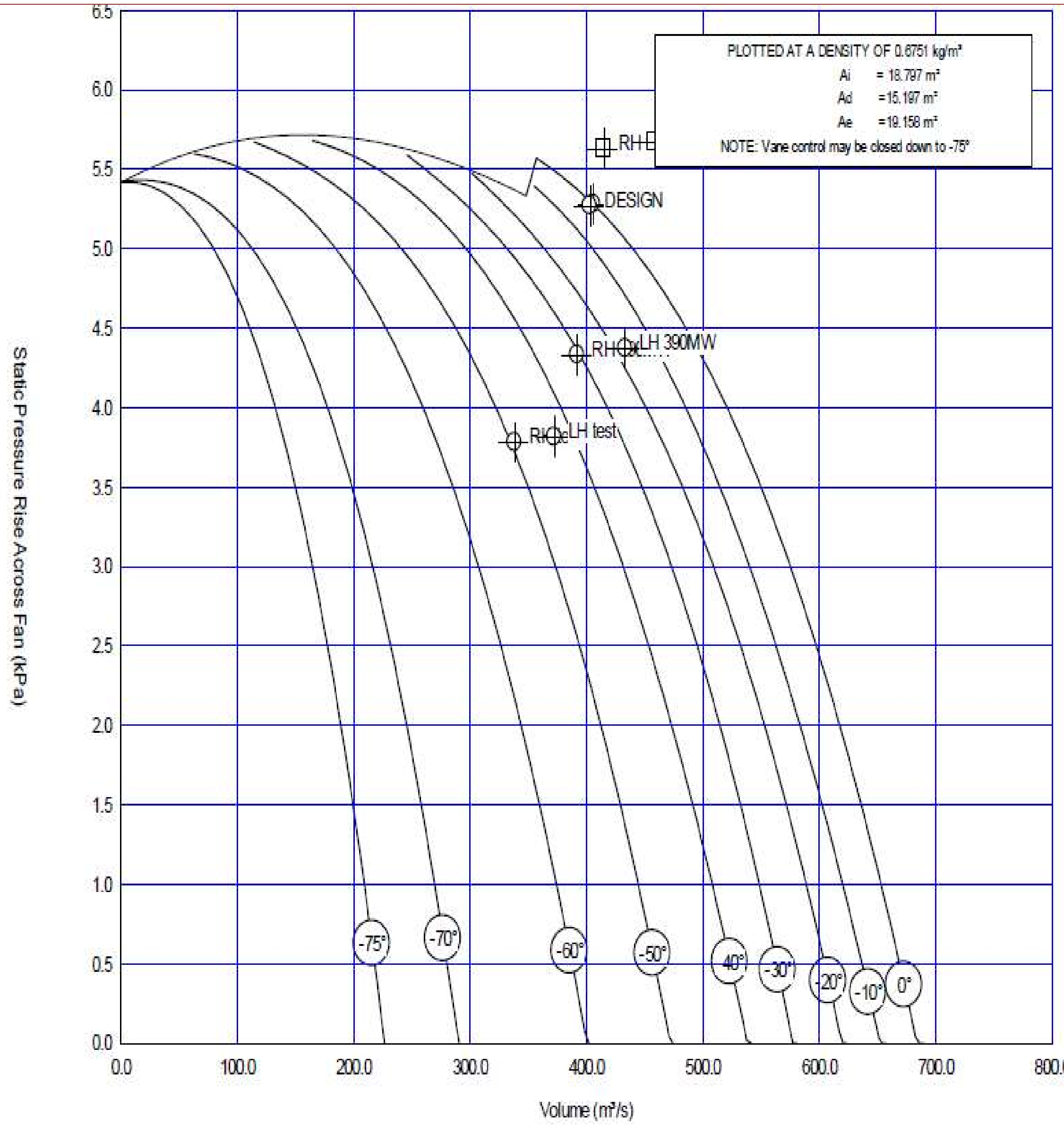

Figure 16 : ID fan performance curve 400MWe (Howden,2004)

3.4.3 FD fans

The existing FD fans are still in use and have not been upgraded. They have been replaced during the 1990's as they became fatigue life expected. The 400MWe duty point remains well within the fan capabilities. Even with high leakage and increase in windbox pressures the duty point will still be at acceptable points with margin to spare. (Howden, 2009)

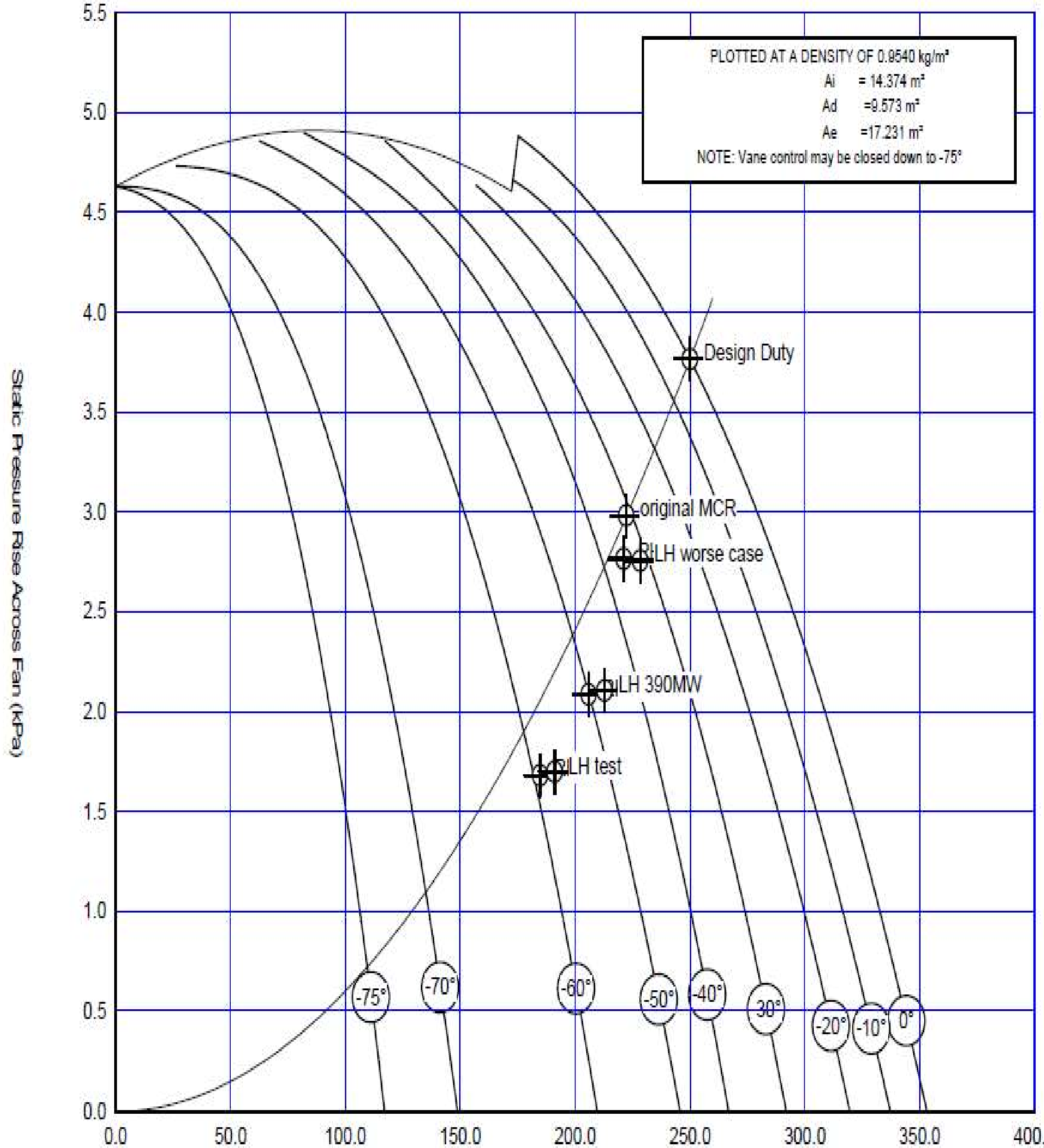

Figure 17 : FD fan performance curve (Howden,2004)

Table 2 : Air heater specifications (Howden, 2009)

Type	Rothemuhle	Ljungstrom (Howden HZW 29)
Boiler Unit	1	2 to 6
Number per boiler	2	2
Heater designation	10.6 dia	LJ29
Heating surface of heaters	38 647 m^2	41 338 m^2
Area of gas space	31,59 m^2	33,54 m^2
Area of air space		30,75 m^2
Weight of elements per heater	90 428 kg	128 820 kg
Element Material (Hot End)	Mild Steel	Mild Steel
Element Thickness mm	0.5	0.8
Element Profile	V75W	2.78DU
Element Depth mm	820	1016
Element Material (Cold End)	Mild Steel	Corten Steel
Element Thickness mm	0.8	0.8
Element Profile	V75W	2.78DU
Element Depth mm	250	356

Table 3 : Original design parameters (C-schedule)

Parameter	Original Design
Air heater gas inlet temperature	334 °C
Air heater gas outlet temperature	134 °C
Air heater air inlet temperature	38 °C
Air heater air outlet temperature	268 °C
Air heater gas inlet flow	420,8 kg/s
Air heater gas outlet flow	462,4 kg/s
Air heater air inlet flow	394,4 kg/s
Air heater air outlet flow	352,8 kg/s
Air leakage	10 % (41,6 kg/s)
Hot end differential	
Cold end differential	
ID fan flow	462,4 kg/s (538 kg/s
FD fan flow	394,4 kg/s (450 kg/s)
Air heater gas side pressure drop	810 Pa
Air heater air side pressure drop	660 Pa

3.5 Conclusion

This chapter discusses the plant layout at Arnot power station. It also discusses the limitations that were identified during the ACIP upgrades. The limiting factor on the equipment was identified as the ID fans. The 400MWe load case will only be sustainable providing that tramp air ingress, air heater leakage and pressure drop over the FFP and pressure drop over the air heater are maintained.

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4 Chapter 4: Modelling methodology

4.1 Introduction

The model was based on first principal thermodynamics and the derivative of the mass-energy balance formula for bi-sector regenerative air heaters. The chapter also consists of the coal analysis for the test as well as the combustion model before and after the air heater.

Firstly, the formulas that are used in the air heater performance model will be discussed starting with the energy balance and follow as stated below:

- Leakage
- Undiluted gas exit temperature
- Temperature differentials
- Effectiveness
- Pressure drop
- Pressure differential
- X-ratio

Secondly, the coal conversion is added to evaluate the condition of the coal burned after the ACIP upgrades.

4.2 Formulas for model

When performing the energy balance, we assume that the radiation losses are insignificant. For a bi-sector air heater the energy balance is:

$$\dot{M}_{g1} \times Cp_g \times \Delta T_g = \dot{M}_{a2} \times Cp_a \times \Delta T_a$$

It is convenient to measure the air inlet flow to the heater from test points of the FD fan outlet. It is then possible to calculate the other flows from the heat balance equation:

$$\dot{M}_{g1} = \frac{\dot{M}_{a1} \times \Delta T_a \times Cp_a}{(\Delta T_g \times Cp_g) + (\Delta T_a \times Cp_a \times L)}$$

$$\dot{M}_{a2} = \frac{\dot{M}_{g1} \times \Delta T_a \times Cp_g}{\Delta T_a \times Cp_a}$$

$$\dot{M}_{g2} = \dot{M}_{g1} \times (1 + L)$$

$\dot{M}$ Mass flow rate [kg/s]
T Temperature [°C]
C_p Specific heat [J/kg°C]

L Leakage [%]

The leakage expressed as a percentage of the gas inlet mass flow is calculated by:

$$L = \left(\frac{O_{2o} \quad - O_{2in}}{20.9 - O_{2out}}\right) \times Factor$$

O_2 Oxygen

$$Factor = \left(\frac{1.28701}{\rho_g} - 1.6011 \times k\right) \times 99$$

k % mass moisture in flue gas entering air heater

This factor covert from oxygen content in mass to oxygen content in volume as well as for dry gas composition factor. This is a requirement when sampling with a portable analyser on a dry basis.

Undiluted gas exit temperature is the temperature at which the gas would have left the heater if there were no leakage and is expressed as:

$$T_{g2NL} = L \times (T_{g2} - T_{a1}) + T_{g2}$$

T Temperature [°C]
L Leakage [%]

Temperature differentials are the difference in temperature between gas inlet and undiluted gas outlet as well as air inlet and air outlet:

$$\Delta T_g = T_{g1} - T_{g2NL}$$

$$\Delta T_a = T_{a2} - T_{a1}$$

Gas side effectiveness is the ratio of the gas temperature drop, to the difference between the air inlet and gas inlet temperature:

$$\eta_g = \frac{T_{g1} - T_{g2NL}}{T_{g1} - T_{a1}} \times 100$$

η_g Effectiveness [%]

Air side effectiveness is the ratio of the air temperature drop, to the difference between the air inlet and the gas inlet temperature:

$$\eta_a = \frac{T_{a1} - T_{a2}}{T_{g1} - T_{a1}} \times 100$$

η_a Effectiveness [%]

Pressure drop is the difference between inlet and outlet pressure of gas in/gas out and air in/air out:

$$\Delta \mathrm{P_g} = P_{sg1} - P_{sg}$$

$$\Delta \mathrm{P_a} = P_{sa1} - P_{sa2}$$

ΔP Pressure difference [kPa]

Hot end and cold end pressure differential are the difference in pressure on the hot end gas and air and on the cold end gas and air:

$$HEPD = P_{sa2} - P_{sg1}$$

$$CEPD = P_{sa} \ - P_{sg2}$$

4.3 *Thermal performance*

Air heater thermal performance is analysed by means of the undiluted gas exit temperature corrected for off-design conditions. The corrections cater for deviations in the air and gas inlet temperatures, X-ratio and gas inlet mass flow.

