Eirik Torp, Master thesis, NTNU 2018
Title: Economic considerations and operational improvements of an existing biomass fired CFB boiler

Abstract

Norske Skog's paper mill at Skogn,Norway, needs steam for its paper machines and various other parts of the mill. The main steam producer is a 50MW CFB boiler, which burns a range of different fuels. It is currently constrained below its rated capacity. The aim of this thesis has been to investigate reasons for this, as well as quantify potential losses and suggest improvements.

The reasons for the constrained boiler is that air is leaking into the flue gas system at various points. This increases the total amount of gas in the flue gas system, and causes the flue gas fan to become a bottleneck for the operation.

The economic loss associated with this is estimated for the period 3 Jun 2017 - 4 Jun 2018 to be 8.57 MNOK. This should provide a decent incentive for fixing the leaks.

There have also been some investigations into other parts of the process, something that lead to the decicion of turning off a simple pump. This small change is estimated to save the company thousands of NOK each month.

Sammendrag

Norske Skogs papirfabrikk i Skogn, Norge, trenger damp til sine papirmaskiner og ulike andre deler av fabrikken. Hoveddampprodusenten er en 50MW CFB-kjele, som brenner en rekke forskjellige typer brensel. Den er for tiden begrenset under sin nominelle kapasitet. Målet med denne oppgaven har vært å undersøke årsaker til dette, samt kvantifisere potensielle tap og foreslå forbedringer.

Årsaken til at kjelen er begrenset er at luft lekker inn i røykgassystemet på forskjellige steder. Dette øker den totale mengden gass i røykgassystemet, og får røykgassviften til å bli en flaskehals for dampproduskjonen.

Det økonomiske tapet i forbindelse med dette er estimert for perioden 3. juni 2017 - 4. juni 2018 til å være 8,57 MNOK. Dette bør gi et godt argument for å reparere disse lekkasjene.

Det har også vært noen undersøkelser i andre deler av prosessen. Dette har ført til en avgjørelse om å slå av en enkelt pumpe. Denne lille endringen er beregnet å spare selskapet tusenvis av kroner hver måned.

Preface

This master's thesis was written as the final part of the study program Chemical Engineering at the Norwegian University of Science and Technology (NTNU). The thesis was written for the Process System Engineering group at the Department of Chemical Engineering in cooperation with Norske Skog Skogn.

I would like to thank my supervisor Prof. Sigurd Skogestad, and my co-supervisors Andreas Burheim Volden and Cristina Zotica. They have all three been extremely helpful throughout the work with this thesis and vital for the completion of it. I would also like to thank Norske Skog Skogn for letting me write my thesis in collaboration with them.

Declaration of Compliance

I declare that this is an independent work according to the exam regulations of the Norwegian University of Science and Technology (NTNU).

Trondheim, 08.12.2018

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Eirik Torp

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List of Symbols

Symbol	Description	Unit
$_{Air}$	Subscript. Refers to air	-
Cp_{Air}	Molar heat capacity of air	J/(K·mol)
$Cp_{Air,T}$	Molar heat capacity of air at a certain T	J/(K·mol)
$Cp_{C,T}$	Molar heat capacity of component C at a certain T	J/(K·mol)
Cp_f	Specific heat capacity of air/flue gas mixture	kJ/(kg·K)
Cp_w	Specific heat capacity of water	kJ/(kg·K)
$_{Flue}$	Subscript. Refers to flue gas	-
$_{f,in}$	Subscript. Air/flue gas mixture into ECO1	
$_{f,out}$	Subscript. Air/flue gas mixture out of ECO1	
$\Delta\dot{H}$	Enthalpy flow rate	kJ/(kg·s)
$\Delta h_{F,w}$	Specific enthalpy of formation of water	J/(kg·s)
$\dot{m}$	Mass flow rate	kg/s
$_{Mix}$	Subscript. Refers to air/flue gas mixture	-
M_{Mix}	Molar mass of air/flue gas mixture	kg/mol
$\dot{n}$	Molar flow rate	mol/s
ΔP	Pressure change	bar
$\dot{Q}_{Heat,Air}$	Heat needed to heat air leaking in	W

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Symbol	**Description**	**Unit**
q	Volumetric flow rate	m^3/s
R	Universal gas constant	J/(K·mol)
T	Temperature	K or °C
ΔT_m	Logarithmic mean temperature difference	K or °C
T_{Room}	Temperature boiler room	°C
T_0	Standard temperature	K or °C
UA	Heat transfer coefficient	W/K
$_{w,feed}$	Subscript. Feed water	-
$_{w,in}$	Subscript. Water inlet to ECO1	-
$_{w,recycle}$	Subscript. Water recycle	-
x_{Air}	Molar fraction of air in flue gas mixture	-
x_{O_2}	Molar fraction of O_2	-
ρ	Density	kg/m^3

List of Abbreviations

Abbreviation	Description
AP	Air preheater
CC	Composition controller
CFB	Circulating fluidized bed
CV	Controlled variable
DOF	Degrees of freedom
ECO	Economizer
ECO1	Economizer 1 (First part of the Economizer)
ECO2	Economizer 2 (Second part of the Economizer)
ESP	Electrostatic percipitator
FB	Fluidized bed
FC	Flow controller
FI	Flow measurement
LC	Level controller
MNOK	Million Norwegian kroner
MV	Manipulated variable
MW_{th}	Thermal output
Nm^3/s	Normal cubic meter per second
NOK	Norwegian kroner
PC	Pressure controller
SAP	Steam Air preheater
SH	Superheater
TC	Temperature controller
TPM	Throughput manipulator
Vol%	Volume percentage

1 Introduction

Norske Skog Skogn is the largest producer of newsprint in Norway, and one of the largest producers in Europe. They have three paper machines with a combined production capacity of around 590 000 tons. The paper mill exports most of its paper to Europe, with the UK as its main market. [1]

Figure 1.1: Norske Skog's paper mill at Skogn. [1]

The paper machines require heat in the form of steam, and apart from heat recovery from the earlier stages of the process, the paper mill has 3 potential steam producers: An electric boiler, a bark fired boiler (Boiler5), and a circulating fluidized bed (CFB) boiler (Boiler6). The CFB boiler is the main steam producer.

The CFB technology is currently widely used in coal fired power plants. The technology has certain advantages compared to other technologies in terms of emissions, and it has therefore, because of recent years focus on air pollution and CO2-emissions, gained increase attention as a "cleaner" alternative to other technologies used in coal fired power plants. [3]

Although the main application for the CFB technology traditionally has been coal fired power plants, it is also implemented successfully, and increasingly, for other applications such as biomass and waste fired power plants. [4]

The CFB at Norske Skog (Boiler6) is a Compact CFB delivered from Foster Wheeler in 1997. It has a rated capacity of 50 MW_{th}, although it is currently constrained below this capacity. Having Boiler6, the main steam provider at Norske Skog Skogn, constrained below rated capacity is not an ideal situation. Especially taking into account that this boiler generally burns the fuel with the lowest cost. It is therefore desirable from the company's side to look into ways to quantify the potential losses, as well as investigating possible improvements to the existing operation.

1.1 Scope of work

The main objectives of this thesis is therefore first to get an overview of Boiler6; how it works, how it is operated, what the active constraints are, etc. This is to be achieved through visits to the paper mill, investigation of the boiler and its manuals, conversations with the engineers and operators, and investigations of the control system. The second objective is to identify possible improvements to how it is operated and try to quantify the economic loss resulting from having the boiler operating below its rated capacity.

2 Process description

This section gives a description of both the CFB technology in general, and the spesific CFB boiler at Norske Skog Skogn.

2.1 Steam production at Norske Skog

The steam network at Norske Skog supplies steam to the the paper machines, as well as a range of other applications such as heating of water, buildings and other machines. Apart from heat recovery in the refining process, the paper mill has three different steam suppliers: One electric boiler (Boiler4, 43 MW), one bark-fired boiler (Boiler5, 35 MW), and one biomass fired CFB boiler (Boiler6, 50 MW).

Because of the high cost of using electricity to produces thermal energy, the electric boiler is used as little as possible.

The bark-fired boiler burns bark that originates from debarking the wood logs prior to pulp production. When the paper mill uses a lot of virgin fiber (wood logs) in the paper production, bark is in plentiful supply, and it is relatively cheap to use the bark boiler. However, when the paper mill uses a lot of recycled paper, the amount of bark produced at the paper mill is low, and the company is forced to buy bark from other suppliers. This is more expensive, and the cost of using the bark boiler increases accordingly.

The biomass-fired CFB (Boiler6), which is the main focus of this thesis, burns a range of different fuels, including bio sludge, bark, demolition wood, oil (mainly for start-up) and reject from the deinked pulp plant mainly consisting of various plastics and foreign materials. The sludge burned in the CFB boiler are either reject from the deinked pulp plant, or sludge from the waste water treatment plant at the site. Demolition wood is supplied from external sources and has a very low cost compared to electricity and, if the plant uses a lot of recycled paper, bark. If the plant use a lot of virgin fiber (wood logs), the cost of bark can almost be regarded as negative because the company has to pay to get rid of it if they do not burn it themselves. Anyways, the CFB boiler can, as mentioned, burn a range of different fuels, including bark. It is therefore usually desirable to use the CFB as much as possible at the expense of the other two boilers.

It is also worth mentioning that there is supposed be a steam turbine connected to the steam production of Boiler6. This turbine is used to generate electricity when there is spare steam production capacity in Boiler6. This turbine is however currently out of order and will not be a part of the investigations in this thesis.

2.2 The CFB technology

The *criculating fluidized bed*-technology (or CFB) can be regarded as a special case of the general *fluidized bed*-technology (or FB). FB is a phenomena that occurs when a fluid is passed through a bed of solid particles and the drag from the fluid offsets the gravitational pull on the particles. This makes the particles "float", and the solid/fluid-mixture is said to be a FB (Figure 2.1).

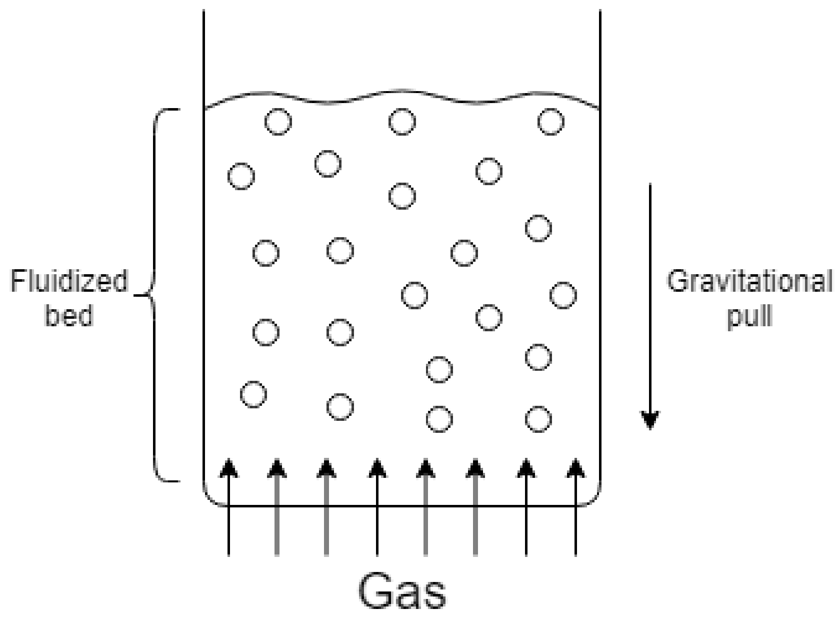

Figure 2.1: Fluidized bed.

The FB behaves in much the same ways as a liquid. For example: [3]

- The static pressure at any point is roughly equal to the weight of the particles per cross section above that point.

- Objects will fall or rise in the FB depending on their relative density to the FB around them.

- The particles are well mixed and the FB maintains a nearly uniform temperature when it is heated from any side.

The fluidized bed phenomena allows for a high contact surface between the fluid and the particles, and thus a higher reaction rate.

If the fluid flow through the particles is increased above the terminal velocity of the particles and the particles are carried away with the fluid, it is called a *fast fluidization*. If this fluidized flow is recycled back to the bottom of the combustion chamber, the setup is called a *circulating fluidized bed* (CFB).

Figure 2.2 gives a drawing of a simple CFB boiler to show the main concept of the technology. The primary air enters through the bottom of the combustion chamber, while the secondary air enters at several places further up in the combustion chamber. This is done to make sure that there is a mostly uniform distribution of oxygen throughout the bed. The fuel is fed into the combustion chamber near the bottom of the bed. The gas and solid particles are separated in the cyclone so that the solid particles are returned to the combustion chamber.

In addition to the solid fuel, there are also solid bed materials circulating in the CFB. This might be limestone if the fuel has a high sulfur content that needs to be removed, or it might be an inert material such as sand if the the fuel has a low sulfur and ash content (The boiler investigated in this thesis uses sand). The reason for having inert bed materials in the CFB is that they increase the thermal inertia of the CFB and contributes to the mixing of fuel and air. The CFB has a very good heat distribution, and the temperature is nearly constant (bounded between 800-900°C) throughout the combustion chamber. [3]

The relatively low temperature (800-900°C), large combustion zone and high bed velocity gives the CFB technology a range of advantages over competing technologies. The major ones are:

- The good mixing achieved by having a fast fluidization along with the thermal inertia provided by the bed material makes it possible to burn a wide range of fuels.