X-ratio is defined as the ratio of the heat capacity of the air passing through the air heater to the heat capacity of the gas passing through the air heater and is given as follow:

$$X_R = \frac{T_{g1} - T_{g2NL}}{T_{a2} - T_{a1}}$$

T_{g2NL} Undiluted gas exit temperature [°C]

T Temperature [°C]

4.4 Excel model

	Leakage %
	10
	Undiluted gas exit temp °C
	=(G33/100)*(L5-L16)+L5
	Temperature differential °C
Gas side	=D5-G36
Air side	=D16-L16
	Effectiveness %
Gas side	=(D5-G36)/(D5-L16)*100
Air side	=(D16-L16)/(D5-L16)*100
	Pressure drop kPa
Gas side	=D6-L6
Air side	=L17-D17
	Pressure differentials kPa
HEPD	=D17-D6
CEPD	=L17-L6
	Mean specific heat kJ/kg°C
Gas	=((D5*C37)-(G36*C37))/(D5-G36)
Air	=((L16*C38)-(D16*C38))/(L16-D16)
	Mass flows kg/s
Gas in	420,8
Air out	=((G59*G39*G55)/(G40*G56))
Gas out	=G59*(1+(G33/100))
	Log mean temp diff °C
	=((L5-L16)-(D5-D16))/LN((L5-L16)/(D5-D16))
	Heat transfer MW
	=(D18*(D16-L16)*G56)/1000
	X-ratio
	=(D5-G36)/(D16-L16)

Figure 18 : Excel model formulas

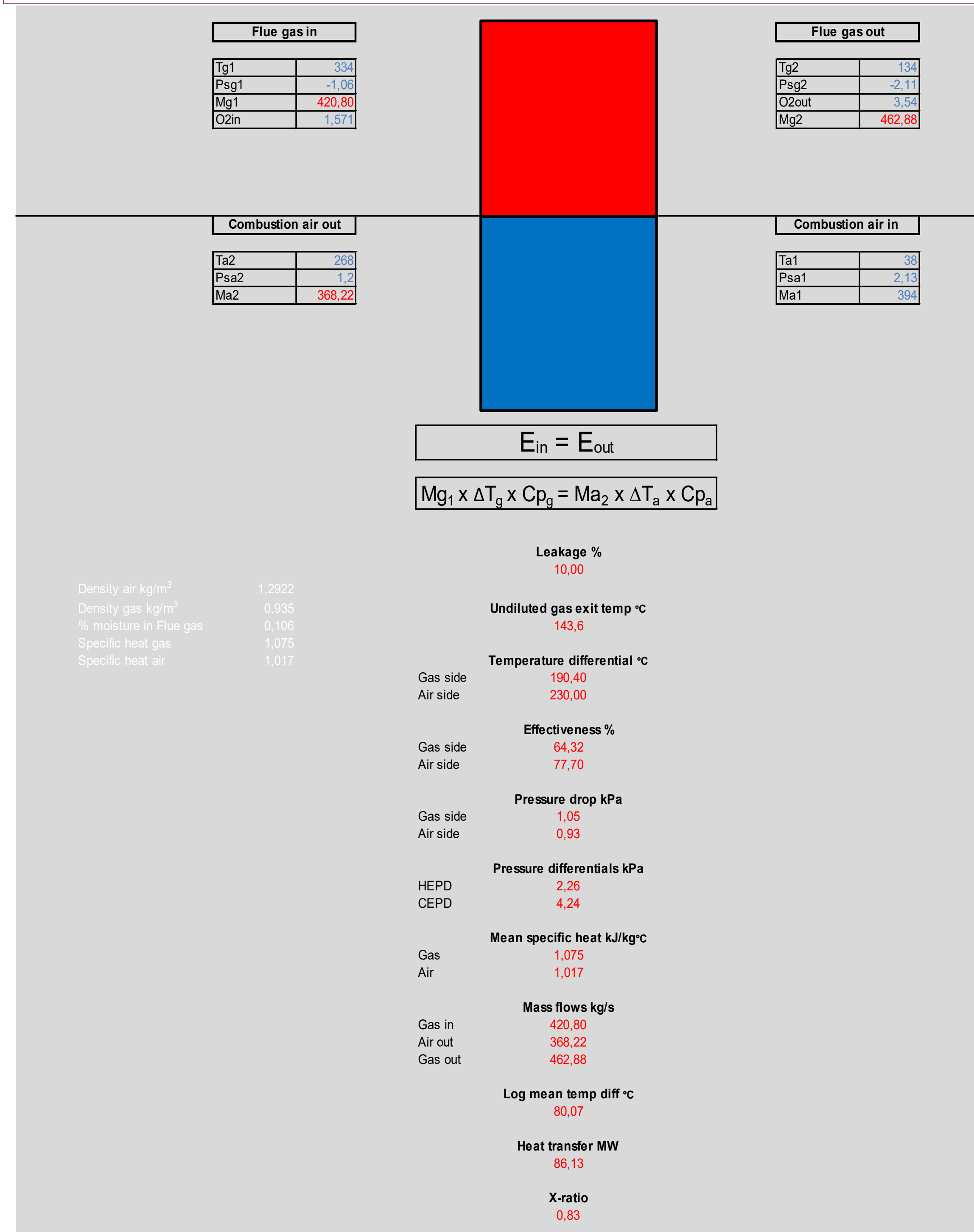

Figure 19 : Excel model

4.5 Coal quality

The coal analysis for this study was measured during the acceptance test for boiler 3 in 2011. The approximate analysis was obtained from the measurements. This was fed into the combustion model to convert from as received to the dry basis.

The coal analysis was used as an indication of the changes in coal quality from design to the current condition. The coal quality has a relatively large effect on fan performance as more or less air is required for different coal qualities.

Table 4 : Coal conversion

COAL ANALYSIS CONVERSION						
APPROXIMATE ANALYSIS						
CONVERSION BASED ON INITIAL M_s (AS RECIEVED), M_i (AIR DRIED), VOLATILES & ASH (DRY BASIS), C_{FIXED} BY DIFFERENCE (DRY BASIS):						
Sample type:			AS RECEIVED		AIR Dried	Dry Basis:
Conversion:			As Received from Air Dried	As Received from Dry Basis	Air Dried from Dry basis	Dry Basis: **Given**
Conversion formula:			$100/(100-M_{s\,AR})$	$100/(100 - M_{s\,AR} - Mi_{AD} / ((100-M_{s\,AR})*100))$	$100/(100-M_{i\,AD})$	
Conversion factor:			1,0449	1,081	1,0341	
GRAVIMETRIC %	**SYMBOL**	**UNITS**	Corrected	Corrected	Corrected	Corrected
Carbon $_{Fixed}$ (by difference)	C_{FIX}	%	49,955	49,955	52,200	**53,981**
Volatile matter	VM	%	19,619	19,619	20,500	**21,200**
Ash	Ash	%	22,968	22,968	24,000	**24,819**
Surface moisture	M_s	%	**4,300**	4,300	0,000	0,000
Inherent moisture	M_i	%	3,158	3,158	**3,300**	0,000
Total moisture	M_t	%	7,458	7,458	3,300	0,000
Gross calorific value	GCV_v	[MJ/kg]	21,284	21,284	22,240	**22,999**
Total			100,000	100,000	100,000	100,000

COAL ANALYSIS CONVERSION						
ULTIMATE ANALYSIS						
CONVERSION BASED ON INITIAL M_s (AS RECIEVED), M_i (AIR DRIED), N, O (by difference), C_{Tot}, ASH, S, H (DRY BASIS):						
Sample type:			AS RECEIVED		AIR Dried	Dry Basis:
Conversion:			As Received from Air Dried	As Received from Dry Basis	Air Dried from Dry basis	Dry Basis: **Given**
Conversion formula:			$100/(100-M_{s\,AR})$	$100/(100 - M_{s\,AR} - Mi_{AD} / ((100-M_{s\,AR})*100))$	$100/(100-M_{i\,AD})$	
Conversion factor:			1,0449	1,081	1,0341	
GRAVIMETRIC %	**SYMBOL**	**UNITS**	Corrected	Corrected	Corrected	Corrected
Nitrogen	N	%	1,286	1,286	1,344	**1,390**
Oxygen (by difference)	O	%	9,646	9,646	10,080	**10,423**
Carbon $_{Total}$	C_{Tot}	%	54,822	54,822	57,285	**59,240**
Ash	A	%	22,968	22,968	24,000	**24,819**
Sulphur	S	%	0,805	0,805	0,841	**0,870**
Hydrogen	H	%	3,015	3,015	3,150	**3,257**
Surface Moisture	M_s	%	**4,300**	4,300	0,000	0,000
Inherent Moisture	M_i	%	3,158	3,158	**3,300**	0,000
Total moisture	M_t	%	7,458	7,458	3,300	0,000
Total			100,000	100,000	100,000	100,000

4.6 Discussion

The chapter discussed the numerical formulas that was used to set up the air heater performance model. The numerical model calculates the following parameters that is critical to the overall assessment of the air heater performance.

- Leakage
- Undiluted gas exit temperature
- Temperature differentials
- Effectiveness
- Pressure drop
- Pressure differential
- X-ratio

The model will be incorporated into the combustion model of Prof. C.P Storm. From the combustion model the air heater performance test model can be populated. The model can be used to evaluate different operational and combustion condition. The optimal operational condition and air heater performance range can be derived.

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5 Chapter 5: Plant measurements

5.1 Introduction

This chapter describes the process and procedure that was followed in the measuring of the actual plant operational parameters. All information provided as per performance test procedure.

5.2 Traverse discussion

The air heater leakage can be calculated by either using the difference in mass flow between the inlet and outlet of the flue gas or the air stream. However, obtaining accurate velocity reading in large ducting's is very difficult, therefore, the leakage from mass flow calculations is inaccurate. A more accurate leakage calculation can be based on the measuring of oxygen content in the flue gas stream at the inlet and outlet of the air heater.

5.3 Location of test points in rectangular ducts

ASME PTC 4.3 was used to determine the location of the sampling point in the ducting. According to the standard, the duct should be divided into equal areas for measurement of pressure, flow, and temperature. The number and arrangement of the sampling points will be determined by the size of the ducting. A total of no less than 4 points should be present in the ducting.

The sampling points were already installed in the plant before the test. The effort was made to ensure that the measuring point would give an adequate representation of the stratification of flow in the ducting.

The position of the sampling points will be discussed below.

5.4 Test points

Inlet to air heater (economizer outlet): test ports are underneath the grating. These should be 4 in total across the duct. They are 1" in dimension. Missing ports must be installed and equally spaced as per existing.

The outlet from air heater: test ports are on the discharge duct immediately after the air heater. These should be 5 in total across the duct. They are about 2" in dimension.

Inlet to air heater (from FD fan): These are 5 test points and must be 2" socket and plugs as shown below.

Figure 20 : Test points air inlet

The outlet from air heater: A single sampling point to be installed of dimension ½" socket and plug. This can be on top of the air duct (hot end side) or anywhere suitable to can measure temperature and pressure. The sampling point must be before the expansion joint.

Inlet to FD fan: test ports are available (5 x off) but not suitable. These can be modified by reducing the length of the sampling point's pipe length such that the length is equivalent to the plug length.

FD discharge pressure: Install ½" socket and plug above fan discharge flange at the centre.

Figure 21 : FD fan suction test point

Inlet to FFP: Utilize existing rectangular ports. A special blanking plate (2 off) to be used during testing and moved around per point. See attached sketch for reference. The plate should be of the same geometry as the plate hidden inside the top plate.

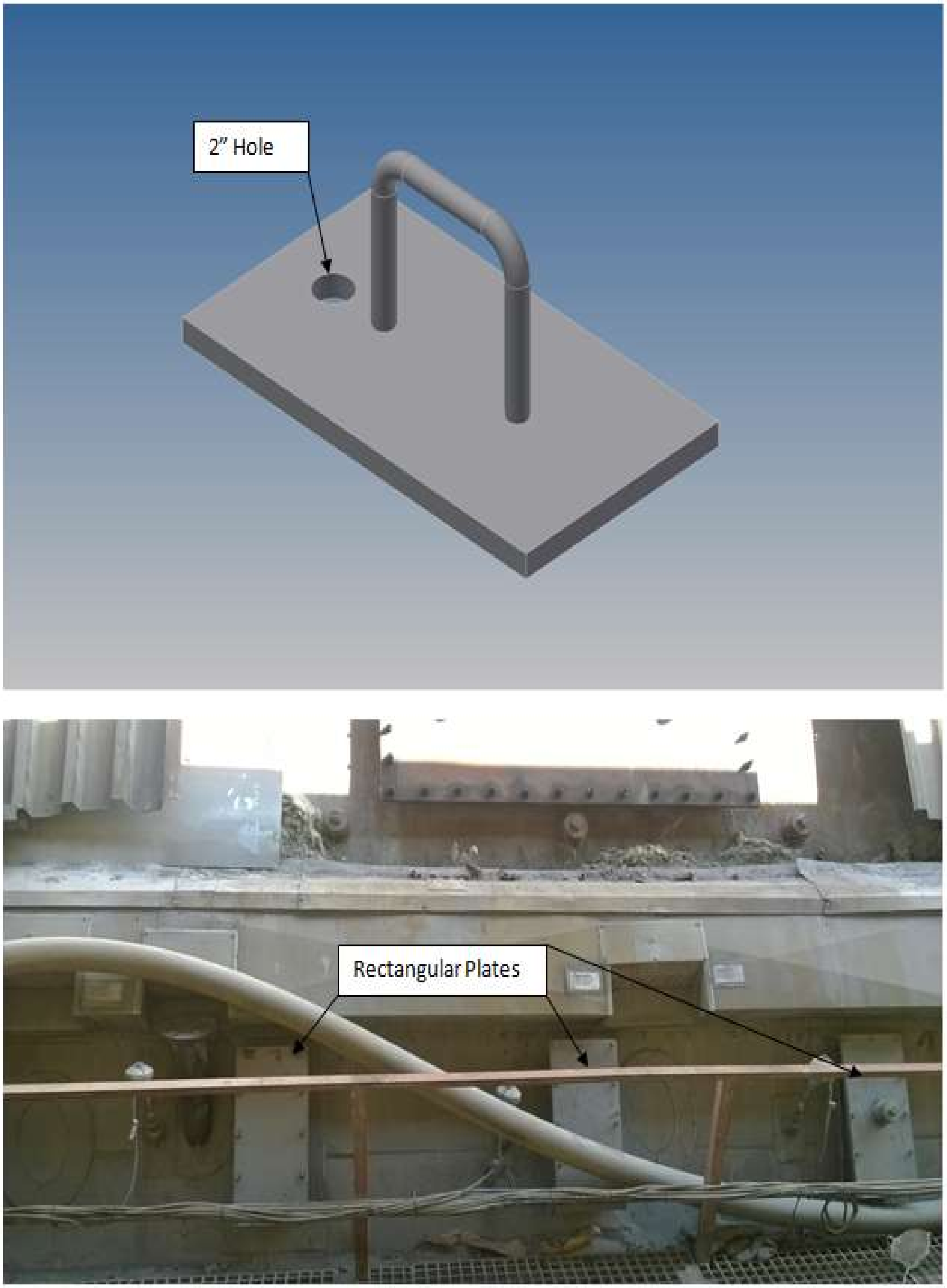

Figure 22 : FFP inlet test plate and points

FFP OUTLET: Use the same plate as above as the tapping points are similar to the FFP inlet geometry.