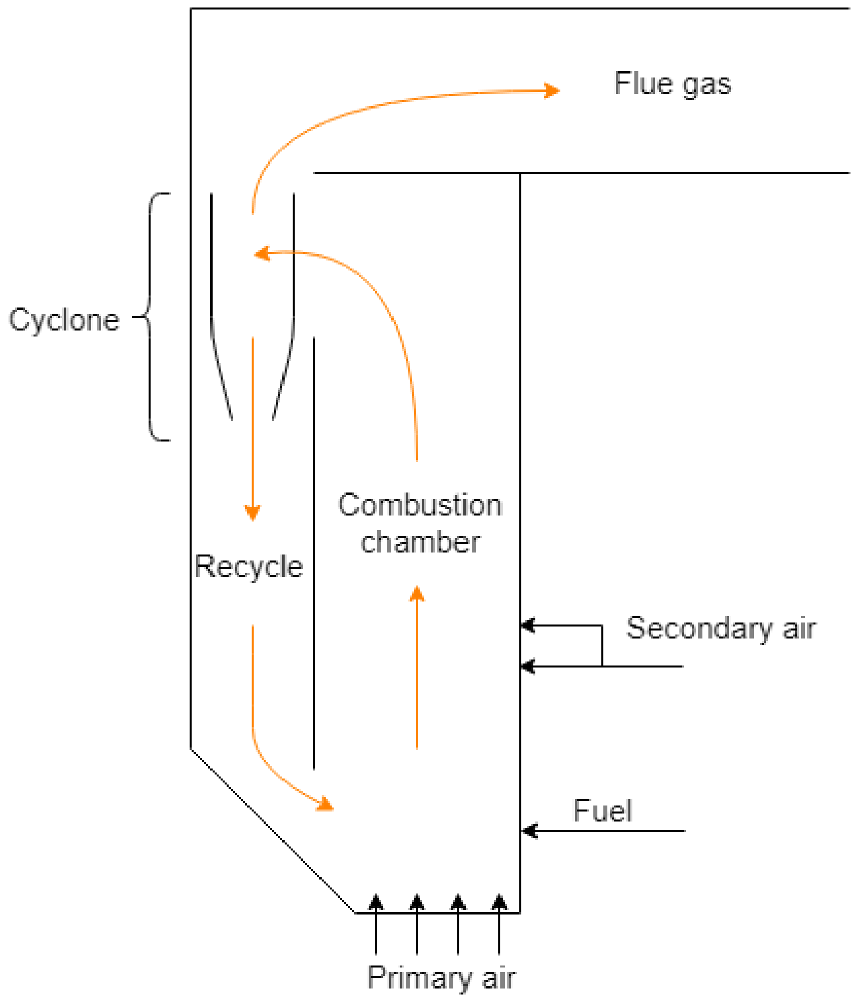

Figure 2.2: Simple example of a CFB boiler where solid fuel is burned with air. The orange arrows show what the main movements through the boiler are.

- The large zone of combustion makes it easier to make more compact boiler designs, because the combustion practically takes place in the entire combustion chamber, recycle and cyclone.
- The low temperature and somewhat uniform distribution of oxygen (because of the secondary air) gives the N_2 in the air relatively little chance to react with the O_2, something that results in very low NO_x-emissions compared to other technologies that operate with higher peak temperatures.
- If the fuel has a high sulfur content, the CFB-technology provides a cheap method for sulfur removal by simply letting limestone circulate with the fuel.

2.3 The CFB boiler at Norske Skog

The CFB boiler at Norske Skog Skogn is a 50 MW Compact CFB. The Compact name stems from the design of the cyclones and integration of applicable surrounding equipment within the boiler construction. This essentially takes up less space compared to a traditional CFB boiler construction. It was delivered in 1997 by Foster Wheeler, one of the leading suppliers of CFB boilers worldwide at the time. Foster Wheeler's CFB business is now a part of Sumitomo SHI FW. Figure 2.3 gives an idea of the physical size and proportions of the boiler. It shows the CFB boiler along with the economizer and air preheater (described in Section 2.3.5)

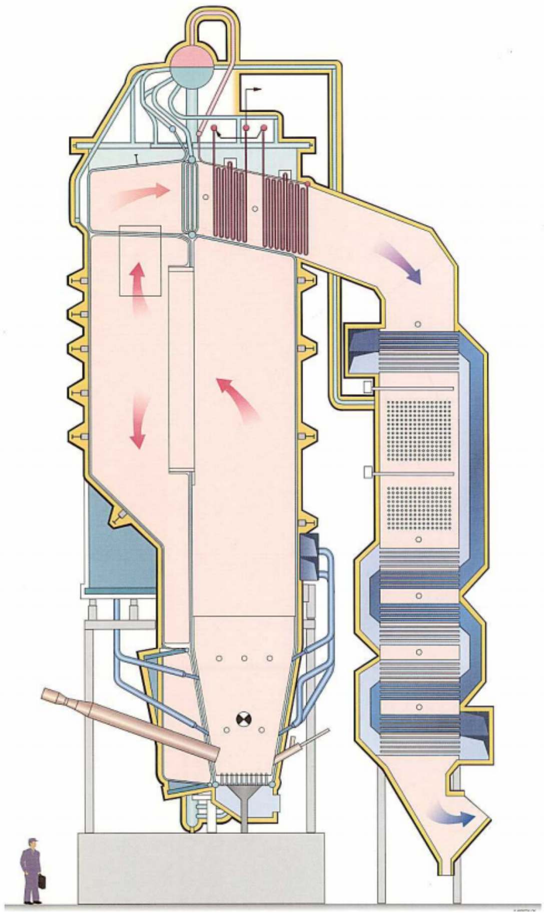

Figure 2.3: Drawing of the CFB boiler (including economizer and air preheater) at Norske Skog Skogn. [2]

A more schematic representation of the boiler with names of the most important components, as well as the flows of steam, water, air and flue gas, is given in Figure 2.4.

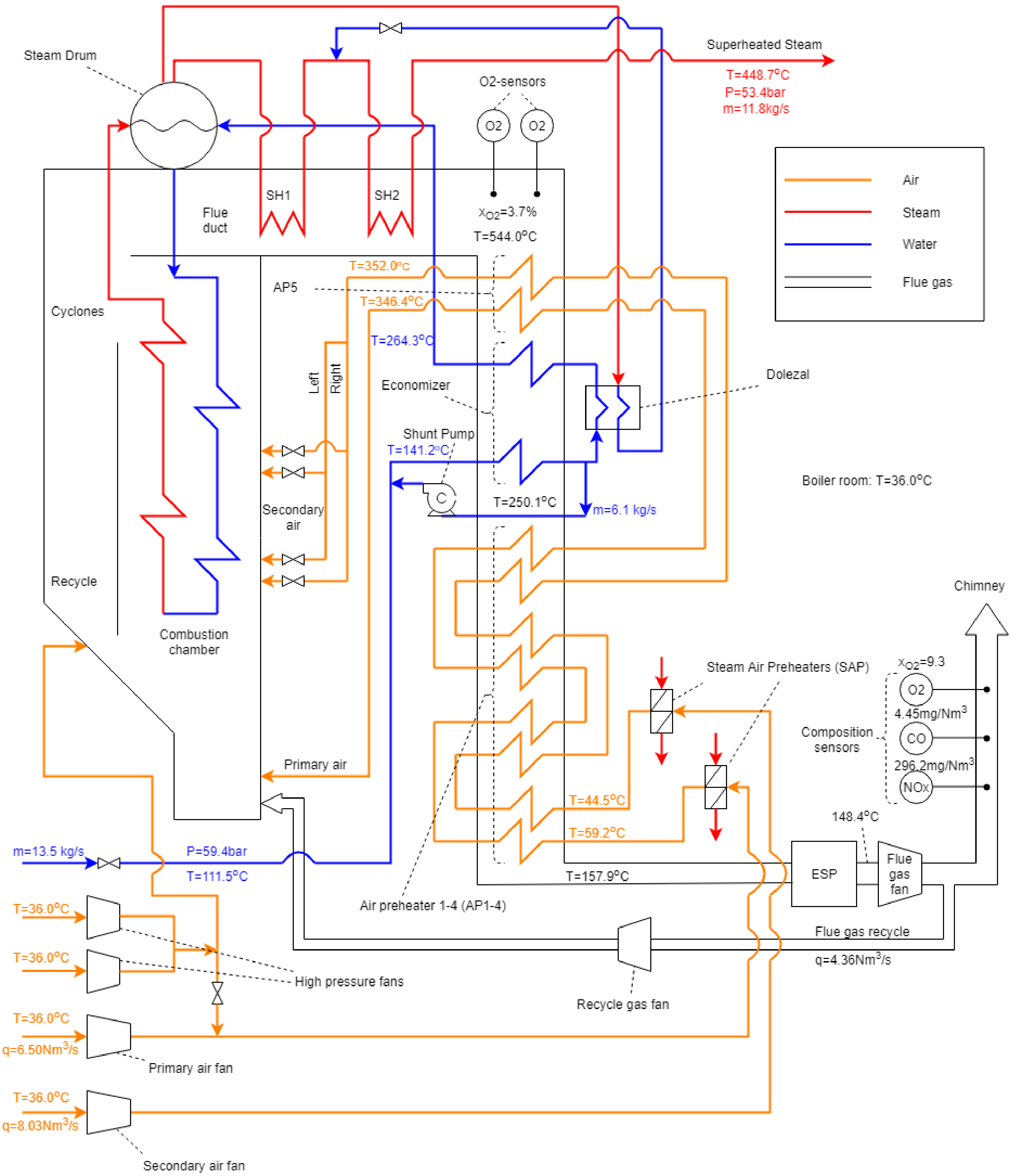

Figure 2.4: Simplified schematic representation of Boiler6.

2.3.1 Fuel

The fuel can either be various solid fuels or it can be fuel oil.

The fuel oil is mainly used for start ups, because the boiler needs some preheating before it can burn solid fuel. This is because the ignition of the solid fuel in a CFB happens by simply letting it come in contact with the already hot fluidized bed particles. The fuel oil enters the combustion chamber through oil lances. It has a high sulfur content compared to the solid fuels used. This means that there is an increased risk of sulfurous acid (H_2SO_3) condensing on the walls in the flue gas system. This can have some consequences for how the shunt pump should be operated (see Section 2.3.5.3).

During normal operation, the CFB usually burns a range of different solid fuels. This can be bark, demolition wood, biosludge from the waste water treatment, or other types of sludge derived from waste from the paper production. The wide range of different fuels, uneven mixing of the fuel before it enters the combustion chamber, and no good way of estimating the composition of the fuel, makes it challenging to operate the boiler so that the emissions are kept below their limits.

The solid fuel enters the combustion chamber near the base through two separate screws that are situated on each side of the combustion chamber. As the fuel sometimes is unevenly mixed, the relative speed of the screws must sometimes be adjusted so that fuel and oxygen are evenly mixed in the bed (see Section 4.4).

2.3.2 Water/steam cycle

The feed water comes in at around 60bar and 112°C. It is preheated to around 264°C by exchanging heat with the flue gas in the economizer. The econonmizer is split into two sections, ECO1 and ECO2. ECO1 is the first section the feed water enters. Between the two economizer sections, the water goes through the Dolezal heat exchanger. This is further explained in Section 2.3.5.1. There is also a shunt pump that drives a recycle loop around ECO1. This is further discussed in Section 2.3.5.3.

After the economizer, the water enters the steam drum. From the steam drum, water falls from the bottom of the drum down through evaporator tubes that goes along the combustion chamber. Here, the water evaporates and becomes steam. The steam is returned to the steam drum (see Figure 2.4). From the top of the steam drum, saturated steam exits and flows into the superheaters (SH1 and SH2). Here, the steam is heated above its dew point. There is some water that enters the superheated steam through an attemperator between SH1 and SH2. This is saturated steam from the steam drum that has be condensed in the Dolezal (see Section 2.3.5.1). This water makes sure the steam enters the steam network with the desired temperature of 450°C (see Section 4.3).

2.3.3 Flue gas

The flue gas forms in the combustion chamber, comes out at the top of the cyclones and enters the flue gas channel. There it exchanges heat with the superheaters (SH1 and SH2), before it enters the ECO/AP where it exchanges heat with the the feed water and primary and secondary air. After all this it exits the ECO/AP with an average temperature of around 158°C. It is desirable to keep the temperature of the flue gas above 130°C to reduce the risk of sulfourus acid (H_2SO_3) condensing on the walls of the flue gas channel. According to the design, this is controlled by the steam air preheaters (see Section 2.3.5.4), although they are not in use because of leaks (see Section 2.3.6). After the ECO/AP, the flue gas goes through the ESP where particles are removed(see Section 2.3.5.5). After that it goes through the flue gas fan, which is the fan that propels the flue gas through the system.

After the flue gas fan, the stream splits into an exit stream and a flue gas recycle stream. The flue gas recycle stream recycles some of the flue gas back to the combustion chamber. It is propelled by the flue gas recycle fan. The reason for having this recycle, is because this relatively cool flue gas (around 150°C) can be used to control the bed temperature (around 860°C) and make sure that it does not become too high.

The flue gas that does not enter the recycle passes an analyzer where environmental pollutants (such as CO and NO_x) are measured. It finally exits through the chimney.

2.3.4 Air

Air is fed into the combustion chamber as either primary air, secondary air or tertiary air (high pressure air).

The primary air is supplied from the boiler room by the primary air fan. It is preheated to around 346°C by exchanging heat with the flue gas. This is done in the air preheater as shown in Figure 2.4. It finally enters the combustion chamber at its base where it is responsible for fluidizing the solid fuel.

The secondary air is supplied from the boiler room by the secondary air fan. It is also preheated in the air preheater by exchanging heat with the flue gas. After the air preheater it is split up into four different streams, each with its own control valve (see Figure 4.1 in Section 4.2). The four streams represent an upper and a lower level with a left and a right hand side for both levels. The four streams are then split up further, so the secondary air finally enters the combustion chamber through a total of 22 separate openings. The number of openings used to be 32, but ten openings have been permanently shut down due to uneven distribution of oxygen. The reason for having the secondary air is to get a more even distribution of oxygen in the combustion chamber.

The flow instruments used to measure the primary air and the different inlets of the secondary air are not dimensioned for this kind of use. The reason for this is probably some kind of miss that happened when the boiler was constructed. This means that there might be some unknown uncertainties in the air flow measurements.

The third way air is supplied to the combustion chamber is through the two high pressure fans. They are running at a constant speed, and the air is used for the "recycle knee", where the recycled material enters the combustion chamber again. The "recycle knee" is a tight opening, which means that there is a risk of having the recycled material blocking this opening. The high pressure air is used to blow away sand so that the opening stays open. The high pressure air that is not used enters the primary air through a valve (see Figure 2.4).

Apart from these three intended ways of adding air to the combustion chamber, there are also some unintended leaks that causes air to leak into the system at various places. This is further explained in Section 2.3.6.

2.3.5 Components

This subsection contains a list of the most significant components of the CFB. All process variables listed are averages over a period of two weeks(see Section 5) and are included to give an idea of typical operating conditions. Some of the components are subjected to leaks, something that is further described in Section 2.3.6.

2.3.5.1 Dolezal

The Dolezal is the name of the heat exchanger situated next to the economizer (Figure 2.4). The purpose of the exchanger is to condense saturated steam from the steam drum so that it can be used to control the temperature of the superheated steam. The feed water between Economizer 1 and 2 makes up the cold side of the exchanger. The hot side of the exchanger is made up of saturated steam that is condensed to water. The two main challenges with this kind of setup for temperature control of the superheated steam temperature are:

- Because there is no apparatus to drive the steam/water from the steam drum, through the Dolezal, and into the superheated steam after Superheater1, the only driving force is the the pressure difference between the steam drum and the steam after Superheater1. This ΔP is around 1 bar. Because of the low ΔP, it can be difficult to achieve turbulent flow through the attemperator, something that affects the distribution of the cooling water. Low turbulence lowers the atomization degree of steam which in turn leads to unequal attemperation and potentially phase separation (i.e., cooling water and steam co-existing over a distance before fully mixing). This increases wear on piping and could lead to corrosion problems.
- Because the cold side of the Dolezal is the feed water between Economizer 1 and 2, the temperature might be too high to achieve sufficient cooling. During normal operation, this cold side temperature is usually around 207 °C (This is a calculated value, see Section 5.3)

2.3.5.2 Economizer and Air Preheater

The Economizer(ECO) and the Air preheater(AP) are used to recover the heat of the flue gas after it has passed the superheaters. The different parts of the ECO/AP are, sorted from high to low flue gas temperature, AP5, ECO2, ECO1, AP4, AP3, AP2 and AP1. Primary and Secondary air is preheated in the AP from around 59 and 45°C to around 346 and 352°C. The feed water is preheated in the ECO from around 112°C to around 264°C.

2.3.5.3 Shunt pump

The shunt pump is used to pump the water recycle from after ECO1 to before ECO1 (see Figure 2.4). The water recycle has a higher temperature than the feed water, and thus by recycling the water, the water inlet temperature of ECO1 increases.

When the inlet temperature of water increases, the temperature difference over the heat exchanger decreases, something that leads to less heat transferred from the flue gas to the water. This means that having the shunt pump turned on leads to less heat recovered from the flue gas, something that leads to an energy loss if this heat is not recovered somewhere else.

There is also another reason for having the shunt pump running. Because the economizer is a heat exchanger with a liquid (water) on one side and a gas (flue gas) on the other side, the heat exchanger tubes will have a temperature much closer to that of the liquid than the temperature of the gas. [5] This means that the tubes in the economizer will have a temperature much closer to the water, which without the shunt pump would have a temperature around 112°C. If the flue gas has a high SO_2 content, it means that there is an increased risk of sulfurous acid (H_2SO_3) condensing and causing corrosion on the tubes. The instruction manual gives a temperature limit of 130°C which the flue gas should not fall below. Using the shunt pump at maximum capacity gives a water inlet of around 141°C.

The shunt pump is further investigated in Section 3.3.

2.3.5.4 Steam Air preheater

The steam air preheaters purpose is to make sure that the temperature of the flue gas does not drop below 130°C. By staying above this temperature the risk of corrosive acids (such as H_2SO_3) condensating on the walls of the flue gas channel is minimized. The way they are supposed to function is that they heat the primary and secondary air before it enters the AP. This causes a smaller temperature difference across the AP, which results in less heat transferred from the flue gas and the temperature in it remains higher.

The SAPs are currently leaking, so they are not used during normal operation. They are usually not necessary either, because the flue gas temperature is usually above 130°C.

2.3.5.5 Electrostatic precipitator (ESP)

The ESP is responsible for removing particulate matter from the flue gas. It functions by using high voltage to ionize the flue gas and its particulate matter. The ionized particles are directed towards charged ground plates, and thus are removed from the flue gas.

2.3.5.6 Flue gas fan

The flue gas fan is a speed controlled centrifugal fan that is situated after the ESP, but before the flue gas recycle split (see Figure 2.4). It has a power of 400 kW and a capacity of 29.3 Nm^3/s. The purpose of the flue gas fan is to make sure that the pressure inside the combustion is lower than outside, so that flue gas in the boiler room is avoided.

The flue gas fan is an active constraint for the steam production. This is mainly due to leaks at various points throughout the flue gas system. This is further explained in Section 2.3.6

2.3.5.7 Primary air fan

The primary air fan is a 400 kW centrifugal fan with a capacity of 9.6 Nm^3/s. It drives the primary air into the system. It is not speed regulated, and the primary air flow is instead controlled by a valve (see Figure 4.1 in Section 4.2).

2.3.5.8 Secondary air fan

The secondary air fan is is a 200 kW speed controlled centrifugal fan with a capacity of 10.5 Nm^3/s. In maintains the pressure of the secondary air. The flow of secondary air is instead controlled by the four different valves that distribute the secondary air into the different parts of the combustion chamber. The distribution of secondary air is further explained in Section 2.3.4.

2.3.5.9 Flue gas recycle fan

The flue gas recycle fan is a 288 kW speed controlled centrifugal fan with a capacity of 10.0 Nm^3/s. It drives the flue gas recycle stream and, because this stream has a much lower temperature than the combustion chamber bed, it regulates the temperature of the bed and makes sure that it does not become too hot.

2.3.5.10 High pressure fans

The high pressure fans runs at constant speed and supplies tertiary air to the combustion chamber. The tertiary air is used to keep the recycle knee open as explained in Section 2.3.4.

2.3.6 Leaks

Figure 2.5 gives a representation of where air leaks into the flue gas system. Notice that there are leaks from both the primary and secondary air into the flue gas channel.

The leaks lead to range of problems:

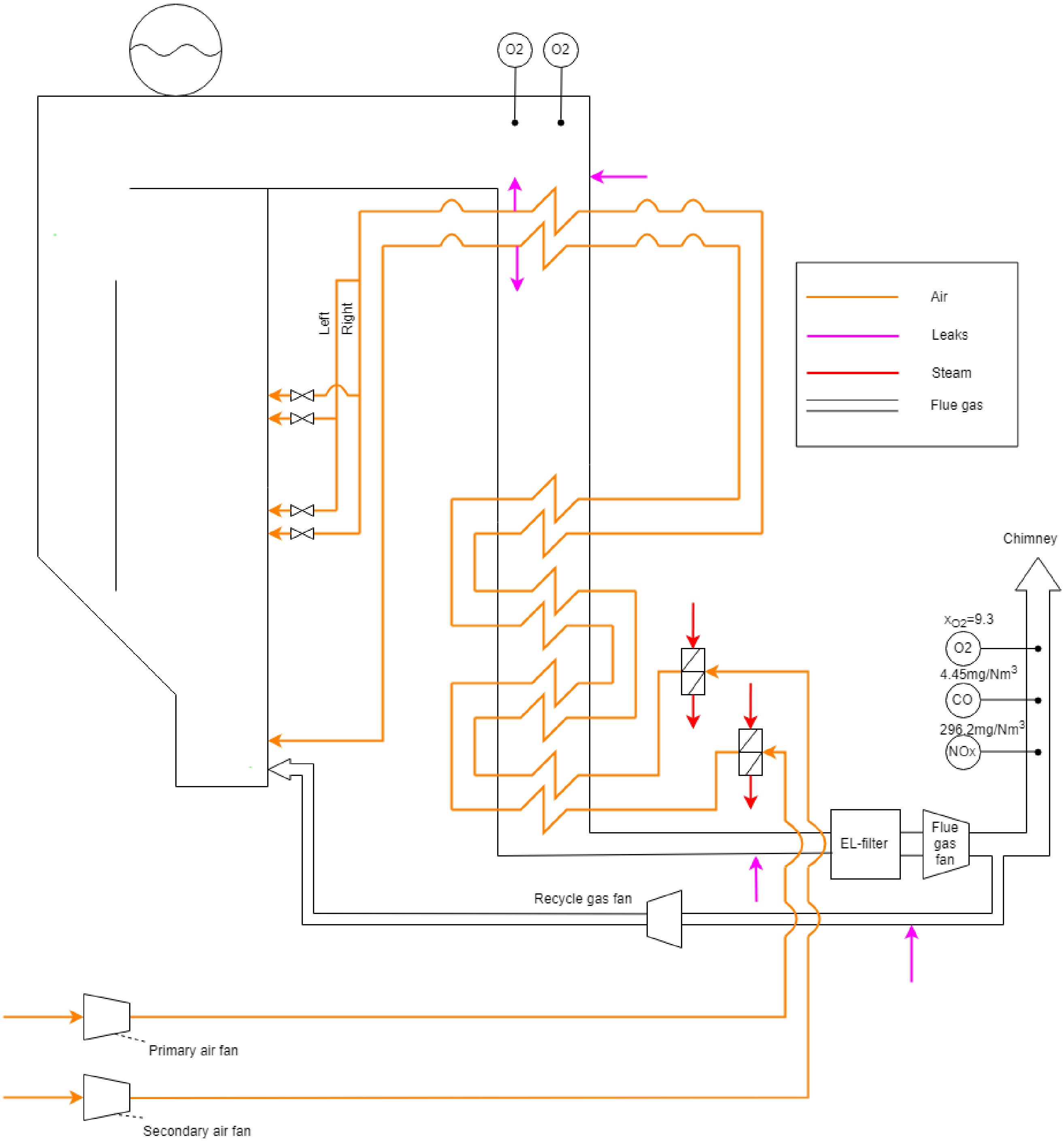

Figure 2.5: Schematic representation of where air leaks into the flue gas system.

- Perhaps the most important effect of the air leaking into the flue gas system is the increased mass flow through the flue gas fan, something that forces the fan to operate at maximum capacity. This is considered the main reason why steam production is limited below rated capacity. The amount of air leaking into the flue gas between the first O2-measurment and the flue gas fan is estimated to constitute roughly 32% of the resulting mix of air and flue gas (calculated in Section 5.1). This means that the air leaking in represents an increase of almost 50% of the total amount of gas that has to pass through the flue gas fan.

- Another effect is the fact that the air leaking in has a temperature of around 36°C, and thus ends up cooling down the flue gas. This lowers the temperature of the flue gas and reduces the amount of heat that can be recovered in the ECO/AP.

- Air leaking into the flue gas also causes the oxygen level in the flue gas recycle stream to be considerably higher than it should be. The level is >9.3vol%, compared to 3.72vol% (which is the O_2-level before the ECO/AP) if there had been no leaks. The recycle is used to cool down and control the temperature of the bed. Any oxygen in this stream is therefore undesired because extra oxygen leads to increased combustion and thus extra heat, rendering the recycle stream less effective as a cooling mechanism. To compensate for this, the recycle stream has to be larger, something that leads to an increased load on both the flue gas recycle fan and the flue gas fan. This adds to the problem of having the flue gas fan as an upper constraint for the steam production.

- There is also a leakage in the steam preheaters (not shown in Figure 2.5). This is not a contributor to air leaking into the flue gas, but rather a leakage of steam into the primary and secondary air streams and the boiler room. This leakage renders the steam preheaters useless, and they are therefore not used during normal operation. This means that the flue gas temperature after the economizer and air preheater is not regulated by the steam preheaters as intended in the design (Section 2.3.5.4)

- Corrosion in the AP have prompted the company to close down some of the tubes in the heat exchangers. This has reduced the heat exchanger surface and hence the heat recovery of the system. If the leaks were to be fixed, the heat recovery (and boiler efficiency) would increase. Another problem arising because some of the heat exchanger tubes are closed down, is that the overall resistance in the heat exchanger is increased. This resistance makes some of the secondary air valves go into maximum position, reducing the degrees of freedom available to control the distribution of secondary air.

3 | Mass and Energy Balances

This section describes how the equations used in Section 5 are derived and what the assumptions behind them are.