ID FAN OUTLET: A single point is required for pressure and temperature measurement. Install ½" socket and plug.

5.5 Test process

5.5.1 Pre-test checks

It is recommended that the air heater is physically inspected prior to the test to note the condition of all parts which may affect performance. The condition and cleanliness of elements should be examined and the air preheater placed in proper operating condition. Any external air bypasses or re-circulating dampers must be checked for sealing effectiveness, and any expansion joints between the test points should be checked for integrity as far as reasonably practical. All heating elements should be commercially clean (normal operating cleanliness) before testing can commence. All on-load soot blowing must be carried out prior to the commencement of the test, and no cleaning shall be permitted during the test until the test has been declared complete by Howden engineer. All de-ashing must be carried out prior to the test period because such an operation can produce large amounts of water vapour and can lead to air ingress. All sealing actuators must be checked for proper operation prior the test. Full load condition should be reached and stabilized at least 1 hour before the actual test. A sample of the fuel and flue gas should preferably be sent for laboratory analysis to determine the fuel and flue gas constituents. It is vital for the calculation of the leakage that the moisture content and density of the flue gas is known (a standard factor 0.89 is acceptable if the constituents are not known) (Falconer, 2009)

5.5.2 Testing & Duration

Air and gas flow through the air preheater should remain essentially constant, and the O_2 levels must be steady throughout the test. The steam generator output shall be set as close as possible to the design value and shall be held stable for at least an hour prior to the start of each test. Testing shall commence only when the parties to the test certify that the unit is operating to their satisfaction and is, therefore, ready for the test. Each test run shall be of minimum two hours' duration but sufficiently long to permit the taking of a complete set of consistent readings for a single air preheater. Should inconsistencies in the observed data be detected during a run or during the computations that would cause obviously untrue results, the run shall be noted as suspicious and might be rejected. A run that has been rejected shall be repeated to attain the objectives of the test.

To allow complete analysis of the test data, printouts of control panel recordings at regular intervals should be supplied by the Power Station detailing the following:
· Boiler load
· Fuel flow
· All recorded temperatures, mass flows, and pressures
· Fan amps
· Any other factors which may affect heater performance i.e. Control Damper positions, etc.
(Falconer, 2009)

5.5.3 Apparatus Instrumentation

Thermocouples
Calibrated NiCr: NiAl thermocouples (Type K) shall be used with a calibrated Digital Thermometer or with an Analogue to Digital mill volt converter connected to a data logger. The thermocouples shall comply with the following specifications:
Operating Temperature Range (minimum): 0 - 1000ºC
Accuracy: ± 1.5ºC

Manometers
Calibrated Digital Micro-manometers or calibrated 4–20 mA pressure transmitters connected to a data logger would normally be used.

Analysers
Gas analysers shall be used to determine the % concentration of Oxygen in the gas streams.
These can be of any type e.g.:
Paramagnetic Attraction
Catalytic Combustion
Electrochemical Membrane Diffusion
Zirconia Cell Diffusion

The calibration of this equipment shall be checked on site prior to and after each test to ensure consistency of readings and freedom from any significant deviations. A 100% N_2 gas bottle is used for the zero-calibration setting and a ±95% N_2 + ±5% O_2 gas mix is used for the span calibration setting.
(Falconer, 2009)

5.5.4 Test objective

Site testing on regenerative air heaters are conducted to determine the following operational characteristics:

- Air to gas side leakage
- Gas and air pressure drop
- Gas and air temperature
- Gas and air side effectiveness
- Thermal operating margin

5.5.5 Testing Procedure

Sampling points in the gas inlet and outlet ducts shall be located as close to the air heater as practical, and as consistent as possible with good velocity measurement practice. Dynamic pressure traverses will be taken simultaneously. Due to the probable stratification at the gas outlet caused by the rotation of the air heater the traverse section should be located as far from the rotor as possible but prior to any expansion joints which may allow air ingress to influence the measurements.

Flue gas temperatures must be recorded at the same locations and instants as those used for flue gas sampling, and so a thermocouple shall be fitted to the sampling probe. Flue gas O_2 measurements are required to calculate air preheater leakage. For very steady boiler load conditions with minimal time variations in the boiler outlet/air preheater inlet O_2 levels, oxygen measurement is carried out by comprehensive traverses over the duct using a single sampling probe. Static pressures shall be measured by connecting manometer to pressure tapping on the faces of the preheater transition ducts. Experience has shown that side-wall static tapping gives sufficiently accurate values of average static pressures compared to those obtained with traverses, however if no side-wall tapping is available, traverses should be employed using a probe.

Flue gas and air quantity shall be determined by calculation from either pitot tube traverse or heat balance. If the air inlet flow is measured, a check should be made for any air drawn off after the FD fans before the air heater.

The following parameters must be measured during the test:

- Gas inlet (Economiser outlet): $O_2, Temperature\ and\ pressure$
- Gas outlet (Turbulator): $O_2, Temperature\ and\ flow$
- FFP inlet: $O_2, Temperature, pressure\ and\ flow$
- FFP outlet: $O_2, Temperature, pressure\ and\ flow$
- Air inlet (Atmosphere to FD fan): $Temperature, pressure\ and\ flow$
- Secondary air: $Temperature, pressure\ and\ flow$

(Falconer, 2009)

5.5.6 Analysis of heater performance

The analysis of the air heater performance is confined to the following:

- Comparison of test thermal performance with design
- Comparison of test air leakage with design
- Comparison of test air and gas side pressure drop with design

Conditions which affect air heater performance and which should be given special considerations are:

- Quantity of air and gas passing through the air heater
- Temperature of air and gas entering the air heater
- Air heater leakage
- Recirculating and bypass of air to control cold end temperatures
- Tempering air and setting infiltration which did not pass through the air heater
- Fouling or corroded elements

From the oxygen readings recorded, the amount of air heater leakage is calculated and once corrected using the appropriate formula is compared with the design specification.

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6 Chapter 6: Model validation

6.1 Introduction

The primary objective is to validate that the performance model in excel is accurate and results credible. To validate this, the Howden UK air heater selection program is used. The variables such as temperature, flow and oxygen are used to verify the validity of the excel performance model.

6.2 Interpretation

The objective of this study is to develop an air heater performance model that can be used during on load operations and validated with test data.

Validation one is a simulation with design data from the C-schedule. The air heaters designed for a specific air heater performance and leakage.

From the results show the flow and temperature of the design C-schedule and the performance model are these same thus verifying according to the design data.

Validation two is a simulation using data from the performance test that was done in November 2015 on unit 3.

Validation of the excel model was done with measured data from the performance test. The data was fed into the air heater selection program and the results from the program can be seen in figure 24 below. The same test data was then fed into the excel performance model and the results can be seen in figure 25.

With the results from the two programs it can be seen that there is a small difference in the results, thus, the excel performance model is adequate for the purpose of evaluating on load performance of the air heater. The program was done with an error of 10% and would still give acceptable results to be used for online monitoring.

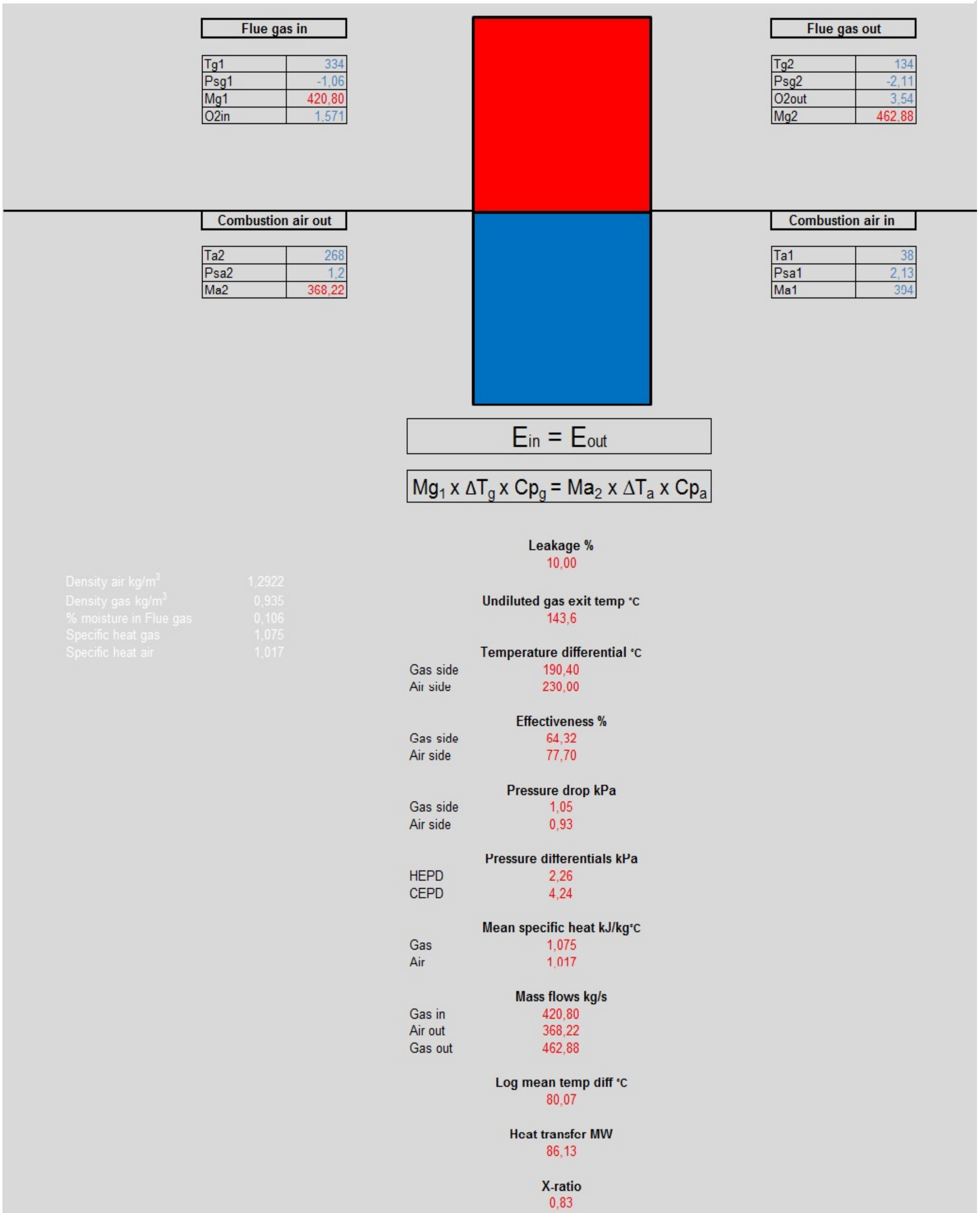


Figure 23 : Validation with design data

PRE HEATER PERFORMANCE
JOB NUMBER UNIT 3
CLIENT ARNOT P.S.
ENGINEER:
DATE:

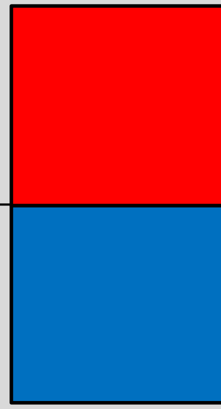
PRE HEATER PERFORMANCE CALCULATIONS

INPUTS: DESCRIPTION	INPUTS: VALUES	CALCULATIONS	
CALCULATION FOR MASS FLOW AIR ENTERING PREHEATER (\			
1. Barometric Pressure (kPa)	83,000	1. Density in Duct (kg/m3)	0,940
2. Duct Temperature Ta1 (Cel.)	42,500		
3. Duct Area (m2)	11,210	2. Volumetric Flow Rate (m3/s)	165,830
4. Static Pressure in Duct (kPa)	2,160		
5. Dynamic Pressure in Duct (Pa)	102,870	3. Mass Flow Rate Wa1 (kg/s)	155,907
6. BOILER LOAD %	88,250		
GAS SIDE		0. Undiluted Gas Exit Temp (Cel.)	147,035
1. Inlet temperature Tg1 (Cel.)	316,100	1. Percentage Leakage	10,037
2. Outlet temperature Tg2 (el.)	137,500		
3. Specific Heat form gas Cpg (kJ/kg)	1,075	2. Mass Gas Entering Wg1 (kg/s)	177,075
4. % Oxygen In	1,571		
5. % Oxygen Out	3,540	3. Mass Gas Leaving Wg2 (kg/s)	194,848
		4. Mass Air Leaving Wa2 (kg/s)	139,927
		5. Log Mean Temp. Difference	68,496
AIR SIDE		6. Overall Heat Coefficient	496,343
1. Inlet temperature Ta1 (Cel.)	42,500		
2. Outlet temperature Ta2 (Cel.)	268,650	7. Heat Transfer (MJ/Hr)	115,857
3. Mass Flow Rate Wa1 (kg/sec)	157,700		
4. Specfic Heat Cpa (kJ/kg)	1,017	8. Gas Side Effectiveness %	61,793
		9. Air Side Effectiveness %	82,657

Figure 24 : Validation with current selection model

AIR HEATER DIAGNOSTICS LH

Flue gas in	
Tg1	316,1
Psg1	-1,06
Mg1	177,08
O2in	1,571



Flue gas out	
Tg2	137,5
Psg2	-2,11
O2out	3,54
Mg2	194,85

Combustion air out	
Ta2	268,65
Psa2	1,2
Ma2	139,93

Combustion air in	
Ta1	42,5
Psa1	2,13
Ma1	157,7

$$E_{in} = E_{out}$$

$$Mg_1 \times \Delta T_g \times Cp_g = Ma_2 \times \Delta T_a \times Cp_a$$

Density air kg/m^3	1,2922
Density gas kg/m^3	0,935
% moisture in Flue gas	0,106
Specific heat gas	1,075
Specific heat air	1,017

Leakage %
10,04

Undiluted gas exit temp °C
147,0

Temperature differential °C

Gas side	169,07
Air side	226,15

Effectiveness %

Gas side	61,79
Air side	82,66

Pressure drop kPa

Gas side	1,05
Air side	0,93

Pressure differentials kPa

HEPD	2,26
CEPD	4,24

Mean specific heat kJ/kg°C

Gas	1,075
Air	1,017

Mass flows kg/s

Gas in	177,08
Air out	139,93
Gas out	194,85

Log mean temp diff °C
68,50

Heat transfer MW
32,18

X-ratio
0,75

X-ratio must be < 1.0
Gas side effectiveness < air side effectiveness
Air side effectiveness becomes the limiting factor for air heater thermal performance
The x-ratio is a function of the fuel burned and the combustion process
Lower grade and/or wet coal produces more gas per unit available energy
The x-ratio decreases and gas outlet temperature increases as a consequence
At reduced boiler load the x-ratio swings the opposite way
Lower grade and/or wet coal demand more primary air for drying and PF transport per MW output
More mills usually in service than necessary
The x-ratio increase and gas outlet temperature decreased

Figure 25 : Excel air heater performance model

7 Chapter 7: Assessment of results

7.1 Introduction

The focus of this chapter is to assess the performance of the air heater by using the excel model that was created. The cycle simulation was done firstly for the design data and secondly for the performance test data in order to compare the performance of current operational condition to that of the design condition. The results from the combustion model are also used to verify from a combustion perspective the performance of the air heater.

7.2 Summary of measurements

Table 5 : Measured vs DCS

Description parameter	Unit	Measured	DCS
FD fan inlet flow	kg/s	164,6	202
FD fan inlet temperature	°C	36,2	33,7
Air heater air inlet temperature	°C	36,2	35,6
Air heater air inlet pressure	kPa	2,13	2,14
Air heater air outlet temperature	°C	275,3	274,1
Air heater air outlet pressure	kPa	1,2	1,38
Air heater gas inlet temperature	°C	316	320,7
Air heater gas inlet pressure	kPa	-1,06	-1,02
Air heater gas outlet temperature	°C	132	151,2
Air heater gas outlet pressure	kPa	-2,11	-2,09
Economizer outlet oxygen	%	-	2,47
Avarage air heater inlet oxygen	%	1,57	1,56
Oxygen set point	%	-	2,42
Boiler load	MW	-	353

7.3 Measurement simulation

Results from the simulation of the design data for excel presented in table 6.

Table 6 : Design data simulation

Design data		Gas side	Air side
Temperature in °C		342	32.2
Temperature out °C		159	287.3
Mass flow in kg/s		443.1	353.19
Mass flow out kg/s		479.34	316.95
O2 in		3.06	
O2 out		4.57	
Pressure in		-1.06	1.93
Pressure out		-2.61	1.2
Leakage %	8.18		
Undiluted gas exit temp °C	169.4		
Temperature differential °C		172.63	255.10
Effectiveness %		55.72	82.34
Pressure drop		1.55	0.73
Pressure differentials kPa		2.26	4.54
Mean specific heat kJ/Kg°C		1.075	1.017
Log mean temp diff °C	85.76		
Heat transfer MW	82.23		
X-ratio	0.68		

Results from the simulation of the data as tested presented in table 7.

Table 7 : Test data simulation

Design data		Gas side	Air side
Temperature in °C		316	36.2
Temperature out °C		142	262
Mass flow in kg/s		221.2	195
Mass flow out kg/s		165.6	250.5
O2 in		1.57	
O2 out		4.09	
Pressure in		-1.06	2.13
Pressure out		-2.71	1.7
Leakage %	13.26		
Undiluted gas exit temp °C	156		
Temperature differential °C		159.97	225.8
Effectiveness %		57.17	80.70
Pressure drop		1.65	0.43
Pressure differentials kPa		2.76	4.84
Mean specific heat kJ/Kg°C		1.075	1.017
Log mean temp diff °C	77.02		
Heat transfer MW	38.04		
X-ratio	0.71		

7.4 Discussion performance test

7.4.1 Gas side

The position of all the retractable seals was not optimized before the test was conducted. Due to the incapability of the unit not been able to reach full load the seal position was accepted at 353MWe.

The oxygen set point for the unit measured at the economizer outlet was 2.42% according to the control room data received during the test. The average of the traverse measured on the inlet of the air heater was 1.6% which is much lower than the indicated set point of 2.42%. This is quite concerning as the low oxygen show that the boiler in operated in a lean condition. It has been observed that the traverses measured closest to the duct wall indicated a higher oxygen content, indicating air ingress.

Generally, the leakage should be below 9% as stated in the ACIP report. The measured air heater leakage is 13% which is well above the designed specification. However, if the air heater leakage is calculated using the oxygen set point of 2.42% discarding any ingress leakage measure on the duct wall the leakage become 8.5%. It can, therefore, be deduced that the air heaters performance based on leakage is within design specifications.

7.4.2 Air side

The measure total air flow of the FD fan and the indication in the control room do not correspond. This can be due to faulty measuring equipment in the plant. It was also noted that high leakage consists within the inlet ducting.

7.4.3 Pack condition

The pressure drop for both design and test condition show a drop of close to 1.6kPa. this pressure drop is acceptable as the condition of the air heater is poor. Blockage might be one of the reasons for the 1.65kPa pressure drop but a full internal inspection could give a different result.

7.4.4 Heat transfer

The MW heat transfer between the test data and the original data is indicating a big difference. The difference between the design and test can be contributed to the lower air heater inlet temperature as well as fouling and corrosion of the air heater packs. The condition of the air heater packs could not be evaluated before the test was conducted.

7.5 Conclusion

The results from the excel simulation model show a leakage of 13% for the test data compared to 8% for the design scenario. The main contributor to the higher air heater leakage is the retractable seals that was not optimized before the test was conducted and that the design scenario was for 400MWe and on the day of the testing the unit only achieved 353MWe.

There is also a high difference in the operational temperature of the flue gas entering the air heater. The design was for a temperature of 342°C but the measured temperature was only at 316°C. The difference in temperature can be contributed to the milling arrangement during the time of the test as well as the operational condition and the stability of the boiler at the time.

The overall thermal performance of the heater is much less than that of the original design with only a heat transfer of 38MW. From the following equation $\dot{m}\times Cp\times\Delta T$ the biggest difference is due to the much higher mass flow from the design data. This difference can be contributed to the modifications done in the boiler as well as leakages before the air heater. Corroded packs can also decrease the thermal performance of the air heater but no inspections on the

The pressure drop over the air heater is very close to design, indicating that the air heater is clean with minimal fouling on the packs, but can also be contributed to high erosion on the back as the packs have been in use for over 10 years.

One big concern from the performance assessment is the high mass flow rate of both flue gas and air. From the early performance test data, it was found that the coal condition has changed drastically during the last few years but the reduction in the air required decrease showing that the overall boiler performance is not optimal and the boiler is in fact operated in a lean condition.

8 Chapter 8: Conclusion and recommendations

8.1 Introduction

This chapter aims to conclude the dissertation as well as to give the review of the results obtained from the design model. It also gives the recommendation for the finding from the design model. From the recommendations below it is evident that a full pack replacement could help to improve the overall thermal performance of the air heater.

8.2 Conclusion

The mass energy balance for the air heater $\dot{M}_{g1} \times Cp_g \times \Delta T_g = \dot{M}_{a2} \times Cp_a \times \Delta T_a$ was derived for the assessment of the performance assessment. The formula was also used in the excel model and can be further used in the combustion model.

From the performance assessment, the following air heater problems were highlighted by the measurements:

1. Direct leakage cannot be measure. The air heaters leakage measurement includes the entrained and circumferential leakage as well.
2. Due to the location of measuring points, it is not possible to measure effectively the leakage over the air heater as there are expansion joints after the measuring point.
3. It is very difficult to conform to the equal sampling positions as specified by ASME.

The dominant leakage in the air heater was due to radial seal leakage, because of the high-pressure gradient between the gas and air streams.

The design data that was used in evaluating the current air heater performance clearly show that the current combustion process is not as originally designed for. The low temperatures entering the air heater can account for the reduction in overall air heater performance. The design gas inlet temperature of 342°C was never achieved during the combustion tests as the ID fans were out of capacity before high loads could be achieved.

The current excel model can be used in future evaluation before the scope of work is issued to address the necessary areas of maintenance.

The overall performance of boiler 3 is a clear indication of the current condition of plant and performance. The reduction in dry flue gas losses needs to be considered to increase the air heater performance as well as boiler efficiency.

8.3 Recommendations

A list of recommendations is given below with the discussion of the influences on performance.

8.3.1 Boiler combustion

From the measured data and the design data, it can be seen that the boiler outlet temperature is well below the design temperature. This can be due to mills in service or overall combustion in the boiler. A full combustion test need to be done to evaluate the combustion process as this have a very big influence on air heater performance.

For the boiler combustion test, not only the air and flue gas temperature and flow need to be considered but also the steam temperatures. The temperatures of the steam leaving the fifth stage super heater need to be evaluated with design parameters. As the fifth stage superheater is located in the penthouse it can give an indication of the temperatures entering the rear gas pass. Temperatures from the economizer outlet header is an indication of the temperature leaving the boiler and entering the air heater.

The combustion model added in the appendix can be used as a tool to evaluated the combustion process. The data supplied by the combustion model is based on the current conditions for the air heater performance test.

8.3.2 Leakages

With the high leakage measure, it would be necessary for a full boiler walk down to determine any ingress of air into the boiler and ducting. Excess air also contributes to air heater performance as it increases the amount of airflow but also influence the temperature of the flue gas entering the boiler. This in change can reduce the back-end temperatures which can lead to acid dew point temperatures.

If the seals were not optimized correctly before the test, the seal setting needs to be re-evaluated. From the test, it was found that the seals engagement only occurs at 380MWe. It shows that the seals are load dependent and not temperature dependent. This setting from the DCS needs to be changed. A test of the temperature entering the air heater needs to be conducted to establish the temperature at which full capping of the air heater has occurred. This temperature can then be used for optimizing the seal setting.

The optimized seal setting below was calculated at a metal temperature 310°C. This temperature was reached at 327MWe which is well below the intended 380MWe load at which the seals need to engage.

If the seals can be set to engage at this temperature the air heater leakage can be drastically reduced for the lower load as the unit was struggling to reached the 380MWe full seal engagement load.

Table 8 : Optimized seal setting

Rotor thermal movements			
T	2657,5	mm	Tyre radius
R_{TD}	1880	mm	Rotor depth
R_{PD}	1816	mm	Rotor plate
r_a	2657,5	mm	radius to the adjuster
r_{baffle}	581	mm	radius to baffle
e_{hr}	10,0		hot end radially
e_{cr}	2,2		cold end radially
e_a	4,2		axial expansion
Y_r	5,2		capping at distance r
Y_r	0,3		capping at the baffle
Y_r	5,2		capping at the outer radius

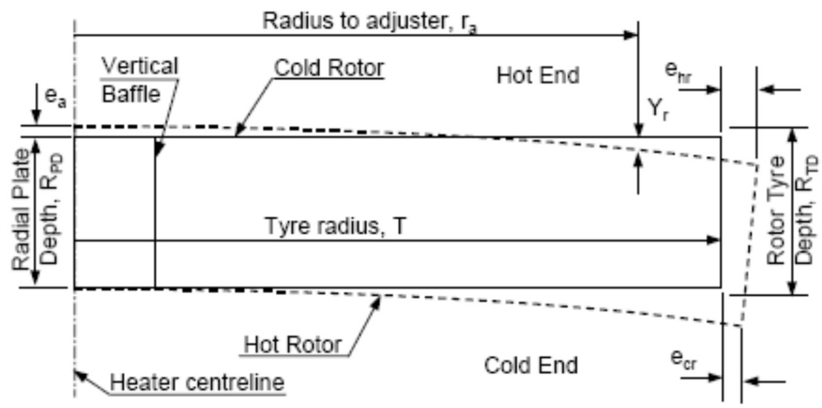


Figure 26 : Air heater capping

8.3.3 Pack replacement

As previously stated in the dissertation the condition of the pack is questionable. The packs have been in use for nearly 10 years and that together with fouling reduce the thermal performance of the air heater. Based upon the new design packs that are currently being introduced at various other power stations and based upon the process conditions given in the 'design coal' data sheet, predicted thermal performance for new replacement element packs will be as detailed in the table below.