3.1 Mass balance of O_2

The vol% of O_2 in the flue gas is measured both before the ECO/AP and after the flue gas fan (Figure 2.5). With this information it is possible to estimate the amount of air leaking into the flue gas between these two measurements as a fraction of the resulting mixture of air and flue gas. This can be done by creating both a steady-state mass balance of the total amount of gas (Equation 3.1.1) and a steady-state mass balance of the amount of oxygen (Equation 3.1.2):

$$\dot{n}_{Mix} = \dot{n}_{Air} + \dot{n}_{Flue} \tag{3.1.1}$$

$$\dot{n}_{Mix} \cdot x_{O_2,Mix} = \dot{n}_{Air} \cdot x_{O_2,Air} + \dot{n}_{Flue} \cdot x_{O_2,Flue} \tag{3.1.2}$$

$\dot{n}_{O_2,Air}$ is the molar flow of the air leaking into the flue gas, $\dot{n}_{O_2,Flue}$ is the molar flow of the flue gas before the ECO/AP, and $\dot{n}_{O_2,Mix}$ is the molar flow of the mixture of air and flue directly after the flue gas fan. x_{O_2} represent the corresponding molar fractions of oxygen.

Combining Equation 3.1.1 and Equation 3.1.2 and rearranging gives a formula for the mole fraction of air in the mixture (Equation 3.1.3):

$$x_{Air} = \frac{\dot{n}_{Air}}{\dot{n}_{Mix}} = \frac{x_{O_2,Mix} - x_{O_2,Flue}}{x_{O_2,Air} - x_{O_2,Flue}} \tag{3.1.3}$$

To use this equation some assumptions have to be made:

- It is assumed that the gas is perfectly mixed at both points of measurement.
- It is assumed that there are no reactions (i.e. no combustion) taking place between the two points of measurement.
- The O_2-sensors measure the O_2-concentration as a volume fraction of O_2 in dry air. The water content of the gases is therefore not taken into account, and it is also assumed that the volume fractions that the sensors measure are equal to the molar fractions.
- The molar fraction of O_2 in air is assumed to be 0.21.

If the volume flow, q_{Mix}, mass density, ρ_{Mix}, and molar mass, M_{Mix}, of the mixture of air and flue gas are known, the absolute molar flow of the air leaking in can be found using Equation 3.1.4:

$$\dot{n}_{Air} = x_{Air} * \frac{q_{Mix} * \rho_{Mix}}{M_{Mix}} \tag{3.1.4}$$

3.2 Heat loss from cold air leaking in

As mentioned in Section 2.3.6, there are numerous problems arising as a consequence of the fact that air is leaking into the flue gas. One of these is the fact that the air leaking in has a much lower temperature than the flue gas. This means that the air cools down the flue gas, and hence reduces the amount of heat that can be recovered in the ECO/AP.

It is difficult to estimate how much this cooling affects the amount of heat recovered in the ECO/AP. One approach to estimate this is to first calculate heat transfer coefficients (UAs) for the different parts of the ECO/AP using current temperature and flow data. If these heat transfer coefficients are assumed to be constant, it is then possible to calculate how changes in temperature and flow of the flue gas affects the heat transfer. This might seem like a straightforward approach, but the somewhat complicated setup of the ECO/AP, few temperature measurements, and possible leakage from the primary and secondary air into the flue gas, makes it impossible to use this approach.

In order to still get an idea of the energy loss, a simpler approach can be chosen instead. It creates an upper estimate of the loss in heat transfer. The method is simply to calculate the energy needed to heat the air from the temperature of the boiler room (c. 36°C) to the temperature of the flue gas before the flue gas fan (c. 148°C). Estimating the energy loss this way assumes that all of the energy used to heat the air leaking in could be recovered in the ECO/AP if the leaks were to be fixed. This is probably an assumption that is too harsh, but the method still creates an upper bound for the energy loss, and thus gives an idea of what order of magnitude the loss is.

In order to calculate the energy needed to heat the air leaking in, an energy balance needs to be constructed. Equation 3.2.1 shows a simplified version where the heat capacity is assumed to be constant:

$$\dot{Q}_{Heat,Air} = \dot{n}_{Air} * Cp_{Air} * (T_{Mix} - T_{Room}) \tag{3.2.1}$$

where $\dot{Q}_{Heat,Air}$ is the energy needed to heat the air, $\dot{n}_{Air}$ is the molar flow of air and T_{Mix} and T_{Room} is the temperatures right before the flue gas fan and in the boiler room respectively. Cp_{Air} is the molar heat capacity of air and is assumed to be constant and equal to the average of the molar heat capacity of air at 36°C and 148°C (Equation 3.2.2):

$$Cp_{Air} = \frac{Cp_{Air,36°C} + Cp_{Air,148°C}}{2} \tag{3.2.2}$$

In order to calculate the molar heat capacities of air at different temperatures, it is assumed that the molar heat capacity of air can be approximated as a linear combination of the molar heat capacities of the components of air (Equation 3.2.3):

$$Cp_{Air,T} = 0.21 * Cp_{O_2,T} + 0.78 * Cp_{N_2,T} + 0.01 * Cp_{Ar,T} \tag{3.2.3}$$

where $Cp_{X,T}$ is the molar heat capacity of component X at temperature T. The molar heat capacities for each component were obtained using the general formula given in Equation 3.2.4. [6]

$$\frac{Cp}{R} = a + b \cdot 10^{-3} \cdot T + c \cdot 10^{-5} \cdot T^2 + d \cdot 10^{-8} \cdot T^3 + e \cdot 10^{-11} \cdot T^4 \tag{3.2.4}$$

where Cp is the heat capacity, R is the universal gas constant, T is the temperature in kelvin, and *a*-*e* are constants specific for each component (The values for these are given in Table A.1 in Appendix A). The equation and constants are obtained from the literature. [6]

By estimating $Cp_{Air,T}$ using Equation 3.2.3, it is assumed that the air can be regarded as an ideal gas. Ideal gas law is also assumed when using Equation 3.2.1, because the air is mixed with the flue gas at the higher temperature, so any mixing effects of the air and the flue gas has to be disregarded.

(3.2.5)

3.3 Loss from having shunt pump running at full capacity

As explained in Section 2.3.5.3, the purpose of the the shunt pump is to reduce the amount of heat recovered in ECO1 (Figure 3.1).

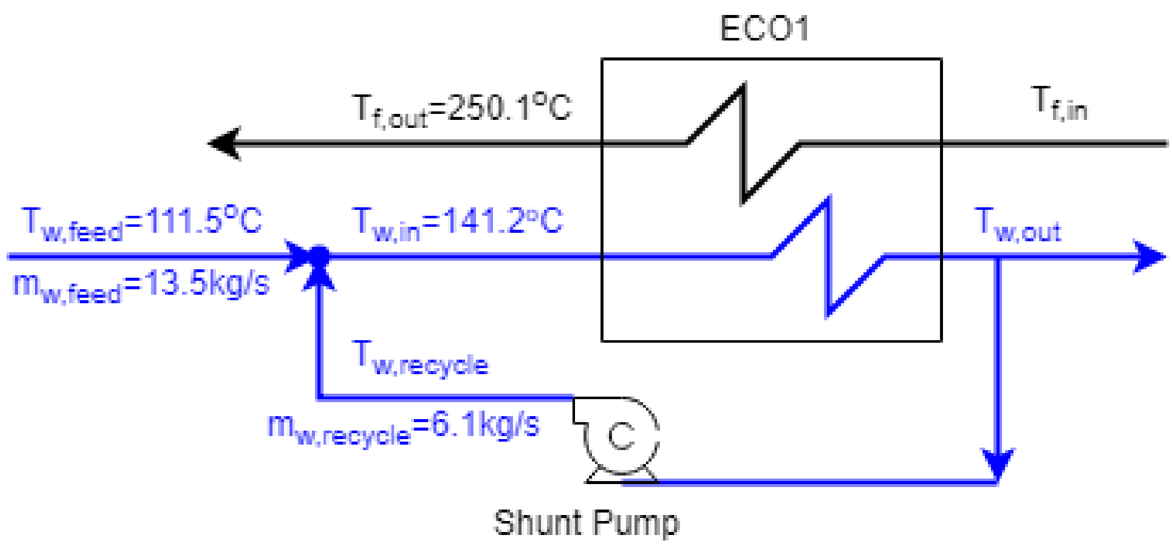

Figure 3.1: Schematic representation of ECO1 and the shunt pump. Blue colored streams refer to water, black colored streams refer to flue gas.

When the pump is running at full capacity, the amount of heat recovered from the flue gas is minimized. This might be desired under certain circumstances, such as when the flue gas temperature falls under a certain temperature (130 °C) where the risk of corrosion increases, or the fuel has a high sulfur content (see Section 2.3.5.3). Under normal operation however, the shunt pump should run at its minimum capacity so that the amount of heat recovered is maximized.

Before the work with this thesis started, the pump was running at full capacity all the time. This is exactly the opposite of how it should have been operated if the goal is to recover as much heat as possible. The reduction in heat recovery constitutes an energy loss.

It would seem that the simplest way to estimate this heat loss is to just turn off the shunt pump and see how the outlet temperature of the flue gas is affected. From the temperature change and heat capacity of the flue gas, an energy loss could then be calculated. (The logic is that turning of the shunt pump results in a lower flue gas temperature. This means that more heat must have been transferred from the flue gas to the feed water. This added energy to the feed water ultimately results in more steam produced.) In reality, the temperature change of the flue gas is so small compared to the general fluctuations of the flue gas temperature that it is not possible to easily measure it.

Instead, another approach can be chosen to estimate the energy loss. By using a few known temperatures and heat capacities, it is possible to estimate a UA-value for ECO1. If this value is assumed to be constant during changes in temperature and water flow, it is possible to get an estimate of the energy loss.

First, in order to calculate the water temperature after it has passed ECO1 ($T_{w,out}$), a steady state energy balance for the mixing of water recycle and feed water can be constructed (Equations 3.3.1).

$$\Delta\dot{H}_{w,in} = \Delta\dot{H}_{w,feed} + \Delta\dot{H}_{w,recycle} \tag{3.3.1a}$$

$$\begin{aligned} &\left(\Delta h_{F,w} + \int_{T_0}^{T_{w,in}} Cp_w dT\right) \cdot (\dot{m}_{w,recycle} + \dot{m}_{w,feed}) = \\ &\left(\Delta h_{F,w} + \int_{T_0}^{T_{w,feed}} Cp_w dT\right) \cdot \dot{m}_{w,feed} + \\ &\left(\Delta h_{F,w} + \int_{T_0}^{T_{w,recycle}} Cp_w dT\right) \cdot \dot{m}_{w,recycle} \end{aligned} \tag{3.3.1b}$$

The subscripts $w, feed$, $w, recycle$ and w, in refers to the feed water stream, the water recycle stream and the water stream entering the heat exchanger (ECO1) respectively. ΔH refers to the enthalpy of a stream, $\Delta h_{F,w}$ is the specific enthalpy of formation of water, Cp_w is the specific heat capacity, T_0 is the standard temperature and $\dot{m}$ is the mass flow of a stream.

Since the only component in Equations 3.3.1 is water, the enthalpy of formation, $\Delta h_{F,w}$, cancels out. Also, because $\int_{T_0}^{T_{w,feed}} Cp_w dT$ cancels out, T_0 can be switched out with $T_{w,feed}$. If Cp_w is assumed to be temperature independent, it also cancels out. This simplifies Equations 3.3.1 to Equation 3.3.2:

$$(T_{w,in} - T_{w,feed}) \cdot (\dot{m}_{w,recycle} + \dot{m}_{w,feed}) = (T_{w,recycle} - T_{w,feed}) \cdot \dot{m}_{w,recycle} \tag{3.3.2}$$

If the heat added to the water recycle by the shunt pump and the heat loss from the water recycle to the surroundings is assumed to be negligible, it follows from Figure 3.1 that the water temperature out of the heat exchanger, $T_{w,out}$, is equal to the temperature of the recycle, $T_{w,recycle}$. Putting this relation into Equation 3.3.2 and rearranging gives Equation 3.3.3:

$$T_{w,out} = T_{w,feed} + (T_{w,in} - T_{w,feed}) \cdot \frac{\dot{m}_{w,recycle} + \dot{m}_{w,feed}}{\dot{m}_{w,recycle}} \tag{3.3.3}$$

Next, the inlet temperature of the flue gas ($T_{f,in}$) needs to be calculated. This can be achieved by creating an energy balance over ECO1 and assuming that the heat added to the water is equal to the heat lost from the flue gas. If the specific heat capacities are assumed to be constant over the temperature range, the energy balance simplifies to Equation 3.3.4:

$$(T_{w,out} - T_{w,in}) \cdot Cp_w \cdot (\dot{m}_{w,feed} + \dot{m}_{w,recycle}) = (T_{f,in} - T_{f,out}) \cdot Cp_f \cdot \dot{m}_f \tag{3.3.4}$$

Cp_f is the specific heat capacity of the flue gas, $\dot{m}_f$ is the mass flow of the flue gas and $T_{f,out}$ is the temperature of the flue gas after ECO1.

Rearranging Equation 3.3.4 gives an expression for $T_{f,in}$ (Equation 3.3.5):

$$T_{f,in} = T_{f,out} + (T_{w,out} - T_{w,in}) \cdot \frac{Cp_w \cdot (\dot{m}_{w,feed} + \dot{m}_{w,recycle})}{Cp_f \cdot \dot{m}_f} \tag{3.3.5}$$

Knowing the temperatures and flows of both water and flue gas on both sides of ECO1 makes it possible to calculate the heat transfer coefficient (UA). This is done by assuming that ECO1 can be regarded as an ideal counter current heat exchanger and using the equation for an ideal heat exchanger in combination with the enthalpy change of water to create the following energy balance (Equation 3.3.6):

$$UA \cdot \Delta T_m = (T_{w,out} - T_{w,in}) \cdot Cp_w \cdot (\dot{m}_{w,feed} + \dot{m}_{w,recycle}) \tag{3.3.6}$$

ΔT_m is the logarithmic mean temperature difference (Equation 3.3.7):

$$\Delta T_m = \frac{(T_{f,in} - T_{w,out}) - (T_{f,out} - T_{w,out})}{ln(T_{f,in} - T_{w,out}) - ln(T_{f,out} - T_{w,out})} \tag{3.3.7}$$

If UA is assumed to be constant during flow and temperature changes, the effect of turning off the shunt pump (i.e. setting $\dot{m}_{w,recycle} = 0$) can then be estimated numerically by using equations 3.3.4 and 3.3.6 and the fact that the only two unknowns during this change are $T_{f,out}$ and $T_{w,out}$ ($T_{w,in}$ is simply equal to $T_{w,feed}$ when the shunt pump is turned off).

The results of these calculations are presented in Section 5.3.

4 | Operation and Control of Boiler6

This chapter present the current operation mode and control structures for Boiler6, with a focus on combustion control, water side control and emission control.

4.1 Operation objectives

The operation objective of Boiler6 is to produce steam according to the demand of the paper mill while staying below emission limits for CO and NO_x.

4.1.1 Throughput manipulator

The throughput manipulator (TPM) of a process plant dictates where to set the production rate. In this case, the TPM of the boiler is the desired steam production, which is decided by the steam demand of the plant. However, to increase the flexibility of the plant, and ensure enough steam for the paper mills in case of boiler trip, steam is stored in an accumulator (i.e. a buffer tank to decouple the steam production in the boiler and the demand of the paper machines). Therefore, the TPM is actually the level in the accumulator. In current operation mode, the steam production is manually controlled by adjusting the fuel flow rate setpoint.

However, the steam production is currently limited by the flue gas fan. At high loads, the flue gas fan reaches its maximum power due to air leaking into the system as described in Section 2.3.6, and it becomes the bottleneck of the process. In this scenario, the steam production has to be reduced below its rated capacity.

4.1.2 Degrees of freedom

The degrees of freedom (DOF) for operation and control of a process represent the available valves, pumps and fans that can be manipulated (therefore are also called manipulated variables (MVs)). The MVs for Boiler6 are:

MV1 Fuel

MV2 Primary air

MV3 Secondary air fan

MV4 Flue gas fan

MV5 Flue gas recycle fan

MV6 Feedwater flowrate

MV7 Attemperation flowrate

MV8 Shunt pump

MV9 Steam Air preahetrs

Note MV8 has a small capacity (Section 3.3), and MV9 is not available for the current operation mode due to damaged equipment (Section 2.3.5.4)

4.1.3 Operational constraints and active constraints

Boiler6 is operated to a series of constraints related to safety, equipment capacity and emission regulations. These are:

- High enough steam pressure and level in the steam accumulator such that the steam production does not become the bottleneck for the downstream paper mills.
- Combustion chamber pressure (e.g.$P \leq -0.1$mbar) to avoid dangerous gases such as CO leaking into the boiler room.
- Bed temperature should be $T \leq$870 °C to avoid NO_x-formation and to avoid sintering of sand and alkali that would deposit in measurement instruments, or on equipment walls.
- Exhaust gas temperature should be $T \geq 130$°C to avoid corrosion due to SO_2.
- Having the superheated steam temperature to high, or allowing large variation of the steam temperature causes high thermal stresses.
- Ratings of the flue gas and flue gas recycle fans.

- Emission levels (e.g. CO, NO_x, SO_2) have to be lower than the allowed limits.
- The pressure of the superheated steam has to be lower than the drum pressure to allow the steam to flow.
- The drum level has to be within its safety limits.

At the optimal operating point, a few of the constraints will be active (i.e. are at their minimum or maximum allowed value). To determine which constraints are active, a steady-state model of the plant can be used to solve an optimization problem. However, common engineering knowledge can also be use to analyze what constraints should be active at the optimal point, and this is what is done in this work. The active constraints are:

- The feedwater temperature should be at its minimum, to maximize the temperature difference in the economizer, which maximizes the heat transfer. It is however important to keep in mind that the tubes in the economizer have a temperature close to the water temperature (see Section 2.3.5.3). This means that the temperature of the flue gas (which must be above 130°C) creates a lower limit when the it has a high SO_2-concentration.
- Flue gas temperature should be at its minimum to maximize heat recovery.
- The combustion chamber pressure should be at its maximum to avoid energy losses related to creating vacuum in the combustion chamber.
- The Bed temperature should be at the maximum to get a good reaction rate.

The plant is however subject to disturbances, and in order to be sure that the active constraints are not exceeded during transient operation, a back-off from optimality is required.

4.1.4 Control variables

The control variables are variables that should be kept at their desired setpoint, firstly to operate the plant safetly on a short time scale, and secondly to achieve optimal operation on a large time scale. In deciding what variables should be controlled, one has to always control the active constraints, and then control the variables to keep the plant near optimal operation in case of disturbances [7]. In addition to the already mentioned active constraints, the superheated steam temperature should also be controlled to avoid large temperature variations, or high thermal stresses. Considering that the pressure in the drum is varying, the steam temperature will fluctuate accordingly, and therefore the steam temperature has to be controlled after the superheater. The control variables for Boiler6 are:

CV1 Steam pressure (and flowrate)

CV2 Drum level

CV3 Superheated steam temperature

CV4 Bed temperature

CV5 Furnance pressure

CV6 Exhaust gas temperature

CV7 Feedwater temperature

CV8 CO level in flue gas

CV9 NO_x level in flue gas

CV10 O_2 level in flue gas

The control system for Boiler6 was originally designed to control the exhaust gas temperature with the steam air preheaters (see Section 2.3.5.4). However, due to large leaks in these heat exchangers, they are not currently in operation, which leads to not having a degree of freedom to control this CV.

4.1.5 Disturbances

The main disturbances in operating Boiler6 are:

D1 Fuel composition

D2 Air leaking into the flue gas

D3 Feedwater temperature

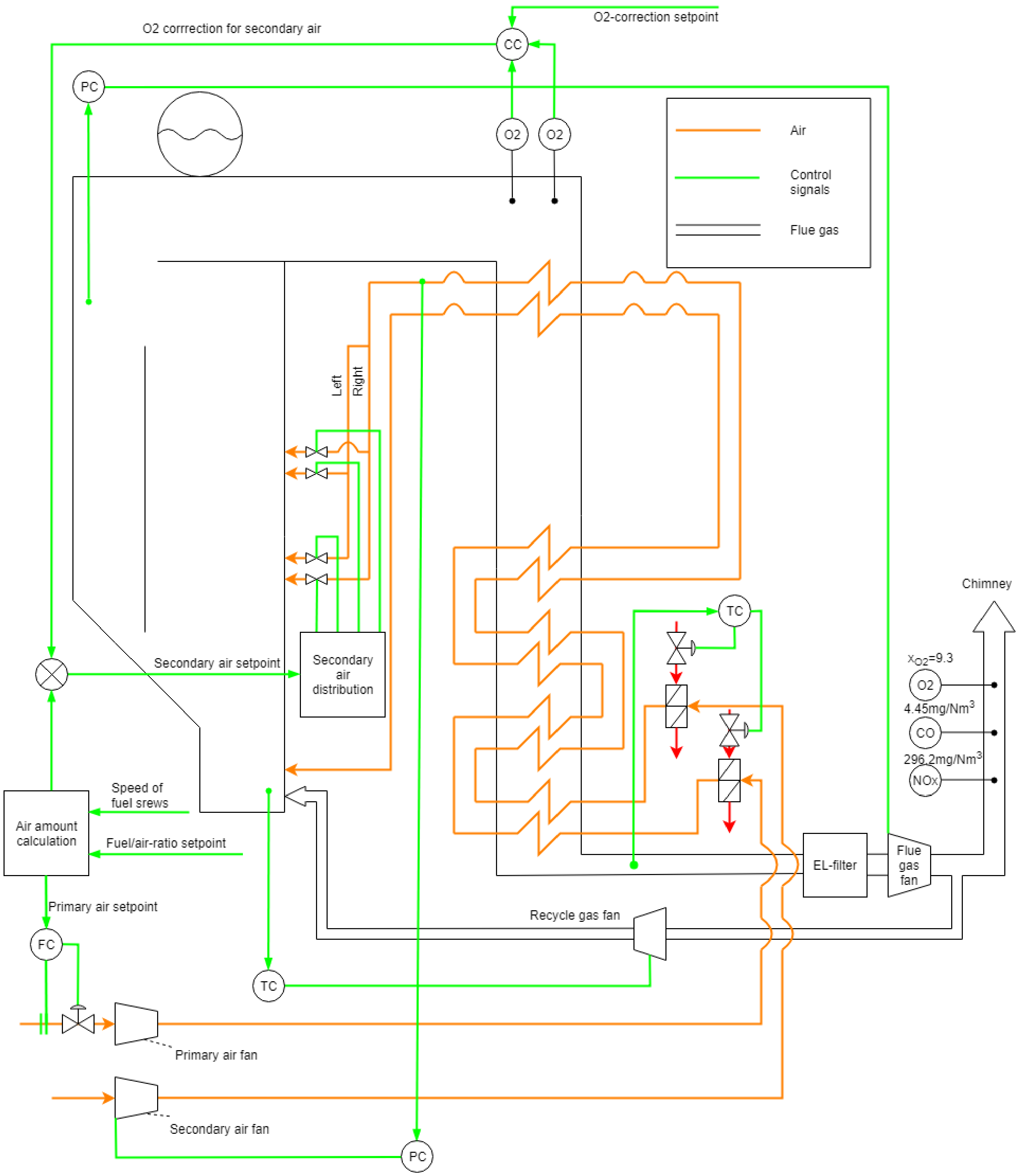

Figure 4.1: Simplified scheme of how the combustion is controlled.

4.2 Combustion Control

Figure 4.1 gives a representation of how the combustion is controlled. Note that the temperature control of the flue gas after the ECO/AP by using the steam air preheaters is included although they are not currently in use because of leaks.

As can be seen from the figure, temperature in the flue gas bed is controlled by the the flue gas recycle fan, pressure in the combustion chamber is controlled by the flue gas fan, primary air is controlled by the valve before the primary air fan, and the O_2-level in the flue gas channel is controlled by the composition controller which gives a correction factor to the amount of secondary air. The correction factor is allowed to vary between 0.6 and 1.2.

In addition to the control loops, Figure 4.1 also shows two calculation blocks. The information in these blocks are usually not changed during normal operation, and the only inputs are show in the figure. This means that the only input to the secondary air distribution is the total amount of secondary air calculated, and that the inputs to the air amount calculation is the measured speed of the two fuel screws, and a fuel/air-ratio setpoint that is used to control the emissions (see Section 4.4).

The parts that are not shown in the figure, are the fuel screws. The speed of thees are manually set by the operators so that a desired steam production is achieved. The desired steam production is based on the level in the downstream accumulators, which is the TPM for the process (see Section 4.1.1).

4.3 Water Side control

Figure 4.2 gives a simplified representation of the control loops of the water/steam side of boiler6.

The outlet temperature of the superheated steam is controlled by using the attemperator. For simplicity, this loop is shown as a single feedback loop, although it actually is a cascade loop where the TC shown in the figure gives a setpoint to an inner control loop. This inner TC measures the steam temperature between the attemperator and SH2, and adjusts the attemperator accordingly. This setup gives a faster response, but because the inner TC is situated closer to the attemperator, it might have some extra disturbances because of uneven evaporation of the water that is added to the steam.

The water inlet temperature of ECO1 is controlled by the shunt pump. When burning fuel with low sulfur content, the shunt pump should be operated at its minimum value

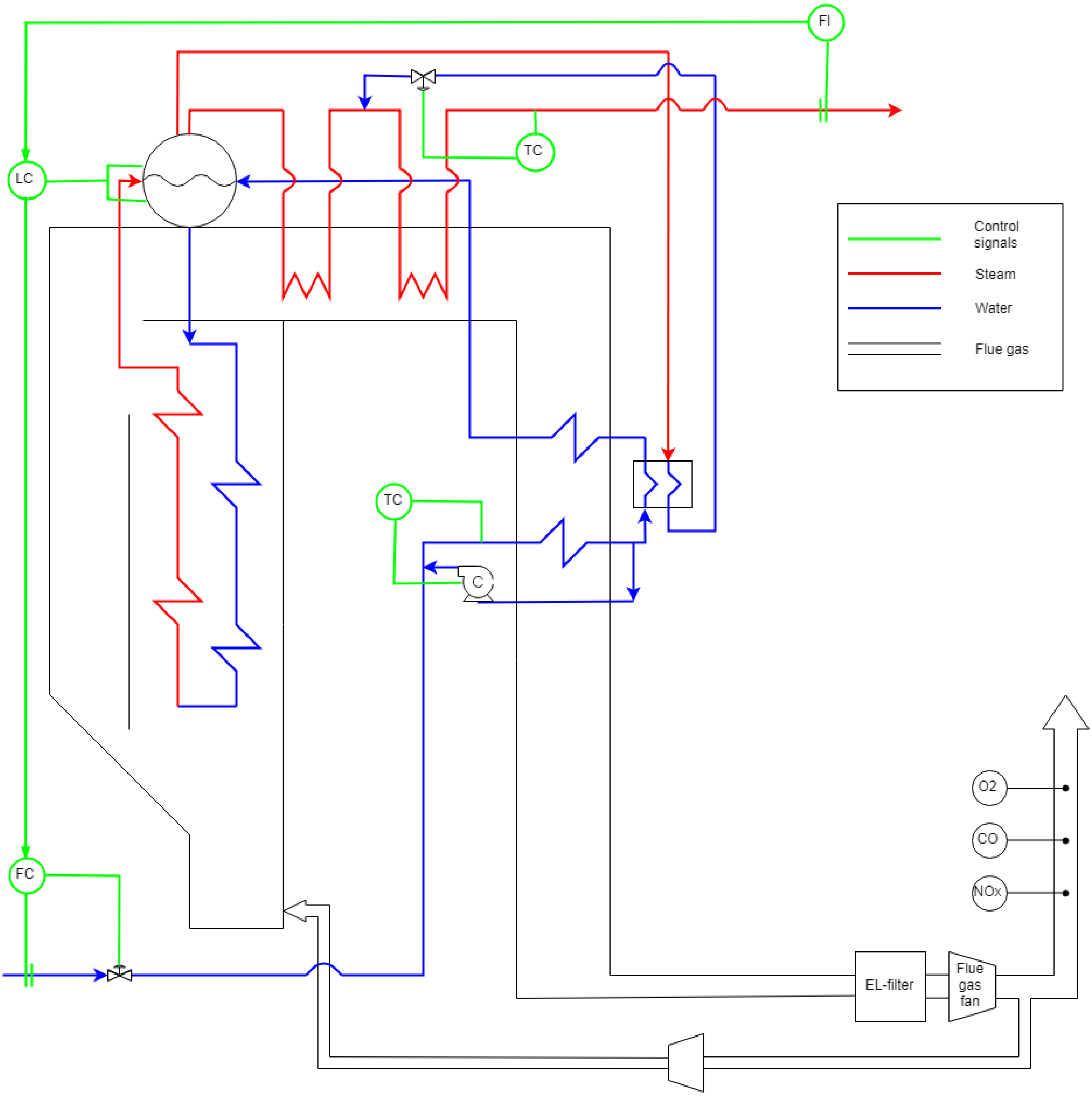

Figure 4.2: Simplified scheme of how the water/steam side is controlled.