The data specified in the table are predictions made on the current plant conditions. For these predictions to have relevance the boiler performance need to be evaluated as well as the combustion process optimization.

Table 9 : Thermal performance after pack change

(flow per blr)		TARGET 400MW	PREDICTED 100% MCR	PREDICTED 95% MCR	PREDICTED 75% MCR
GAS INLET TEMP	degC		331	327	309
GAS INLET FLOW	kg/s		422.8	399.7	331.0
AIR INLET TEMP	degC	25	25	25	25
AIR OUTLET FLOW	kg/s		360.5	339.0	275.0
AIR LEAKAGE	%	6.50	6.50 design (9.00 max)		
GAS OUTLET TEMP (undiluted)	degC		136.0	133.9	125.6
GAS OUTLET TEMP (diluted)	degC	135 max	129.2 (126.8)	126.9	118.0
GAS OUTLET FLOW (diluted)	kg/s		450.3 (460.9)	427.1	358.43
AIR INLET FLOW	kg/s		388.0 (398.6)	366.4	302.3
AIR OUTLET TEMP	degC	268	264	263	255
GAS SIDE PRES DROP	kPa	0.800 max	1.048	0.944	0.66
AIR SIDE PRES DROP	kPa	0.800 max	0.768	0.690	0.477

Discussion

- The output parameters are based upon 6.5% direct air-to-gas leakage. Whilst every effort will be made to achieve the design value, actual leakage may range up to 9%. Data presented in brackets

- Part load leakage values cannot be quantified since air-to-gas ΔP driving the leakage will vary non-linearly throughout the load range.

- Rotor speed is 1.277rpm per site information

- The specified maximum gas side pressure drop of 0.960kPa is unrealistic for an air heater of this size at the gas flow increase caused by moving from 350 to 400MW (ΔP escalates per a ‘square law’ with flow). Actual value estimated to be 1.048kPa

- With an air inlet temperature of 25°C, the expected gas outlet temperature of 264°C would not be possible. However, if the air inlet temperature were to be per the site test value of

32°C, the air outlet temperature would increase from 264°C up to 266°C. A corresponding diluted gas outlet temperature increase from 129°C to 134°C would also occur.

- Predicted air and gas outlet temperatures shown above apply exclusively to the specified input parameters (air and gas inlet temperature and mass flow, gas analysis etc.) at 100% MCR boiler load. If the input parameters change, there will be corresponding changes in output. These changes need to be considered in the overall air heater performance.

- Air-to-gas leakage is based on 0_2 pick-ups as measured volumetrically between the air heater gas inlet and outlet. This is regarded internationally as an acceptable method.

8.4 Further studies

- Evaluate the effect on trap air ingress through the boiler that is not taken into account for combustion.
- Evaluate the attemperating air for the milling plant that was not taken into account for this study.
- Evaluate the results of the performance test with a full boiler combustion test and optimize air heater model accordingly.

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10 Chapter 10: Appendix

10.1 Performance test

10.1.1 Raw data

Table 10 : Air heater inlet data

Air heater inlet				
Sampling point		O_2	Temperature	Pressure
A	1	2,82	299	1,06
	2	2,72	306	1,04
	3	2,07	309	1,09
	4	2,98	314	1,08
	5	2,79	319	1,04
B	1	1,93	304	0,99
	2	1,7	312	1,06
	3	1,11	320	1,10
	4	0,99	325	1,10
	5	1,02	328	1,05
C	1	1,58	308	0,97
	2	1,24	314	1,02
	3	1,09	319	1,09
	4	0,98	324	1,08
	5	0,88	326	1,09
D	1	1,52	308	1,00
	2	1,17	314	0,98
	3	1,05	318	1,03
	4	1	324	1,07
	5	0,98	327	1,21
E	1			
	2			
	3			
	4			
	5			

Table 11 : Air heater outlet data

Air heater outlet				
Sampling point		O_2	Temperature	Pressure
A	1	1,9	140	2,15
	2	1,97	143	2,14
	3	1,61	145	1,79
	4	1,5	145	2,16
	5			
B	1	2,79	140	2,16
	2	1,54	144	2,14
	3	1,01	150	2,14
	4			
	5			
C	1	2,84	121	2,10
	2	3,9	124	2,12
	3	2,35	133	2,15
	4			
	5			
D	1	6,67	111	2,13
	2	6,49	112	2,13
	3	6,29	112	2,12
	4			
	5			
E	1			
	2			
	3			
	4			
	5			

Table 12 : FFP inlet data

FFP inlet				
Sampling point		O_2	Temperature	Pressure
A	1	5,2	126	2,99
	2	5,13	131	3,01
	3	5,09	130	2,94
	4	4,9	130	2,96
	5	5,21	130	3,08
B	1	3,95	119	3,08
	2	3,86	139	3,01
	3	3,6	139	3,02
	4	3,58	142	3,03
	5	5,4	143	3,00
C	1	4,55	141	3,06
	2	4,43	143	3,05
	3	4,09	143	3,07
	4	4	144	2,98
	5	3,53	146	3,10
D	1	3,37	138	3,09
	2	3,34	143	3,04
	3	3,8	143	3,02
	4	3,57	146	3,07
	5	3,41	146	3,05
E	1	3,55	144	3,07
	2	3,4	146	3,01
	3	3,35	146	3,00
	4	3,39	147	2,94
	5	3,93	139	2,95

Table 13 : FFP outlet data

FFP outlet				
Sampling point		O_2	Temperature	Pressure
A	1	6,76	107	5,07
	2	6,4	126	5,10
	3	6,11	129	5,10
	4	6,24	130	5,12
	5	6,13	129	5,00
B	1	7,22	95	5,00
	2	6,96	117	5,13
	3	6,97	118	4,96
	4	7,03	124	5,01
	5	6,92	131	5,07
C	1	7,12	96	5,02
	2	7,41	109	5,01
	3	8,55	108	5,11
	4	8,41	107	5,16
	5	8,34	107	5,08
D	1	9,47	95	5,07
	2	9,38	111	5,14
	3	9,29	120	5,09
	4	5,3	128	5,03
	5	5,25	132	5,25
E	1	9,98	95	5,18
	2	8,82	124	5,00
	3	5,91	126	5,09
	4	5,63	129	5,08
	5	5,56	129	5,05

10.1.2 Calculations

Calculating the density, volume flow and mass flow at the FD fan inlet:

$$\rho = \frac{P_{abs}}{R \times T}$$

$$= \frac{83.3 - 0.2268}{0.2871(273.15 + 36.21)}$$

$$= 0.935\, kg/m^3$$

$$Q = A \times \sqrt{\frac{2 \times P_v}{\rho}}$$

$$= 13.193 \times \sqrt{\frac{2 \times 83.21}{0.935}}$$

$$= 176.011\, m^3/s \quad @\, 72\%\ vane\ position$$

$$\dot{M} = Q \times \rho$$

$$= 176.011 \times 0.935$$

$$= 164.6\, kg/s$$

Calculating the density, volume flow and mass flow from FD fan outlet to air heater inlet

$$\rho = \frac{P_{abs}}{R \times T}$$

$$= \frac{83.3 - 2.13}{0.2871(273.15 + 36.21)}$$

$$= 0.962\, kg/\ \ ^3$$

$$Q = A \times \sqrt{\frac{2 \times P_v}{\rho}}$$

$$= 11.211 \times \sqrt{\frac{2 \times 102.87}{0.962}}$$

$$= 163.951\, m^3/s$$

$$\dot{M} = Q \times \rho$$

$$= 163.951 \times 0.962$$

$$= 157.72\, kg/s$$

Calculating FD fan static pressure rise and static pressure (subscript $_{in}$ for inlet)

$$\Delta P_{sf} = P_{Sout} - P_{sin}$$

$$= 1.986 - (-0.2268)$$

$$= 2.21\, kPa$$

$$P_{sf} = \Delta P_{sf} - P_{vin}$$

$$= 2.21 - 0.08321$$

$$= 2.13\, kPa$$

Calculating the density, volume flow and mass flow of the ID fan inlet

$$\rho = \frac{P_{abs}}{R \times T}$$

$$= \frac{83.4 - 5.38}{0.274(273.15 + 120.07)}$$

$$= 0.724\, kg/m^3$$

$$Q = A\times\sqrt{\frac{2\times P_v}{\rho}}$$

$$= 14.724\times\sqrt{\frac{2\times 162.7}{0.724}}$$

$$= 312.151\, m^3/s \ \ @\, 94\%\ vane\ position$$

$$\dot{M} = Q\times\rho$$

$$= 312.151\times 0.724$$

$$= 225.997\, kg/s$$

Calculating ID fan static pressure rise and static pressure (subscript $_{in}$ for inlet)

$$\Delta P_{sf} = P_{Sout} - P_{sin}$$

$$= 0.1348 - (-5.38)$$

$$= 5.24\, kPa$$

$$P_{sf} = \Delta P_{sf} - P_{vin}$$

$$= 5.24 - 0.1627$$

$$= 5.08\, kPa$$

Calculating the density, volume flow and mass flow of the air heater

$$\rho = \frac{P_{abs}}{R\times T}$$

$$= \frac{83 + 1.2}{0.2871(273.15 + 275.3)}$$

$$= 0.535\, kg/m^3$$

$$Q = A\times \sqrt{\frac{2\times P_v}{\rho}}$$

$$= 14.724\times \sqrt{\frac{2\times 108.4}{0.535}}$$

$$= 296.3\, m^3/s$$

$$\dot{M} = Q\times\rho$$

$$= 296.3\times 0.535$$

$$= 158.520\, kg/s$$

Determine the O_2 factor for leakage using the coal analysis and gas properties program version 2.2.0 of Howden.

$$Factor = \left(\frac{1.28701}{\rho_g} - 1.6011\times k\right)\times 99$$

$$= \left(\frac{1.28701}{1.351} - 1.6011\times 0.03042\right)\times 99$$

$$= 89.5$$

Air heater to turbulator

$$\%\ Leakage = \frac{O_{2in} - O_{2out}}{20.9 - O_{2out}}\times Factor$$

$$= \frac{2.99 - 1.571}{20.9 - 2.99}\times 89.5$$

$$= 7.1\%$$

Air heater to FFP inlet

$$\% \; Leakage = \frac{O_{2i} \quad - O_{2out}}{20.9 - O_{2out}} \times Factor$$

$$= \frac{4.09 - 1.571}{20.9 - 4.09} \times 89.5$$

$$= 13.6\%$$

Air heater to FFP outlet (Overall leakage)

$$\% \; Leakage = \frac{O_{2i} \quad - O_{2out}}{20.9 - O_{2out}} \times Factor$$

$$= \frac{7.38 - 1.571}{20.9 - 7.38} \times 89.5$$

$$= 38.9\%$$

FFP inlet to FFP outlet

$$\% \; Leakage = \frac{O_{2i} \quad - O_{2out}}{20.9 - O_{2out}} \times Factor$$

$$= \frac{7.38 - 4.09}{20.9 - 7.38} \times 89.5$$

$$= 22.1\%$$

10.1.3 Simulations

AIR HEATER DIAGNOSTICS LH

Flue gas in

Tg1	342
Psg1	-1,06
Mg1	443,10
O2in	3,06

Flue gas out

Tg2	159
Psg2	-2,61
O2out	4,57
Mg2	479,34

Combustion air out

Ta2	287,3
Psa2	1,2
Ma2	316,95

Combustion air in

Ta1	32,2
Psa1	1,93
Ma1	353,19

$$E_{in} = E_{out}$$

$$Mg_1 \times \Delta T_g \times Cp_g = Ma_2 \times \Delta T_a \times Cp_a$$

Leakage %
8,18

Undiluted gas exit temp °C
169,4

Temperature differential °C
Gas side 172,63
Air side 255,10

Effectiveness %
Gas side 55,72
Air side 82,34

Pressure drop kPa
Gas side 1,55
Air side 0,73

Pressure differentials kPa
HEPD 2,26
CEPD 4,54

Mean specific heat kJ/kg°C
Gas 1,075
Air 1,017

Mass flows kg/s
Gas in 443,10
Air out 316,95
Gas out 479,34

Log mean temp diff °C
85,76

Heat transfer MW
82,23

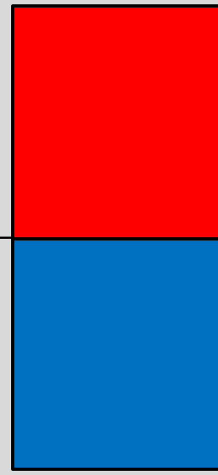
X-ratio
0,68

Figure 27 : Original design data

AIR HEATER DIAGNOSTICS LH

Flue gas in	
Tg1	316
Psg1	-1,06
Mg1	221,22
O2in	1,571

Flue gas out	
Tg2	142
Psg2	-2,71
O2out	4,09
Mg2	250,55



Combustion air out	
Ta2	262
Psa2	1,7
Ma2	165,67

Combustion air in	
Ta1	36,2
Psa1	2,13
Ma1	195

$$E_{in} = E_{out}$$

$$Mg_1 \times \Delta T_g \times Cp_g = Ma_2 \times \Delta T_a \times Cp_a$$

Density air kg/m^3	1,2922
Density gas kg/m^3	0,935
% moisture in Flue gas	0,106
Specific heat gas	1,075
Specific heat air	1,017