to ensure maximum heat recovery (see Section 3.3). But if the fuel has a higher sulfur content (such as fuel oil), it might be wiser to to have a higher water inlet temperature to reduce the risk of corrosion (see Section 2.3.5.3).

The level in the drum is controlled by the feed water. The way the setup works is that the feed water set point is set equal to the superheated steam flow plus a correction term from the drum level controller. This setup is called a three element drum water level control.[8] By using the steam flow as a feedforward signal input to the control structure, dynamic phenomena such as "shrink" and "swell" have a reduced effect on the feed water regulation.

"Shrink" and "swell" are phenomena that occurs when steam flow out of the drum are either decreased or increased. Because of the pressure change in the drum that results from the change in steam outflow, the saturation temperature changes, and there is a change in evaporation rate. If the steam outflow increases, the pressure and saturation temperature drops, the evaporation (and bubble formation) increases resulting in a "swelling" of the level. "Swelling" occurs because more bubbles gives an appearance of a higher water level in the drum. With no input signal from the steam flow rate, there would be an inverse response for the water feed flow in this case. "Shrinking" describes the opposite phenomena.

4.4 Emission Control

In addition to producing the desired steam amount, it is also an objective to keep the emissions of pollutants below the limits set by the government. The two main pollutants that require continuous surveillance by the operators, are CO and NO_X. They need to be kept below limits of 75 mg/Nm^3 and 340 mg/Nm^3 respectively. The limits are average values over 24 hours, meaning that short spikes above the limits are usually okay.

CO and NO_X are usually varying inversely of each other. This is because NO_X-formation is favoured at high O_2-levels and high temperatures, while CO is favoured at lower temperatures and low O_2-levels. This means that usually, high CO-levels means low NO_X-levels and vice versa.

The main way of making sure emissions stay below limits is therefore to change the air/fuel-ratio setpoint (see Figure 4.1). If the NO_X-level is to high, it usually means that there is too much O_2 (and therefore air), while a high CO-level usually means that there is too little O_2.

If changing the air/fuel-ratio setpoint does not give the desired reduction in emissions, the emissions is usually caused by an uneven mixture of fuel and air. This can happen both because of a bad ratio between secondary and primary air, and because of an uneven mixing of the fuel that enters the combustion chamber through the two different fuel screws.

The bad ratio of primary and secondary air can be remedied by changing the setpoint for the O_2 composition controller, which changes the amount of secondary air. Because this controller only affects the amount of secondary air (Figure 4.1), it also indirectly affects the ratio between primary and secondary air. If the reason for the emissions is uneven fuel distribution, this can to some degree be remedied by changing the ratio between the speed of the two fuel screws.

The challenge by using these two approaches (when changing the air/fuel-ratio fails), is that it is no way of knowing exactly what the reason for the increased emissions is. This is therefore bound to be a process of trial and error where different ratios are changed up and down until the emission levels are reduced again.

5 | Energy and economic calculations

This section presents the results of the most important energy and economic calculations. Section 5.1, 5.2 and 5.3 are based on the mass and energy balances derived in Section 3.

Almost all calculations are done using a Matlab script which is listed in Appendix B.

The process parameters that are used in the calculations in Section 5.1, 5.2 and 5.3 are based on a data series covering a period of little over two weeks. The data series contains data for every minute throughout the time period. When the process parameters are used in the calculation, they are averaged using the Matlab function *trimmean* so that 10% of the outliers are excluded.[9] The steam production of Boiler6 during these two weeks was usually between 10 and 14 kg/s as shown by Figure 5.1.

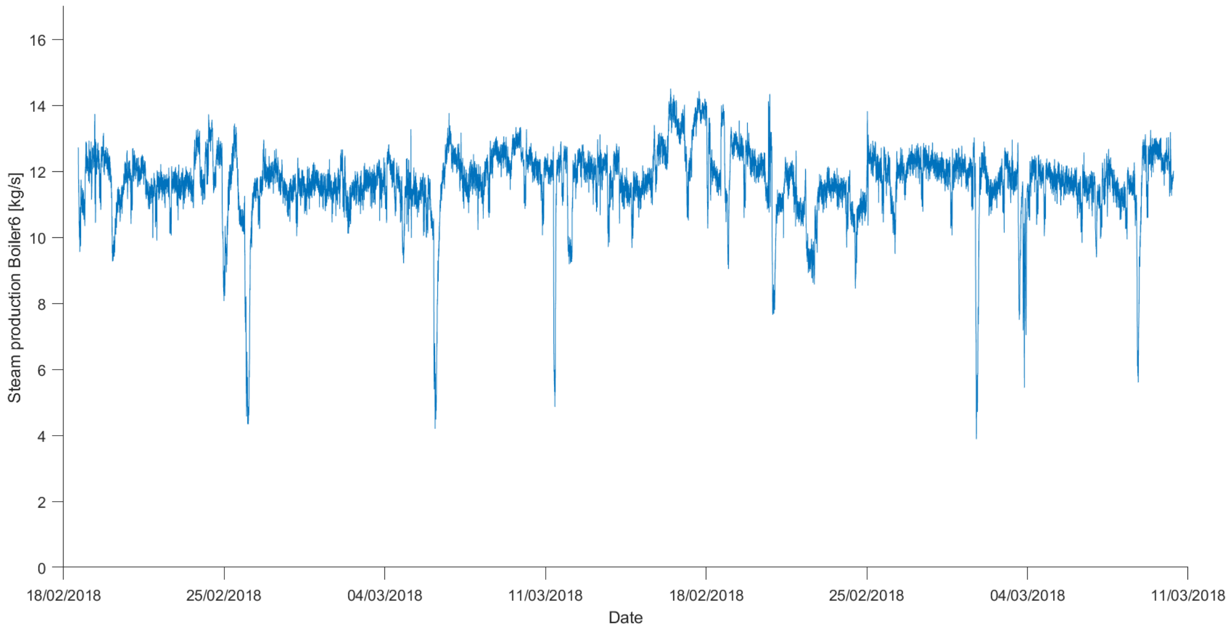

Figure 5.1: Steam production by Boiler6 during the period of roughly two weeks corresponding to the time period of the data used in the calculations in Section 5.1, 5.2 and 5.3.

There is also calculated some economic losses associated with the heat losses calculated in Section 5.2 and 5.3. These calculations are based on the average electricity price that Norske Skog had to pay during the two week data period. This price is 469 NOK/MWh. During this period, the electric boiler was used all the time, meaning that any efficiency increase in Boiler6 would lead to a corresponding reduction in steam produced (and hence electricity used) by the electric boiler. Also, by assuming that the electricity tax (EL-avgift) and grid cost (nettleie) can be neglected (the cost of these are very low compared to the electricity price, and taking into account the already harsh assumption behind the energy loss calculations (see Section 3), this should not have a huge impact on the results), the economic loss can simply be calculated by multiplying the heat loss with the electricty price.

5.1 Amount of leakage

The mole fraction of air leaking into the flue gas was calculated using Equation 5.1.1 derived in Section 3.1:

$$x_{Air} = \frac{\dot{n}_{Air}}{\dot{n}_{Mix}} = \frac{x_{O_2,Mix} - x_{O_2,Flue}}{x_{O_2,Air} - x_{O_2,Flue}} \tag{5.1.1}$$

$x_{O_2,Flue}$ and $x_{O_2,Mix}$ were estimated based on the data series described at the beginning of Section 5. The averaged values for $x_{O_2,Flue}$ and $x_{O_2,Mix}$, together with the assumed value for $x_{O_2,Air}$ and calculated value for x_{air} are presented i Table 5.1:

Table 5.1: Mole fraction of air leaking into the flue gas

$x_{O_2,Flue}$	0.037
$x_{O_2,Mix}$	0.093
$x_{O_2,Air}$	0.21
x_{Air}	0.32

Because there are no flow measurements of the flue gas, there is no direct way of knowing the absolute amount of leakage. It is neither possible to estimate it from a total mass balance due to uncertainties in the fuel composition (see Section 2.3.1), leaks in other parts of the process (see Section 2.3.6), and poor flow measurements of the air inlets (see Section 2.3.4).

The flue gas flow is therefore estimated based on the flue gas fan. There are however too few measurements regarding the flue gas fan to use compressor curves to estimate the flow at each instant. It is therefore assumed that the fan operates at its maximum capacity. This might not be as harsh of an assumption as it seems, because it is assumed that the flue gas fan is the main reason that boiler is constrained below rated capacity, and hence the fan should operate close to its maximum capacity.

The maximum capacity of the flue gas fan is given by the operating manual to be 29.3 Nm^3. Assuming a density of 1.2922 kg/Nm^3 and a Molar mass of 28.3 kg/kmol, Equation 5.1.2 (derived in Section 3.1) can be used to calculate a molar flow of the air leaking into the flue gas.

$$\dot{n}_{Air} = x_{Air} * \frac{q_{Mix} * \rho_{Mix}}{M_{Mix}} \tag{5.1.2}$$

The molar flow of air leaking into the flue gas between the two points of O_2-measurements (Figure 2.5) is calculated to be:

$$\dot{n}_{Air} = 432 \text{ mol/s} \tag{5.1.3}$$

5.2 Heat loss because of air leaking into the flue gas

As described in Section 3.2, there is a heat loss associated with air leaking into the flue gas. Using Equation 5.2.1 along with Cp_{Air} (both derived in Section 3.2), the values for T_{Mix} and T_{Room} from the data series described at the beginning of Section 5, and the $\dot{n}_{Air}$ calculated in the previous section, $\dot{Q}_{Heat,Air}$ can be calculated.

$$\dot{Q}_{Heat,Air} = \dot{n}_{Air} * Cp_{Air} * (T_{Mix} - T_{Room}) \tag{5.2.1}$$

Table 5.2: Loss from heating the air leaking in.

T_{Mix}	148 °C
T_{Room}	36 °C
Cp_{Air}	29.28 J/(K·mol)
$\dot{n}_{Air}$	432 mol/s
$\dot{Q}_{Heat,Air}$	1.42 MW
Economic loss	496 000 NOK/month

Table 5.2 presents the value of the heat loss as well as the values of the parameters used to calculate it.

There is also listed an economic loss in Table 5.2. This is calculated on the basis of the mean electricity price that Norske Skog had to pay during the same two weeks as the data series are from (see beginning of Section 5). Provided that the assumptions behind the calculation of $\dot{Q}_{Heat,Air}$ holds (these are already some harsh assumptions, see Section 3.2) the economic loss listed in the table quite accurately represents the economic value of $\dot{Q}_{Heat,Air}$. The economic loss is listed at a per monthly basis, although the underlying data only stretches a little over two weeks. This is because it is assumed that the two weeks that corresponds to the data points are quite typical for the winter season. Therefore, the economic loss can be regarded as representative for the winter time.

5.3 Heat loss from having the shunt pump running

As explained in Section 2.3.5.3, having the shunt pump running at its maximum capacity represents a heat loss resulting from reduced heat recovery in the economizer. Apart from fuel oil, the fuel burned in Boiler6 usually have a low sulfur content. This means that it is probably not necessary to have it running during normal operation.

By using the equations derived in Section 3.3, the temperatures of the water and flue gas streams in and out of ECO1 can be calculated. Table 5.3 shows the temperature results for these calculations.

Table 5.3: Temperatures before and after the shutdown of the shunt pump. Red = Calculated values, Black = Measured values.

	Before shutdown	After shutdown
$T_{f,out}$	250.1°C	241.5°C
$T_{f,in}$	379.9°C	379.9°C
$T_{w,out}$	207.1°C	213.4°C
$T_{w,in}$	141.2°C	111.5°C

Based on these temperature values as well as estimated mass flows and heat capacities (see Section 3), an energy loss resulting from the lack of heat recovery can be calculated. Combining the heat loss and the cost of operating the pump with the electricity price (as described at the beginning of Section 5) a corresponding economic loss is calculated. The results are listed in Table 5.4.

Table 5.4: Loss form having the shunt pump running at maximum capacity.

$\dot{Q}_{Heat,Shunt}$	0.371 MW
E_{Pump}	0.003 MW
Economic loss	131 000 NOK/month

$\dot{Q}_{Heat}$ represents the energy loss caused by the lack of heat recovery, E_{Pump} represents the energy needed to operate the pump. The economic loss calculated here indicates a substantial loss caused by having the pump running. It is however important to keep in mind the assumptions behind the calculation:

- The heat transfer coefficient, U, is assumed to be constant, but in reality it decreases when the flow through the heat exchanger decreases.[10] This is exactly what happens when the shunt pump is turned off, meaning that the possible heat recovery resulting from turning off the pump is less then the result in Table 5.4.
- It is assumed that the lower flue gas temperature after the ECO does not affect the heat recovery in AP1-4. The flue gas exiting the ECO is the same that goes into to AP1-4 (See Figure 2.4), and a lower flue gas temperature here will in reality reduce the heat recovery in AP1-4 (because of a lower ΔT over the heat exchanger). This also reduces the possible savings listed in Table 5.4. How large this effect is is hard to know, because there are no temperature measurements of the primary and secondary air after AP4, so the ΔT over the hot side of AP4 is unknown. However, because there is not a drastic shift in temperature of the flue gas, it indicates that this effect is small.
- The economic loss corresponding to $E_{Heat,Shunt}$ is calculated using the average price of electric energy for the data period. This is considered an accurate choice for this period, because the electric boiler is used every day during the period which means that increased energy efficiency in Boiler6 leads to less need for electricity. For the rest of the year however, this might not be the case. Taking into account that also the electricity price varies through the year it is clear that extrapolating this cost over the whole year should be done with caution.

The uncertainties listed here indicates that the economic loss is less than listed in Table 5.4. The economic loss will however be substantially higher than zero, and simply turning off a pump is not considered an expensive thing to do.

Another thing worth mentioning is that the reason for having the shunt pump there is (as mentioned in Section 2.3.5.3) to avoid that the flue gas temperature becomes too low when oil is used as a fuel. Oil has a higher sulfur content, so it becomes more important to keep a high enough flue gas temperature to avoid that SO_2 condenses in the system and cause corrosion.

5.4 Loss because Boiler6 is constrained

As mentioned in Section 2.3.6, the leaks throughout the flue gas system makes the flue gas fan become a bottleneck that limits the maximum power output from the boiler. This means that in addition to the direct cost of having to operate Boiler6 with leaks, there is also an alternative cost associated with the fact that Boiler5 and the electric boiler need to be utilized to cover the "gap" left by the Boiler6 that is constrained below capacity. This cost stems from the fact that the cost of fuel for Boiler6 is considerably less than the the cost of electricity (for the electric boiler).

Figure 5.2 shows the daily average power output from Boiler6 and the electric boiler. The data series starts at the end of 2013 for Boiler6 and the middle of 2015 for the electric boiler. Both series end in June 2018. The data is modified such that for the days when Boiler6 has a power output lower than $20MW_{th}$, both the data for Boiler6 and the electric boiler are removed. This is done because it is assumed that for these days there are other reasons for the lower output of Boiler6 (e.g. maintenance, low steam demand, lack of fuel).

As can be clearly seen from Figure 5.2, the power output of Boiler6 has a higher output during the winter season compared to the summer season; at least for the years 2014-2016. This is expected since the steam demand of the paper mill is higher during the winter. For the years 2017 and 2018, the power output does not show the same peaks during the winter season, suggesting that the boiler is constrained below capacity. The data for the electric boiler does not go as far back in time as the data for Boiler6. Nevertheless, it is clear that the utilization is higher during the winter of 2017 and 2018 than the winter of 2016. This suggests that the electric boiler now acts as a "peaker" during periods of high steam demand because Boiler6 cannot perform at the same level as before.

As mentioned above, the cost of the fuel used in Boiler6 is considerably less than the cost of electricity. This suggest that using the electric boiler as a "peaker" might

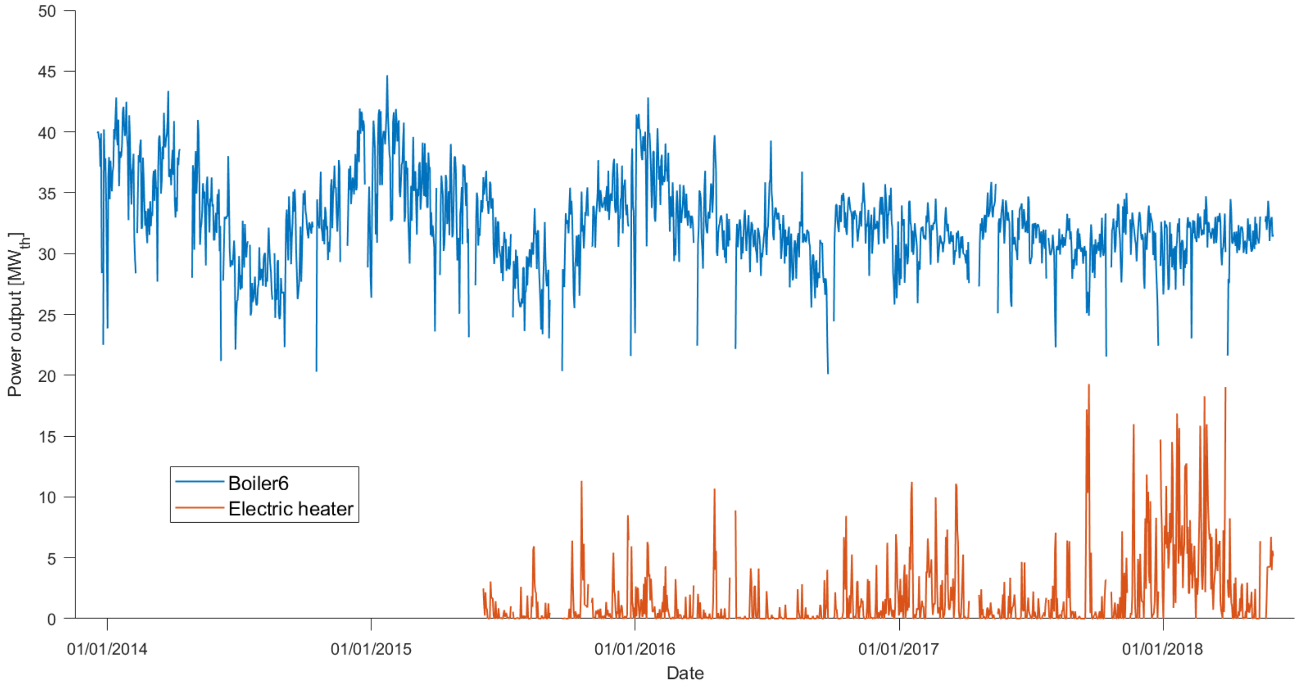

Figure 5.2: Daily average power output of Boiler6 and the electric boiler.

constitute a significant economic loss.

To estimate the economic loss associated with the use of the electric boiler instead of Boiler6, the daily average power output of the two boilers (the same data that is shown in Figure 5.2) together with the daily average electricity price for the corresponding days were used.

The data was modified such that the days where Boiler6 had a power output below $20MW_{th}$ were excluded from the calculation (same as in Figure 5.2). For the days where the sum of the power output from Boiler6 and the electric boiler exceeded $50MW_{th}$ (corresponding to the maximum capacity of Boiler6), the overshooting power output from the electric boiler was not included (an interesting point is that there are only two days over the three year period where this happens). Both modifications were performed such that only the power output of the electric boiler that could have been replaced by a well functioning Boiler6 was included in the cost calculation.

The cost of the fuel used in Boiler6 is very low compared to the electricity price, and it is assumed to be in the same price range as the sum of the electricity tax (EL-avgift) and grid cost (nettleie). By letting these two costs, associated with operating Boiler6 and the electric boiler respectively, cancel out, the calculation can be done using only the electricity price. This simplification is done so that the cost of the fuel for Boiler6 remains secret to some degree.

Table 5.5 lists the results over a few chosen time periods. The period from 5 Jun 2015 - 2 Jun 2018 represents the whole data period. The next three time periods represent periods of one year, beginning and ending in June.

Table 5.5: Economic loss over different time periods resulting from using the electric boiler instead of utilizing the full capacity of Boiler6

Time period	Economic loss [MNOK]
5 Jun 2015 - 2 Jun 2018	12.96
5 Jun 2015 - 4 Jun 2016	1.50
3 Jun 2016 - 2 Jun 2017	2.89
3 Jun 2017 - 2 Jun 2018	8.57
21 Feb 2018 - 8 Mar 2018	1.49

As one would expect from Figure 5.2, the economic loss shows a rising trend. Especially comparing the years 2015-2016 to 2017-2018 shows an increase of over 7 MNOK. In fact, only 16 days during the winter of 2018 (corresponding to the data period used in the energy calculations in Section 5.2 and 5.3) shows almost the same loss as the whole 2015-2016 year.

A few things to keep in mind when evaluating these data:

- The calculations are based on daily averages of the power output. This means that for days where it seems that a well functioning Boiler6 could have rendered the electric boiler redundant, it might have been necessary to use the electric boiler anyway to cover peaks during the day. The effect of this is that the economic loss is somewhat overestimated.
- As the previous item, the electricity price is also based on daily averages. This means that a varying power output from the electric boiler coupled with a varying electricity price could mean that the calculated economic loss deviates from the actual economic loss. The effect of this might lead to both an overestimation and an underestimation of the economic loss.
- It is assumed that a well functioning Boiler6 can have a power output of $50MW_{th}$ regardless of the fuel. This might be a bit optimistic considering the large

variation in fuel quality (see Section 2.3.1). However, it can be seen from Figure 5.2 that the boiler regularly had daily average power outputs in the 40-45 MW_{th} range in the past, indicating that it can perform not too far from the rated capacity even when running on fuel with lower quality.

- The effect of a possible fix of the steam turbine is not taken into account in these calculations. If the steam turbine where to be fixed, it would be possible to produce extra steam for the steam turbine during periods of spare steam generation capacity in Boiler6. This extra steam would be used by the steam turbine to generate electricity. If Boiler6 is constrained below capacity, there would be fewer opportunities to produce extra steam. In this case, the total economic loss of having Boiler6 constrained below capacity would be even higher, because one would have to include in the calculation the revenue of the electricity potentially produced by the steam turbine that is not realized because of lack of steam.
- The effect on the use of Boiler5 (the bark boiler) is not considered. The cost of bark might be higher than the average cost of fuel used in Boiler6, depending on the overall bark situation at the paper mill (see Section 2.1). This means that if Boiler6 was able to perform at its rated capacity, it might to some degree reduce the need for Boiler5, and thus reduce the overall cost (if bark is in short supply).

Although this list of items includes assumptions that suggest that the economic loss is overestimated, it is still clear from Table 5.5 that the economic loss increases roughly 7 MNOK over the course of two years.

It is not clear that it is the leaks that are the main reason for the change in the output of Boiler6 that, from Figure 5.2, seems to happen sometime during 2016, although it is a plausible explanation. Adjustments were made on Boiler6 by Foster Wheeler during the summer of 2016, and the results of these changes might very well be the cause of the abrupt change in maximum capacity that happens during 2016. A further investigation into this matter is out of scope for this thesis, but should be seriously considered for future work regarding the performance of the boiler.

6 Discussion

This sections raises issues that are not already covered in the previous section.

It is assumed in the calculation of heat loss because of air leaking into the flue gas (Section 5.2) that all of the heat that is currently used to heat extra air can be recovered in the economizer/air preheater if the leaks were to be fixed. This means that the temperature of the flue gas after the ESP (Figure 2.4) is assumed to remain constant after a potential fix.

In reality, not all of this heat would be recovered in the event of a fix, and the temperature after the ESP would increase. How much this increase would be is hard to predict, but the temperature is already quite high (148°C, Figure 2.4), so if it were to rise a lot it would get too hot for the flue gas fan that operates in the temperature range of 130°C-160°C.

The fact that the temperature after the ESP already is quite high (and could be even higher if the leaks are fixed), indicates either poor heat recovery or delayed combustion (or both). If there is considerable delayed combustion, it means that the amount of leakage is underestimated (because the calculation is based on the amount of O_2). An underestimation of the amount of leakage also means that the economic loss is underestimated.

The economic losses in Section 5.2 and 5.3 are calculated using the average electricity price for the data period (21 Feb - 8 Mar 2018), and then extrapolated to get a monthly price.

It is possible to argue that using the electricity price for this calculation might not be the most accurate thing to do taken into account the other two options of covering the lost heat: using the bark boiler or increasing the load of Boiler6. However, for the data period used in this calculation the electric boiler is used every day, so using the electricity price is actually quite accurate here. Regardless, it should be stressed that both the electricity price and the use of the electric boiler changes during the year, so the economic loss should not be extrapolated over the whole year.

The Dolezal-scheme for temperature control of the superheated steam had a couple of challenges described in Section 2.3.5.1. Both of the problems listed could be avoided if the superheated steam temperature was instead controlled by the method of injecting feed water directly into the superheated steam. The feed water has both a higher pressure and a lower temperature, resulting in both better water dispersion by the attemperator and larger controller gain. This is done in all other applications of the CFB boiler. [5] However, the prerequisite for doing this is that the feed water has a high enough purity so that corrosion in the steam network is avoided. The CFB boilers that use direct injection of feed water to control the steam temperature usually have a closed system for the steam/water-cycle, something that allows them to maintain a high enough water quality.