Leakage %
13,26

Undiluted gas exit temp °C
156,0

Temperature differential °C

Gas side	159,97
Air side	225,80

Effectiveness %

Gas side	57,17
Air side	80,70

Pressure drop kPa

Gas side	1,65
Air side	0,43

Pressure differentials kPa

HEPD	2,76
CEPD	4,84

Mean specific heat kJ/kg°C

Gas	1,075
Air	1,017

Mass flows kg/s

Gas in	221,22
Air out	165,67
Gas out	250,55

Log mean temp diff °C
77,02

Heat transfer MW
38,04

X-ratio
0,71

Figure 28 : Test data

10.1.4 Combustion model

COMBUSTION, STOÏCHIOMETRIC AIR, EXCESS AIR, AIR LEAKAGE AND FLUEGAS CALCULATIONS									
CHEMICAL REACTIONS									
LH PRACTICAL INCOMPLETE COMBUSTION					RH PRACTICAL INCOMPLETE COMBUSTION				
2C	+	O_2	=	2CO	2C	+	O_2	=	2CO
24.022		31.999		56.021	24.022		31.999		56.021
1.000		1.332		2.332	1.000		1.332		2.332
54.464		72.550		127.014	55.067		73.352		128.419
2CO	+	O_2	=	$2CO_2$	2CO	+	O_2	=	$2CO_2$
56.021		31.999		88.020	56.021		31.999		88.020
1.000		0.571		1.571	1.000		0.571		1.571
127.014		72.550		199.564	128.419		73.352		201.771
127.002		72.543		199.545	128.408		73.346		201.753
S	+	O_2	=	SO_2	S	+	O_2	=	SO_2
32.060		31.999		64.059	32.060		31.999		64.059
1.000		0.998		1.998	1.000		0.998		1.998
0.565		0.564		1.128	0.565		0.564		1.128
$2H_2$	+	O_2	=	$2H_2O$	$2H_2$	+	O_2	=	$2H_2O$
4.032		31.999		36.030	4.032		31.999		36.030
1.000		7.937		8.937	1.000		7.937		8.937
3.015		23.926		26.941	3.015		23.926		26.941
N_2	+	$2O_2$	=	$2NO_2$	N_2	+	$2O_2$	=	$2NO_2$
28.013		63.998		92.011	28.013		63.998		92.011
1.000		2.285		3.285	1.000		2.285		3.285
0.294		0.672		0.966	0.268		0.613		0.881

STOÏCHIOMETRIC GAS (GRAVIMETRIC WET)									
IDEAL				Coalflow [kg/s]		PRACTICAL			
GAS	[kg gas/100kg coal]	[kg/s]	% Gravimetric	14.871	15.615	GAS	[kg gas/100kg coal]	[kg/s]	% Gravimetric
CO_2 (Ostwald CO_2^{++})	199.545	29.674	25.657	Stoïchiometric Oxygen [kg O_2 / 100 kg coal]		CO_2 (Ostwald CO_2^{++})	201.753	31.504	25.718
CO	0.012	0.002	0.001	161.123	162.669	CO	0.011	0.002	0.001
SO_2	1.128	0.168	0.145	Stoïchiometric Air [kg air / 100 kg coal]		SO_2	1.128	0.176	0.144
NOx	0.966	0.144	0.124	695.997	702.675	NOx	0.881	0.138	0.112
O_2	0.000	0.000	0.000	Stoïchiometric Oxygen [kg O_2 / s]		O_2	0.000	0.000	0.000
H_2O from H_2	26.941	4.006	3.464	23.961	25.401	H_2O from H_2	26.941	4.207	3.434
H_2O from coal	7.458	1.109	0.959	Stoïchiometric Air [kg air / s]		H_2O from coal	7.458	1.165	0.951
H_2O from air	5.791	0.861	0.745	103.502	109.723	H_2O from air	5.260	0.821	0.671
N_2 from stoïchiometric air	534.874	79.541	68.773	CO [kg_{CO} / $100kg_{COAL}$]	CO [kg_{CO} / $100kg_{COAL}$]	N_2 from stoïchiometric air	540.006	84.322	68.835

N_2 from excess air	0.000	0.000	0.000	0.011651	0.011356	N_2 from excess air	0.000	0.000	0.000
N_2 from coal	1.026	0.153	0.132	% NO_x volumetric	N_2 for % NOx grav.	N_2 from coal	1.052	0.164	0.134
Total Flue Gas mass	777.742	115.658	100.000	0.294218	0.268262	Total Flue Gas mass	784.492	122.499	100.000
Total H_2O	40.190	5.977	5.168	Stoïchiometric A : F	Stoïchiometric A : F	Total H_2O	39.659	6.193	5.055
Total N_2	535.900	79.694	68.905	6.960	7.027	Total N_2	541.058	84.487	68.969
Gas Temperature	[°C] / [K]	1082.667	1355.817	Gas Vol flow [m³/s]	Gas Vol flow [m³/s]	Gas Temperature	[°C] / [K]	1084.333	1357.483
Gas Pressure	[kPa gauge]/[kPa asb]	-0.080	84.646	469.440	498.727	Gas Pressure	[kPa gauge]/[kPa asb]	-0.140	84.586
Gasflow [Nm³/s]	84.793	Gasflow [Sm³/s]	80.055	Gas density [kg/m³]	Gas density [kg/m³]	Gasflow [Sm³/s]	84.884	Gasflow [Nm³/s]	89.908
Gas density [Nkg/m³]	1.364	Gas density [Skg/m³]	1.445	0.24637	0.24562	Gas density [Skg/m³]	1.443	Gas density [Nkg/m³]	1.362

STOÏCHIOMETRIC GAS (VOLUMETRIC DRY)

IDEAL					PRACTICAL				
GAS	[kg gas/100kg gas]	[kg gas / kmole gas]	[kmole / 100kg]	% Volumetric	GAS	[kg gas/100kg gas]	[kg gas / kmole gas]	[kmole / 100kg]	% Volumetric
CO_2 (Ostwald CO_2^{++})	25.657	44.010	0.583	19.129	CO_2 (Ostwald CO_2^{++})	25.718	44.010	0.584	19.152
CO	0.001	28.010	0.000	0.001755	CO	0.001	28.010	0.000	0.001694
SO_2	0.145	64.059	0.002	0.074	SO_2	0.144	64.059	0.002	0.074
NOx	0.124	46.006	0.003	0.088619	NOx	0.112	46.006	0.002	0.080016
O_2	0.000	31.999	0.000	0.000	O_2	0.000	31.999	0.000	0.000
H_2O from H_2	0.000	18.015	0.000	0.000	H_2O from H_2	0.000	18.015	0.000	0.000
H_2O from coal	0.000	18.015	0.000	0.000	H_2O from coal	0.000	18.015	0.000	0.000
H_2O from air	0.000	18.015	0.000	0.000	H_2O from air	0.000	18.015	0.000	0.000
N_2 from stoïchiometric air	68.773	28.013	2.455	80.552	N_2 from stoïchiometric air	68.835	28.013	2.457	80.535
N_2 from excess air	0.000	28.013	0.000	0.000	N_2 from excess air	0.000	28.013	0.000	0.000
N_2 from coal	0.132	28.013	0.005	0.155	N_2 from coal	0.134	28.013	0.005	0.157
Total Flue Gas	94.832	Molecular mass of Fluegas [kg/kmole]	3.048	100.000	Total Flue Gas	94.945	Molecular mass of Fluegas [kg/kmole]	3.051	100.000
Total H_2O	0.000		$R_{FLUEGAS}$ [kJ/kg-K]	0.000	Total H_2O	0.000		$R_{FLUEGAS}$ [kJ/kg-K]	0.000
Total N_2	68.905	32.81156	0.25340	80.707	Total N_2	68.969	32.77493	0.25368	80.692

BURNER OUTLET GAS, INCLUDING EXCESS AIR, MEASURABLE IN-LEAKAGES & AIRHEATER LEAKAGE (GRAVIMETRIC WET)

IDEAL				Airflow supplied dry [kg/s]		PRACTICAL			
GAS	[kg gas/100kg coal]	[kg/s]	% Gravimetric	106.152	101.082	GAS	[kg gas/100kg coal]	[kg/s]	% Gravimetric
CO_2 (Ostwald CO_2^{+})	199.545	29.674	25.078	Airflow supplied [kg air/100kg coal]		CO_2 (Ostwald CO_2^{+})	201.753	31.504	26.745
CO	0.012	0.002	0.001	713.819	647.338	CO	0.011	0.002	0.002
SO_2	1.128	0.168	0.142	Excess air [kg air / 100kg coal]		SO_2	1.128	0.176	0.150
NOx	0.966	0.144	0.121	17.822	-55.337	NOx	0.881	0.138	0.117
O_2	4.126	0.614	0.518	Excess air [kg/s]		O_2	12.811	2.000	1.698
H_2O from H_2	26.941	4.006	3.386	2.650	-8.641	H_2O from H_2	26.941	4.207	3.571

H_2O from coal	7.458	1.109	0.937	Excess air [%]		H_2O from coal	7.458	1.165	0.989
H_2O from air	5.939	0.883	0.746	2.561	-7.875	H_2O from air	4.846	0.757	0.642
N_2 from stoichiometric air	534.874	79.541	67.220	A : F		N_2 from stoichiometric air	540.006	84.322	71.585
N_2 from excess air	13.696	2.037	1.721	7.138	6.473	N_2 from excess air	-42.527	-6.641	-5.637
N_2 from coal	1.026	0.153	0.129	Anticipated Air Duct leakage [kg/s]		N_2 from coal	1.052	0.164	0.140
Total Flue Gas	**795.712**	**118.330**	**100.000**	0.000	0.000	**Total Flue Gas**	**754.361**	**117.794**	**100.000**
Total H_2O	40.338	5.999	5.069	Anticipated Airheater leakage [kg/s]		Total H_2O	39.245	6.128	5.202
Total N_2	549.596	81.731	69.070	9.25133	12.69812	Total N_2	498.531	77.846	66.087
Gas Temperature	[°C] / [K]	1082.667	1355.817	Gas Vol flow [m^3/s]	Gas Vol flow [m^3/s]	Gas Temperature	[°C] / [K]	1084.333	1357.483
Gas Pressure	[kPa gauge]/[kPa asb]	-0.080	84.646	481.676	475.438	Gas Pressure	[kPa gauge]/[kPa asb]	-0.140	84.586
Gasflow [Nm^3/s]	87.003	Gasflow [Sm^3/s]	82.141	Gas density [kg/m^3]	Gas density [kg/m^3]	Gasflow [Sm^3/s]	80.921	Gasflow [Nm^3/s]	85.710
Gas density [Nkg/m^3]	1.360	Gas density [Skg/m^3]	1.441	0.246	0.248	Gas density [Skg/m^3]	1.456	Gas density [Nkg/m^3]	1.374

BURNER OUTLET GAS, INCLUDING EXCESS AIR, MEASURABLE IN-LEAKAGES & AIRHEATER LEAKAGE (VOLUMETRIC DRY)									
IDEAL					PRACTICAL				
GAS	[kg gas/100kg gas]	[kg gas / kmole gas]	[kmole / 100kg]	% Volumetric	GAS	[kg gas/100kg gas]	[kg gas / kmole gas]	[kmole / 100kg]	% Volumetric
CO_2 (Ostwald CO_2^+)	25.078	44.010	0.570	18.643	CO_2 (Ostwald CO_2^+)	26.745	44.010	0.608	20.091
CO	0.001	28.010	0.000	0.001710	CO	0.002	28.010	0.000	0.001777
SO_2	0.142	64.059	0.002	0.072	SO_2	0.150	64.059	0.002	0.077
NOx	0.121	46.006	0.003	0.086368	NOx	0.117	46.006	0.003	0.083936
O_2	0.518	31.999	0.016	0.530	O_2	1.698	31.999	0.053	1.755
H_2O from H_2	0.000	18.015	0.000	0.000	H_2O from H_2	0.000	18.015	0.000	0.000
H_2O from coal	0.000	18.015	0.000	0.000	H_2O from coal	0.000	18.015	0.000	0.000
H_2O from air	0.000	18.015	0.000	0.000	H_2O from air	0.000	18.015	0.000	0.000
N_2 from stoichiometric air	67.220	28.013	2.400	78.506	N_2 from stoichiometric air	71.585	28.013	2.555	84.480
N_2 from excess air	1.721	28.013	0.061	2.010	N_2 from excess air	-5.637	28.013	-0.201	-6.653
N_2 from coal	0.129	28.013	0.005	0.151	N_2 from coal	0.140	28.013	0.005	0.165
Total Flue Gas	**94.931**	Molecular mass of Fluegas [kg/kmole]	**3.057**	**100.000**	Total Flue Gas	**94.798**	Molecular mass of Fluegas [kg/kmole]	**3.025**	**100.000**
Total H_2O	0.000		$R_{FLUEGAS}$ [kJ/kg-K]	0.000	Total H_2O	0.000		$R_{FLUEGAS}$ [kJ/kg-K]	0.000
Total N_2	69.070	**32.71689**	**0.25413**	80.667	Total N_2	66.087	**33.05995**	**0.25150**	77.992