This closed steam cycle setup is unfortunately not applicable for the pulp and paper industry for a nmber of reasons:

- Complex and large steam networks with multiple sources and sinks makes follow-up on steam contamination difficult.
- Steam critical equipment such as turbines and drying cylinders sets high demands for purity and are costly to repair or replace.

The cost/benefit for a closed steam cycle simply favors open systems. It seems therefore that the Dolezal-system, which avoids the corrosion problem by only using condensed steam to cool the superheated steam, is the best choice for the boiler at Norske Skog.

There is still possible to do something about the temperature problem without switching out the Dolezal system. If the feed water that makes up the cold side of the heat exchanger is taken from before ECO1 instead of after, the temperature would be much lower (112°C instead of 207°C), resulting in a more efficient condensation and a larger controller gain. The reason that this is not done for Boiler6 might have something to do with where pipes and other components are physically placed at the site.

7 | Conclusion

The amount of air leaking into the flue gas between the two O_2-measurements are calculated to be 32% of the resulting mixture of air and flue gas. This number might be even higher if combustion takes place between the two points of measurement as discussed in Section 6. There is a heat loss associated with having the cooler air leaking in and reducing the amount of heat that can be recovered. This heat loss and corresponding economic loss is calculated (subjected to a few harsh assumptions discussed in Section 6) to be 1.42 MW and 496000 NOK/month respectively.

The indirect loss from having Boiler6 being constrained below rated capacity is considered the most important economic loss, because the assumptions behind this calculation are more reasonable. The economic loss was calculated to be as much as 8.57 MNOK for the year from 3 Jun 2017 - 2 Jun 2018. During only two weeks in the winter of 2018, the economic loss was estimated to be 1.49. These numbers provide good arguments for repairing the all the leaks that causes this. An alternative to fixing the leaks is to invest in a larger flue gas fan, so that this no longer is a bottleneck.

In addition to the economic loss resulting the leaks, the economic loss from having the shunt pump operating at its maximum capacity was also estimated to be 131 000 NOK/month. Even though this might be an exaggerated sum, simply turning off the pump is a very cheap remedy to avoid this loss. It is recommended that the shunt pump be turned all the time, except for when the fuel has a high sulfur content.

8 Bibliography

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A | Calculation of heat capacities

The equation and parameters used to calculate the heat capacity of a specific component are listed in this appendix. They are all obtained from the literature. [6]

$$\frac{Cp}{R} = a + b \cdot 10^{-3} \cdot T + c \cdot 10^{-5} \cdot T^2 + d \cdot 10^{-8} \cdot T^3 + e \cdot 10^{-11} \cdot T^4 \tag{A.0.1}$$

Table A.1: Parameters used with Equation A.0.1.

Component	***a***	***b***	***c***	***d***	***e***
Ar	2.500	0.000	0.000	0.000	0.000
CO_2	3.259	1.356	1.502	-2.374	1.056
H_2O	4.395	-4.186	1.405	-1.564	0.632
N_2	3.539	-0.261	0.007	0.157	-0.099
O_2	3.630	-1.794	0.658	-0.601	0.179

B | Matlab scripts and functions used in the calculation

```
%% Imports for leakage calculation

datafile = "C:\Users\ehtor\Downloads\Datasett_1_K6.xlsx";

O2aveHot = trimmean(xlsread(datafile, 'O2', 'D3:D21600'), 10)/100;
%Average value of O2-sensors before Eco

O2Cold = trimmean(xlsread(datafile, 'O2', 'E3:E21600'), 10)/100;
%Value of O2-sensor at the chimney

[num, txt, timeDate_raw] = xlsread(datafile, 'O2', 'A3:A21600');
%Reads date and time into the variable timeDate_raw

for i = 781:1440:21598
    timeDate_raw(i) = strcat(timeDate_raw(i), ' 00:00:00');
end
%Fixing bug in date and time import

dateFormat = 'dd/mm/yyyy HH:MM:SS'; %Specify format of date and time import

dateConv = datenum(timeDate_raw, dateFormat);
%Convert text string into date numbers

%% Calculate the percentage of air that has leaked into the flue gas, x_air
%Need O2aveHot, O2Cold

%x_air = n_air/n_flue, n_air + n_origFlue = n_flue,
```

```
%O2aveHot*n_origFlue + 0.21*n_air = O2Cold*n_flue
%O2aveHot*(n_flue-n_air) + 0.21*n_air = O2Cold*n_flue
%n_air*(0.21-O2aveHot) = n_flue*(O2Cold-O2aveHot)
%x_air = n_air/n_flue = (O2Cold-O2aveHot)/(0.21-O2aveHot)

x_air = (O2Cold-O2aveHot)/(0.21-O2aveHot);

%% Imports and specifications for Energy loss calculations

T_kjelrom = trimmean(xlsread(datafile, 'Sheet1', 'AG3:AG21600'), 10);
%Temperature outside the boiler

T_elfilt = trimmean(xlsread(datafile, 'Sheet1', 'M3:M21600'), 10);
%Temperature after EL-filter

Cp_air = (Cp_air_T(T_elfilt+273.15) + Cp_air_T(T_kjelrom+273.15))/2;
%[J/K*mol]
%Assume Cp_air is constant and equal to the average of the Cp at
%36 and 148 degC

%n_air = x_air*n_flue, n_flue = m_flue/M_flue

M_flue = dot([4.5 11.28 62.47 21 0.74]*(1-x_air) ...
/100+[0.21 0 0.78 0 0.01]*x_air, [32 44.01 28.01 18.02 39.95])/1000;
%[kg/mol}

m_flue = 29.3*1.2922; %[kg/s] Flue gas flow. Density calculated by using
n_air = x_air*m_flue/M_flue; %[mol/s]

%% Energy loss because of heating of extra air, E_heatLoss

E_heatLoss = n_air*Cp_air*(T_elfilt-T_kjelrom)/(10^6); %[MJ/s]=[MW]

%% Economic loss because of heating of extra air, Monthly_Loss

El_price = mean(xlsread("C:\Users\ehtor\OneDrive\Dokumenter ...
\MasterNorskeSkog\ELKjel.xlsx", 'Sheet1', 'E1529:E1544'));
%Average electricity price over the data period (21.feb-8.mar)

Monthly_Loss = E_heatLoss*24*31*El_price;
```

```
%% Imports and specifications for Shunt pump calcultaions

Tw_in = trimmean(xlsread(datafile, 'Sheet1', 'W3:W21600'), 10);
%[degC] Water after mix with recycle, before eco

Tw_feed = trimmean(xlsread(datafile, 'Sheet1', 'U3:U21600'), 10);
%[degC] Water temp before mix

Tf_out = trimmean(xlsread(datafile, 'Sheet1', 'AF3:AF21600'), 10);
%[degC] Flue temp after eco

m_feed = trimmean(xlsread(datafile, 'Sheet1', 'AE3:AE21600'), 10);
%[kg/s] Feed water flow

m_recycle = 6.1; %[kg/s] Water recycle flow

Cp_w = 4.35;
%[kJ/kg*K] Assume Cp_w = constant throughout temperature range
%(c. 110-215 degC)

Cp_f = (Cp_f_T(380+273.15, x_air) + Cp_f_T(Tf_out+273.15, x_air))/2;
%[kJ/kg*K]

%Assume Cp_f is constant and equal to the average of the Cp at
%380 and 250 degC

%% Calculate outlet temperature of water, Tw_out
%Need Tw_in, Tw_feed, m_feed, m_recycle

%(Tw_out-Tw_feed)*Cp_w*m_recycle = (Tw_in-Tw_feed)*Cp_w*(m_recycle + m_feed)
%Tw_out*m_recycle = Tw_in*m_recycle + (Tw_in-Tw_feed)*m_feed

Tw_out = Tw_in + (Tw_in-Tw_feed)*m_feed/m_recycle;

%% Calculate inlet temperature of Flue gas, Tf_in
%Need: Cp_w, Cp_f, m_feed+m_recycle, m_flue, Tw_in, Tw_out, Tf_out

%(Tw_out-Tw_in)*Cp_w*(m_feed + m_recycle) = (Tf_in-Tf_out)*Cp_f*m_flue

```

```
Tf_in = Tf_out + (Tw_out-Tw_in)*(Cp_w/Cp_f)*(m_feed + m_recycle)/m_flue;

%% Calculate UA
%Need: Cp_w, m_feed+m_recycle, Tw_in, Tw_out, Tf_in, Tf_out

%(Tw_out-Tw_in)*Cp_w*(m_feed + m_recycle) =
%UA*((Tf_in-Tw_out)-(Tf_out-Tw_in))/ln((Tf_in-Tw_out)/(Tf_out-Tw_in))

UA = (Tw_out-Tw_in)*Cp_w*(m_feed + m_recycle) ...
/deltaTm(Tf_in, Tf_out, Tw_in, Tw_out);

%% Calculate theoretical Tw_out and Tf_out when shunt pump is turned off
%Need: Tw_feed, Tf_in, Cp_w, Cp_f, m_feed, m_flue

syms Tw_out_new Tf_out_new

eqns = [Cp_w*m_feed*(Tw_out_new-Tw_feed) ...
== UA*deltaTm(Tf_in, Tf_out_new, Tw_feed, Tw_out_new), ...
    Cp_w*m_feed*(Tw_out_new-Tw_feed) == Cp_f*m_flue*(Tf_in-Tf_out_new)];
%eqns = [Tw_out_new == 5, Tf_out_new == Tw_out_new + 10]

vars = [Tw_out_new, Tf_out_new];
init_guess = [Tw_out, Tf_out];

[T_water, T_flue] = vpasolve(eqns, vars, init_guess)

%% Energy loss because of heating of extra air, E_heatLoss_shunt

E_heatLoss_shunt = m_flue*Cp_f*(Tf_out-T_flue)*10^(-3); %[MW]

%% Economic loss because of running the shunt pump, Monthly_Loss

E_pump = 3e-3; %[MW], shunt pump energy demand

Monthly_Loss_shunt = (E_heatLoss_shunt + E_pump)*24*31*El_price;
```

```
function result = Cp_air_T(T)
%Calculates Cp of air as a function of temperature.

result = dot([0.21 0.78 0.01], [Cp_O2(T) Cp_N2(T) Cp_Ar()]);
%[J/K*mol] Composition dry air: [O2 N2 Ar]

end
```

```
function Cp = Cp_Ar()
Cp = (2.5)*8.3145;
end
```

```
function Cp = Cp_CO2(T)
Cp = (3.259 + 1.356e-3*T + 1.502e-5*T^2 + -2.374e-8*T^3 ...
+ 1.056e-11*T^4)*8.3145;
end
```

```
function Cp = Cp_N2(T)
Cp = (3.539 + -0.261e-3*T + 0.007e-5*T^2 + 0.157e-8*T^3 ...
+ -0.099e-11*T^4)*8.3145;
end
```

```
function Cp = Cp_H2O_g(T)
Cp = (4.395 + -4.186e-3*T + 1.405e-5*T^2 + -1.564e-8*T^3 ...
+ 0.632e-11*T^4)*8.3145;
end
```

```
function Cp = Cp_O2(T)
Cp = (3.630 + -1.794e-3*T + 0.658e-5*T^2 + -0.601e-8*T^3 ...
+ 0.179e-11*T^4)*8.3145;
end
```

```
function result = deltaTm(Th_in, Th_out, Tc_in, Tc_out)
%Ideal counter-current heat exchanger
result = ((Th_in-Tc_out)-(Th_out-Tc_in))/ ...
log((Th_in-Tc_out)/(Th_out-Tc_in));

end
```

```
function result = Cp_f_T(T,x_air)
%Calculates Cp of flue as a function of temperature and
%amount of air leaking in.
comp_flue = [4.5 11.28 62.47 21 0.74]*(1-x_air)/100 ...
+[0.21 0 0.78 0 0.01]*x_air; %Composition flue[O2 CO2 N2 H2O Ar]

Cp_mol = dot(comp_flue, ...
[Cp_O2(T) Cp_CO2(T) Cp_N2(T) Cp_H2O_g(T) Cp_Ar()]); %[J/K*mol]

result = Cp_mol/dot(comp_flue, [32 44.01 28.01 18.02 39.95]);
%[kJ/kg*K]

end
```

```
%% Treatment of data used to compare Boiler6 and electric boiler

datafile = "C:\Users\ehtor\OneDrive\Dokumenter ...
\MasterNorskeSkog\ELKjelFiksDarlig.xlsx";

powerEl = xlsread(datafile, 'Sheet1', 'B3:B1630');
%Use of Electric heater

power6 = xlsread(datafile, 'Sheet1', 'D3:D1630'); %Use of Boiler6

[num, txt, timeDate_raw] = xlsread(datafile, 'Sheet1', 'A3:A1630');
%Reads date and time into the variable timeDate_raw

[num, txt, timeDate_raw2] = xlsread(datafile, 'Sheet1', 'A537:A1630');
%Reads date and time into the variable timeDate_raw

dateFormat = 'dd/mm/yyyy HH:MM:SS';
%Specify format of date and time import

dateConv = datenum(timeDate_raw, dateFormat);
%Convert text string into date numbers

dateConv2 = datenum(timeDate_raw2, dateFormat);
%Convert text string into date numbers

for i = 1:length(power6)
    if power6(i)<20
        power6(i)=nan;
        if i>534
            powerEl(i-534) = nan;
        end
    end
end
```