ECONOMISER OUTLET GAS INCLUDING EXCESS AIR, MEASURABLE IN-LEAKAGES, AIRHEATER LEAKAGE & UNMEASURABLE FURNACE IN-LEAKAGE (GRAVIMETRIC WET)									
IDEAL				Airflow supplied dry [kg/s]		PRACTICAL			
GAS	[kg gas/100kg coal]	[kg/s]	% Gravimetric	119.260	128.531	GAS	[kg gas/100kg coal]	[kg/s]	% Gravimetric
CO_2 (Ostwald CO_2^+)	199.545	29.674	22.558	Airflow supplied [kg air/100kg coal]		CO_2 (Ostwald CO_2^+)	201.753	31.504	22.272
CO	0.012	0.002	0.001	801.964	823.123	CO	0.011	0.002	0.001

SO_2	1.128	0.168	0.128	Excess air [kg air / 100kg coal]		SO_2	1.128	0.176	0.125
NOx	0.966	0.144	0.109	105.967	120.448	NOx	0.881	0.138	0.097
O_2	24.531	3.648	2.773	Excess air [kg/s]		O_2	27.884	4.354	3.078
H_2O from H_2	26.941	4.006	3.046	15.758	18.808	H_2O from H_2	26.941	4.207	2.974
H_2O from coal	7.458	1.109	0.843	Excess air [%]		H_2O from coal	7.458	1.165	0.823
H_2O from air	6.672	0.992	0.754	15.225	17.141	H_2O from air	6.162	0.962	0.680
N_2 from stoïchiometric air	534.874	79.541	60.466	A : F		N_2 from stoïchiometric air	540.006	84.322	59.614
N_2 from excess air	81.436	12.110	9.206	8.020	8.231	N_2 from excess air	92.564	14.454	10.219
N_2 from coal	1.026	0.153	0.116	Econ Outlet CO_2 / O_2 [%] Volumetric Dry		N_2 from coal	1.052	0.164	0.116
Total Flue Gas	884.590	131.548	100.000	16.680	16.410	Total Flue Gas	905.841	141.448	100.000
Total H_2O	41.072	6.108	4.643	2.800	3.100	Total H_2O	40.561	6.334	4.478
Total N_2	617.336	91.804	69.788	Anticipated Furnace in-leakage [kg/s]		Total N_2	633.622	98.941	69.949
Total check	884.590			13.108	27.449	Total check	905.841		
Gas Temperature	[°C] / [K]	322.000	595.150	Gas Vol flow [m³/s]	Gas Vol flow [m³/s]	Gas Temperature	[°C] / [K]	317.000	590.150
Gas Pressure	[kPa gauge]/[kPa asb]	-0.380	84.346	238.849	255.425	Gas Pressure	[kPa gauge]/[kPa asb]	-0.400	84.326
Gasflow [Nm³/s]	97.934	Gasflow [Sm³/s]	92.462	Gas density [kg/m³]	Gas density [kg/m³]	Gasflow [Sm³/s]	99.693	Gasflow [Nm³/s]	105.593
Gas density [Nkg/m³]	1.343	Gas density [Skg/m³]	1.423	0.551	0.554	Gas density [Skg/m³]	1.419	Gas density [Nkg/m³]	1.340

ECONOMISER OUTLET GAS INCLUDING EXCESS AIR, MEASURABLE IN-LEAKAGES, AIRHEATER LEAKAGE & UNMEASURABLE FURNACE IN-LEAKAGE (VOLUMETRIC DRY)

IDEAL					PRACTICAL				
GAS	[kg gas/100kg gas]	[kg gas / kmole gas]	[kmole / 100kg]	% Volumetric	GAS	[kg gas/100kg gas]	[kg gas / kmole gas]	[kmole / 100kg]	% Volumetric
CO_2 (Ostwald CO_2^+)	22.558	44.010	0.513	16.562	CO_2 (Ostwald CO_2^+)	22.272	44.010	0.506	16.308
CO	0.001	28.010	0.000	0.001519	CO	0.001	28.010	0.000	0.001442
SO_2	0.128	64.059	0.002	0.064	SO_2	0.125	64.059	0.002	0.063
NOx	0.109	46.006	0.002	0.076727	NOx	0.097	46.006	0.002	0.068130
O_2	2.773	31.999	0.087	2.800	O_2	3.078	31.999	0.096	3.100
H_2O from H_2	0.000	18.015	0.000	0.000	H_2O from H_2	0.000	18.015	0.000	0.000
H_2O from coal	0.000	18.015	0.000	0.000	H_2O from coal	0.000	18.015	0.000	0.000
H_2O from air	0.000	18.015	0.000	0.000	H_2O from air	0.000	18.015	0.000	0.000
N_2 from stoïchiometric air	60.466	28.013	2.158	69.743	N_2 from stoïchiometric air	59.614	28.013	2.128	68.573
N_2 from excess air	9.206	28.013	0.329	10.619	N_2 from excess air	10.219	28.013	0.365	11.754
N_2 from coal	0.116	28.013	0.004	0.134	N_2 from coal	0.116	28.013	0.004	0.134
Total Flue Gas	95.357	Molecular mass of Fluegas [kg/kmole]	3.095	100.000	Total Flue Gas	95.522	Molecular mass of Fluegas [kg/kmole]	3.103	100.000
Total H_2O	0.000		$R_{FLUEGAS}$ [kJ/kg-K]	0.000	Total H_2O	0.000		$R_{FLUEGAS}$ [kJ/kg-K]	0.000
Total N_2	69.788	32.31151	0.25732	80.495	Total N_2	69.949	32.22327	0.25803	80.460

AIRHEATER OUTLET GAS INCLUDING EXCESS AIR, MEASURABLE IN-LEAKAGES, AIRHEATER LEAKAGE & UNMEASURABLE FURNACE & AIRHEATER IN-LEAKAGE (GRAVIMETRIC WET)

IDEAL				Airflow supplied dry [kg/s]		PRACTICAL			
GAS	[kg gas/100kg coal]	[kg/s]	% Gravimetric	128.512	141.229	GAS	[kg gas/100kg coal]	[kg/s]	% Gravimetric
CO_2 (Ostwald CO_2^+)	199.545	29.674	21.064	Airflow supplied [kg air/100kg coal]		CO_2 (Ostwald CO_2^+)	201.753	31.504	20.425
CO	0.012	0.002	0.001	864.174	904.443	CO	0.011	0.002	0.001
SO_2	1.128	0.168	0.119	Excess air [kg air / 100kg coal]		SO_2	1.128	0.176	0.114
NOx	0.966	0.144	0.102	168.177	201.767	NOx	0.881	0.138	0.089
O_2 (Ostwald $O2^+$)	38.933	5.790	4.110	Excess air [kg/s]		O_2 (Ostwald $O2^+$)	46.709	7.294	4.729
H_2O from H_2	26.941	4.006	2.844	25.010	31.506	H_2O from H_2	26.941	4.207	2.727
H_2O from coal	7.458	1.109	0.787	Excess air [%]		H_2O from coal	7.458	1.165	0.755
H_2O from air	7.190	1.069	0.759	24.164	28.714	H_2O from air	6.771	1.057	0.685
N_2 from stoïchiometric air	534.874	79.541	56.462	A : F		N_2 from stoïchiometric air	540.006	84.322	54.669
N_2 from excess air	129.244	19.220	13.643	8.642	9.044	N_2 from excess air	155.058	24.212	15.698
N_2 from coal	1.026	0.153	0.108	Airheater outlet CO_2 / O_2 % (Volumetric dry)		N_2 from coal	1.052	0.164	0.107
Total Flue Gas	947.318	140.876	100.000	15.470	14.920	Total Flue Gas	987.769	154.241	100.000
Total H_2O	41.589	6.185	4.390	4.120	4.720	Total H_2O	41.170	6.429	4.168
Total N_2	665.145	98.914	70.213	Anticipated Airheater in-leakage [kg/s]		Total N_2	696.117	108.699	70.474
Gas Temperature	[°C] / [K]	145.100	418.250	9.251	12.698	Gas Temperature	[°C] / [K]	155.300	428.450
Gas Pressure	[kPa gauge]/[kPa asb]	-1.100	83.626	Gas Vol flow [m^3/s]	Gas Vol flow [m^3/s]	Gas Pressure	[kPa gauge]/[kPa asb]	-1.150	83.576
Gasflow [Nm^3/s]	105.650	Gasflow [Sm^3/s]	99.746	182.637	205.867	Gasflow [Sm^3/s]	109.691	Gasflow [Nm^3/s]	116.183
Gas density [Nkg/m^3]	1.333	Gas density [Skg/m^3]	1.412	0.771	0.749	Gas density [Skg/m^3]	1.406	Gas density [Nkg/m^3]	1.328

AIRHEATER OUTLET GAS INCLUDING EXCESS AIR, MEASURABLE IN-LEAKAGES, AIRHEATER LEAKAGE & UNMEASURABLE FURNACE & AIRHEATER IN-LEAKAGE (VOLUMETRIC DRY)

IDEAL					PRACTICAL				
GAS	[kg gas/100kg gas]	[kg gas / kmole gas]	[kmole / 100kg]	% Volumetric	GAS	[kg gas/100kg gas]	[kg gas / kmole gas]	[kmole / 100kg]	% Volumetric
CO_2 (Ostwald CO_2^+)	21.064	44.010	0.479	15.352	CO_2 (Ostwald CO_2^+)	20.425	44.010	0.464	14.821
CO	0.001	28.010	0.000	0.001408	CO	0.001	28.010	0.000	0.001311
SO_2	0.119	64.059	0.002	0.060	SO_2	0.114	64.059	0.002	0.057
NOx	0.102	46.006	0.002	0.071124	NOx	0.089	46.006	0.002	0.061921
O_2	4.110	31.999	0.128	4.120	O_2	4.729	31.999	0.148	4.719
H_2O from H_2	0.000	18.015	0.000	0.000	H_2O from H_2	0.000	18.015	0.000	0.000
H_2O from coal	0.000	18.015	0.000	0.000	H_2O from coal	0.000	18.015	0.000	0.000
H_2O from air	0.000	18.015	0.000	0.000	H_2O from air	0.000	18.015	0.000	0.000
N_2 from stoïchiometric air	56.462	28.013	2.016	64.650	N_2 from stoïchiometric air	54.669	28.013	1.952	62.322
N_2 from excess air	13.643	28.013	0.487	15.622	N_2 from excess air	15.698	28.013	0.560	17.895
N_2 from coal	0.108	28.013	0.004	0.124	N_2 from coal	0.107	28.013	0.004	0.121

Total Flue Gas	95.610	Molecular mass of Fluegas [kg/kmole]	3.118	100.000	Total Flue Gas	95.832	Molecular mass of Fluegas [kg/kmole]	3.131	100.000
Total H_2O	0.000		$R_{FLUEGAS}$ [kJ/kg-K]	0.000	Total H_2O	0.000		$R_{FLUEGAS}$ [kJ/kg-K]	0.000
Total N_2	70.213	32.07590	0.25921	80.396	Total N_2	70.474	31.93504	0.26036	80.339

PRECIPITATOR INLET GAS INCLUDING EXCESS AIR, MEASURABLE IN-LEAKAGES, AIRHEATER, & UNMEASURABLE FURNACE LEAKAGE (GRAVIMETRIC WET)

IDEAL				Airflow supplied dry [kg/s]		PRACTICAL			
GAS	[kg gas/100kg coal]	[kg/s]	% Gravimetric	128.512	141.229	GAS	[kg gas/100kg coal]	[kg/s]	% Gravimetric
CO_2 (Ostwald CO_2^+)	199.545	29.674	21.064	Airflow supplied [kg air/100kg coal]		CO_2 (Ostwald CO_2^+)	201.753	31.504	20.425
CO	0.012	0.002	0.001	864.174	904.443	CO	0.011	0.002	0.001
SO_2	1.128	0.168	0.119	Excess air [kg air / 100kg coal]		SO_2	1.128	0.176	0.114
NOx	0.966	0.144	0.102	168.177	201.767	NOx	0.881	0.138	0.089
O_2 (Ostwald $O2^+$)	38.933	5.790	4.110	Excess air [kg/s]		O_2 (Ostwald $O2^+$)	46.709	7.294	4.729
H_2O from H_2	26.941	4.006	2.844	25.010	31.506	H_2O from H_2	26.941	4.207	2.727
H_2O from coal	7.458	1.109	0.787	Excess air [%]		H_2O from coal	7.458	1.165	0.755
H_2O from air	7.190	1.069	0.759	24.164	28.714	H_2O from air	6.771	1.057	0.685
N_2 from stoïchiometric air	534.874	79.541	56.462	A : F		N_2 from stoïchiometric air	540.006	84.322	54.669
N_2 from excess air	129.244	19.220	13.643	8.642	9.044	N_2 from excess air	155.058	24.212	15.698
N_2 from coal	1.026	0.153	0.108	Precip inlet O_2, CO_2 % (Volumetric dry)		N_2 from coal	1.052	0.164	0.107
Total Flue Gas	**947.318**	**140.876**	**100.000**	15.470	14.920	Total Flue Gas	**987.769**	**154.241**	100.000
Total H_2O	41.589	6.185	4.390	4.120	4.720	Total H_2O	41.170	6.429	4.168
Total N_2	665.145	98.914	70.213	Anticipated Airheater in-leakage [kg/s]		Total N_2	696.117	108.699	70.474
Gas Temperature	[°C] / [K]	145.100	418.250	9.251	12.698	Gas Temperature	[°C] / [K]	155.300	428.450
Gas Pressure	[kPa gauge]/[kPa asb]	-1.100	83.626	Gas Vol flow [m³/s]	Gas Vol flow [m³/s]	Gas Pressure	[kPa gauge]/[kPa asb]	-1.150	83.576
Gasflow [Nm³/s]	710.439	Gasflow [Sm³/s]	99.746	182.637	205.867	Gasflow [Sm³/s]	109.691	Gasflow [Nm³/s]	116.183
Gas density [Nkg/m³]	1.333	Gas density [Skg/m³]	1.412	0.771	0.749	Gas density [Skg/m³]	1.406	Gas density [Nkg/m³]	1.328

PRECIPITATOR INLET GAS INCLUDING EXCESS AIR, MEASURABLE IN-LEAKAGES, AIRHEATER, & UNMEASURABLE FURNACE LEAKAGE (VOLUMETRIC DRY)

IDEAL					PRACTICAL				
GAS	[kg gas/100kg gas]	[kg gas / kmole gas]	[kmole / 100kg]	% Volumetric	GAS	[kg gas/100kg gas]	[kg gas / kmole gas]	[kmole / 100kg]	% Volumetric
CO_2 (Ostwald CO_2^+)	21.064	44.010	0.479	15.352	CO_2 (Ostwald CO_2^+)	20.425	44.010	0.464	14.821
CO	0.001	28.010	0.000	0.001408	CO	0.001	28.010	0.000	0.001311
SO_2	0.119	64.059	0.002	0.060	SO_2	0.114	64.059	0.002	0.057
NOx	0.102	46.006	0.002	0.071124	NOx	0.089	46.006	0.002	0.061921
O_2	4.110	31.999	0.128	4.120	O_2	4.729	31.999	0.148	4.719
H_2O from H_2	0.000	18.015	0.000	0.000	H_2O from H_2	0.000	18.015	0.000	0.000
H_2O from coal	0.000	18.015	0.000	0.000	H_2O from coal	0.000	18.015	0.000	0.000
H_2O from air	0.000	18.015	0.000	0.000	H_2O from air	0.000	18.015	0.000	0.000

N_2 from stoïchiometric air	56.462	28.013	2.016	64.650	N_2 from stoïchiometric air	54.669	28.013	1.952	62.322
N_2 from excess air	13.643	28.013	0.487	15.622	N_2 from excess air	15.698	28.013	0.560	17.895
N_2 from coal	0.108	28.013	0.004	0.124	N_2 from coal	0.107	28.013	0.004	0.121
Total Flue Gas	**95.610**	Molecular mass of Fluegas [kg/kmole]	**3.118**	**100.000**	Total Flue Gas	**95.832**	Molecular mass of Fluegas [kg/kmole]	**3.131**	**100.000**
Total H_2O	0.000		$R_{FLUEGAS}$ [kJ/kg-K]	0.000	Total H_2O	0.000		$R_{FLUEGAS}$ [kJ/kg-K]	0.000
Total N_2	70.213	**32.07590**	**0.25921**	80.396	Total N_2	70.474	**31.93504**	**0.26036**	80.339

PRECIPITATOR OUTLET GAS INCLUDING EXCESS AIR, MEASURABLE IN-LEAKAGES, AIRHEATER, UNMEASURABLE FURNACE, & PRECIPITATOR IN-LEAKAGE (GRAVIMETRIC WET)

IDEAL				Airflow supplied dry [kg/s]		PRACTICAL			
GAS	[kg gas/100kg coal]	[kg/s]	% Gravimetric	129.349	142.535	GAS	[kg gas/100kg coal]	[kg/s]	% Gravimetric
CO_2 (Ostwald CO_2^+)	199.545	29.674	20.939	Airflow supplied [kg air/100kg coal]		CO_2 (Ostwald CO_2^+)	201.753	31.504	20.252
CO	0.012	0.002	0.001	869.804	912.805	CO	0.011	0.002	0.001
SO_2	1.128	0.168	0.118	Excess air [kg air / 100kg coal]		SO_2	1.128	0.176	0.113
NOx	0.966	0.144	0.101	173.807	210.130	NOx	0.881	0.138	0.088
O_2 (Ostwald $O2^+$)	40.236	5.984	4.222	Excess air [kg/s]		O_2 (Ostwald $O2^+$)	48.645	7.596	4.883
H_2O from H_2	26.941	4.006	2.827	25.847	32.812	H_2O from H_2	26.941	4.207	2.704
H_2O from coal	7.458	1.109	0.783	Excess air [%]		H_2O from coal	7.458	1.165	0.749
H_2O from air	7.237	1.076	0.759	**24.972**	**29.904**	H_2O from air	6.833	1.067	0.686
N_2 from stoïchiometric air	534.874	79.541	56.126	A : F		N_2 from stoïchiometric air	540.006	84.322	54.207
N_2 from excess air	133.571	19.863	14.016	**8.698**	**9.128**	N_2 from excess air	161.485	25.216	16.210
N_2 from coal	1.026	0.153	0.108	Precip outlet O_2, CO_2 % (Volumetric dry)		N_2 from coal	1.052	0.164	0.106
Total Flue Gas	**952.995**	**141.720**	**100.000**	4.230	4.870	Total Flue Gas	**996.195**	**155.557**	**100.000**
Total H_2O	41.636	6.192	4.369	0.000	0.000	Total H_2O	41.232	6.438	4.139
Total N_2	669.471	99.557	70.249	Anticipated Precipitator in-leakage [kg/s]		Total N_2	702.543	109.703	70.523
Gas Temperature	[°C] / [K]	144.000	417.150	**0.837**	**1.306**	Gas Temperature	[°C] / [K]	144.000	417.150
Gas Pressure	[kPa gauge]/[kPa asb]	-1.300	83.426	Gas Vol flow [m³/s]	Gas Vol flow [m³/s]	Gas Pressure	[kPa gauge]/[kPa asb]	-0.250	84.476
Gasflow [Nm³/s]	**106.348**	Gasflow [Sm³/s]	**100.405**	**183.800**	**200.161**	Gasflow [Sm³/s]	**110.719**	Gasflow [Nm³/s]	**117.272**
Gas density [Nkg/m³]	**1.333**	Gas density [Skg/m³]	**1.411**	**0.771**	**0.777**	Gas density [Skg/m³]	**1.405**	Gas density [Nkg/m³]	**1.326**

PRECIPITATOR OUTLET GAS INCLUDING EXCESS AIR, MEASURABLE IN-LEAKAGES, AIRHEATER, UNMEASURABLE FURNACE, & PRECIPITATOR IN-LEAKAGE (VOLUMETRIC DRY)

IDEAL					PRACTICAL				
GAS	[kg gas/100kg gas]	[kg gas / kmole gas]	[kmole / 100kg]	% Volumetric	GAS	[kg gas/100kg gas]	[kg gas / kmole gas]	[kmole / 100kg]	% Volumetric
CO_2 (Ostwald CO_2^+)	20.939	44.010	0.476	**15.252**	CO_2 (Ostwald CO_2^+)	20.252	44.010	0.460	**14.684**
CO	0.001	28.010	0.000	0.001399	CO	0.001	28.010	0.000	0.001299
SO_2	0.118	64.059	0.002	0.059	SO_2	0.113	64.059	0.002	0.056
NOx	0.101	46.006	0.002	0.070657	NOx	0.088	46.006	0.002	0.061346

O_2 (Ostwald $O2^{+}$)	4.222	31.999	0.132	4.230	O_2 (Ostwald $O2^{+}$)	4.883	31.999	0.153	4.869
H_2O from H_2	0.000	18.015	0.000	0.000	H_2O from H_2	0.000	18.015	0.000	0.000
H_2O from coal	0.000	18.015	0.000	0.000	H_2O from coal	0.000	18.015	0.000	0.000
H_2O from air	0.000	18.015	0.000	0.000	H_2O from air	0.000	18.015	0.000	0.000
N_2 from stoïchiometric air	56.126	28.013	2.004	64.226	N_2 from stoïchiometric air	54.207	28.013	1.935	61.744
N_2 from excess air	14.016	28.013	0.500	16.039	N_2 from excess air	16.210	28.013	0.579	18.464
N_2 from coal	0.108	28.013	0.004	0.123	N_2 from coal	0.106	28.013	0.004	0.120
Total Flue Gas	95.631	Molecular mass of Fluegas [kg/kmole]	3.120	100.000	Total Flue Gas	95.861	Molecular mass of Fluegas [kg/kmole]	3.134	100.000
Total H_2O	0.000		$R_{FLUEGAS}$ [kJ/kg-K]	0.000	Total H_2O	0.000		$R_{FLUEGAS}$ [kJ/kg-K]	0.000
Total N_2	70.249	32.05627	0.25937	80.387	Total N_2	70.523	31.90835	0.26057	80.328

ID FAN OUTLET GAS INCLUDING EXCESS AIR, MEASURABLE IN-LEAKAGES, AIRHEATER, UNMEASURABLE FURNACE, & PRECIPITATOR IN-LEAKAGE (GRAVIMETRIC WET)

IDEAL				Airflow supplied dry [kg/s]		PRACTICAL			
GAS	[kg gas/100kg coal]	[kg/s]	% Gravimetric	129.349	142.535	GAS	[kg gas/100kg coal]	[kg/s]	% Gravimetric
CO_2 (Ostwald CO_2^{+})	199.545	29.674	20.939	Airflow supplied [kg air/100kg coal]		CO_2 (Ostwald CO_2^{+})	201.753	31.504	20.252
CO	0.012	0.002	0.001	869.804	912.805	CO	0.011	0.002	0.001
SO_2	1.128	0.168	0.118	Excess air [kg air / 100kg coal]		SO_2	1.128	0.176	0.113
NOx	0.966	0.144	0.101	173.807	210.130	NOx	0.881	0.138	0.088
O_2 (Ostwald $O2^{+}$)	40.236	5.984	4.222	Excess air [kg/s]		O_2 (Ostwald $O2^{+}$)	48.645	7.596	4.883
H_2O from H_2	26.941	4.006	2.827	25.847	32.812	H_2O from H_2	26.941	4.207	2.704
H_2O from coal	7.458	1.109	0.783	Excess air [%]		H_2O from coal	7.458	1.165	0.749
H_2O from air	7.237	1.076	0.759	24.972	29.904	H_2O from air	6.833	1.067	0.686
N_2 from stoïchiometric air	534.874	79.541	56.126	A : F		N_2 from stoïchiometric air	540.006	84.322	54.207
N_2 from excess air	133.571	19.863	14.016	8.698	9.128	N_2 from excess air	161.485	25.216	16.210
N_2 from coal	1.026	0.153	0.108	ID Fan outlet O_2, CO_2 % (Volumetric dry)		N_2 from coal	1.052	0.164	0.106
Total Flue Gas	952.995	141.720	100.000	4.230	4.870	Total Flue Gas	996.195	155.557	100.000
Total H_2O	41.636	6.192	4.369	15.370	14.790	Total H_2O	41.232	6.438	4.139
Total N_2	669.471	99.557	70.249	Anticipated Precipitator in-leakage [kg/s]		Total N_2	702.543	109.703	70.523
Gas Temperature	[°C] / [K]	149.200	422.350	0.837	1.306	Gas Temperature	[°C] / [K]	154.600	427.750
Gas Pressure	[kPa gauge]/[kPa asb]	-0.380	84.346	Gas Vol flow [m³/s]	Gas Vol flow [m³/s]	Gas Pressure	[kPa gauge]/[kPa asb]	-0.400	84.326
Gasflow [Nm³/s]	106.348	Gasflow [Sm³/s]	100.405	184.061	205.612	Gasflow [Sm³/s]	110.719	Gasflow [Nm³/s]	117.272
Gas density [Nkg/m³]	1.333	Gas density [Skg/m³]	1.411	0.770	0.757	Gas density [Skg/m³]	1.405	Gas density [Nkg/m³]	1.326

ID FAN OUTLET GAS INCLUDING EXCESS AIR, MEASURABLE IN-LEAKAGES, AIRHEATER, UNMEASURABLE FURNACE, & PRECIPITATOR IN-LEAKAGE (VOLUMETRIC DRY)

IDEAL					PRACTICAL				
GAS	[kg gas/100kg gas]	[kg gas / kmole gas]	[kmole / 100kg]	% Volumetric	GAS	[kg gas/100kg gas]	[kg gas / kmole gas]	[kmole / 100kg]	% Volumetric
CO_2 (Ostwald CO_2^{+})	20.939	44.010	0.476	15.252	CO_2 (Ostwald CO_2^{+})	20.252	44.010	0.460	14.684

CO	0.001	28.010	0.000	0.001399	CO	0.001	28.010	0.000	0.001299
SO_2	0.118	64.059	0.002	0.059	SO_2	0.113	64.059	0.002	0.056
NOx	0.101	46.006	0.002	0.070657	NOx	0.088	46.006	0.002	0.061346
O_2	4.222	31.999	0.132	4.230	O_2	4.883	31.999	0.153	4.869
H_2O from H_2	0.000	18.015	0.000	0.000	H_2O from H_2	0.000	18.015	0.000	0.000
H_2O from coal	0.000	18.015	0.000	0.000	H_2O from coal	0.000	18.015	0.000	0.000
H_2O from air	0.000	18.015	0.000	0.000	H_2O from air	0.000	18.015	0.000	0.000
N_2 from stoïchiometric air	56.126	28.013	2.004	64.226	N_2 from stoïchiometric air	54.207	28.013	1.935	61.744
N_2 from excess air	14.016	28.013	0.500	16.039	N_2 from excess air	16.210	28.013	0.579	18.464
N_2 from coal	0.108	28.013	0.004	0.123	N_2 from coal	0.106	28.013	0.004	0.120
Total Flue Gas	95.631	Molecular mass of Fluegas [kg/kmole]	3.120	100.000	Total Flue Gas	95.861	Molecular mass of Fluegas [kg/kmole]	3.134	100.000
Total H_2O	0.000		$R_{FLUEGAS}$ [kJ/kg-K]	0.000	Total H_2O	0.000		$R_{FLUEGAS}$ [kJ/kg-K]	0.000
Total N_2	70.249	32.05627	0.25937	80.387	Total N_2	70.523	31.90835	0.26057	80.328