TAMPERE UNIVERSITY OF TECHNOLOGY

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COMBUSTION IMPROVEMENT OF SMALL-SCALE PELLET BOILER BY CONTINUOUS CONTROL

Master of Science Thesis

Examiner: professor Pentti Lautala
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ABSTRACT

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Combustion control plays a major role in regards of efficiency and emissions in energy production. The goal in this thesis was to design, implement and test feedback combustion control strategies in a commercial small-scale pellet boiler (25 kW) equipped with additional measurement equipment and customized actuators. The boiler was situated in the laboratory of Czech Technical University in Prague. The thesis was conducted as cooperation between Tampere University of Technology and Czech Technical University in Prague.

Introduction of small-scale combustion appliances and wood pellets as a fuel are carried out in this thesis. Combustion theory is also covered in order to make reasoned conclusions when choosing suitable control variables for the combustion control. The potential of some selected combustion control strategies are evaluated in respect of this particular combustion appliance. The control methods that were utilized are presented. The three feedback control strategies designed and tested in this thesis were residual oxygen in flue gas controlled by air or fuel feed and temperature in the upper end of the combustion chamber controlled by fuel feed. Also a cascade control structure which controlled both oxygen and temperature was designed and utilized. The main focus in the combustion control development was to minimize carbon monoxide (CO) emissions simultaneously maintaining high efficiency by reducing the amount of excess combustion air.

It was concluded in this thesis that due to the heavy dynamics of the process from inputs to outputs, the performance improvement of the boiler obtainable by the feedback control was limited. Thus compensation of short term disturbances was out of question with the actuators available. However, the tendency of the process to drift could be avoided with the control strategies that were proposed. Had the primary and secondary air feeds been conducted by two separate fans, the situation would have been very different and the potential of utilizing feedback control would have been greater. Also the long measurement delay of the gas analyzer limited the combustion improvement. Additionally, it was concluded that the operation point plays a significant role in the boiler operation and on partial load the emissions can be higher due to insufficient mixing of gases and lower combustion chamber temperature.

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Poltonhallinnalla on tärkeä rooli energian tuotannon päästöjen vähennyksessä sekä hyötysuhteen ylläpidossa. Tämän diplomityön tavoitteena oli suunnitella, toteuttaa ja testata takaisinkytketyllä säädöllä toteutettuja poltonhallintastrategioita markkinoilla olevaan pienen kokoluokan pellettikattilaan, joka oli varustettu lisämittalaitteilla sekä osittain kustomoiduilla toimilaitteilla. Kattila sijaitsi Prahan teknillisen yliopiston laboratoriossa. Työ tehtiin yhteistyössä Tampereen teknillisen yliopiston ja Prahan teknillisen yliopiston kanssa.

Diplomityön alussa esitellään nykyisiä pienen kokoluokan pellettipolttolaitteita sekä käydään läpi puupelletin ominaisuudet polttoaineena. Polton teoria käydään läpi jotta työn aikana voidaan perustellusti valita sopivat säätösuureet poltonhallinnalle. Muutamien valittujen säätöstrategioiden soveltamispotentiaali kyseessä olevalle kattilalle arvioidaan ja esitellään säädön suunnitteluun käytettävät metodit. Toteutetut ja testatut säätöstrategiat olivat savukaasun jäännöshapen säätö sekä ilman että polttoaineen syöttöä ohjaamalla ja polttokammion yläosan lämpötilan säätö polttoaineen syöttöä ohjaamalla. Edellä mainittujen muuttujien yhtäaikaiseen säätöön suunniteltiin ja testattiin kaskadisäätörakennetta. Päätavoite poltonhallinnan kehitykselle oli minimoida häkäpäästöt (CO) ja ylläpitää samalla korkeaa hyötysuhdetta vähentämällä ylimääräistä polttoilman syöttöä kattilaan.

Työssä ilmeni että prosessin raskas dynamiikka sisäänmenoista ulostuloihin rajoitti mahdollisuuksia parantaa kattilan toimintaa takaisinkytketyllä säädöllä. Tästä johtuen lyhyen aikavälin häiriöiden kompensointi säädöllä oli mahdotonta käytettävissä olevilla toimilaitteilla. Prosessin taipumus ajelehtia vakio sisäänmenoilla toiselle toiminta-alueelle pystyttiin kuitenkin esitetyillä säätömenetelmillä estämään. Mikäli prosessissa olisi ollut erilliset ensiö- ja toisioilmapuhaltimet olisi saatava parannus kattilan toimintaan ollut mitä todennäköisimmin suurempi ja näin ollen takaisinkytketyn säädön soveltamispotentiaali ollut suurempi. Hapen mittaukseen käytetyn kaasuanalysaattorin pitkä mittausviive myös osaltaan rajoitti saavutettavissa olevaa polton parannusta. Lisäksi työssä kävi ilmi että toimintapiste vaikuttaa huomattavasti palamisen laatuun kattilassa. Tämä kävi ilmi erityisesti toimittaessa osateholla, jolloin päästöt olivat suuremmat johtuen kaasujen ja polttoilman huonommasta sekoittumisesta sekä matalammasta polttokammion lämpötilasta.

PREFACE

This work was done in Czech Technical University in Prague (CTU) during spring and summer 2010 in cooperation with Tampere University of Technology (TUT). An article based on the results of this thesis is to be presented in the IFAC world conference in Milan 2011. The article was also a collaboration of CTU and TUT.

I would like to thank my instructors M.Sc. Timo Korpela and Ph.D. Tomas Björkqvist for their support and guidance. I would like to acknowledge the work effort by M.Sc. Korpela he carried out with the correlation analysis supporting this work. I would also like to thank my examiner prof. Pentti Lautala for guidance with the writing process.

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In Tampere on May 19th 2011

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Notations

C	Transfer function of controller
K	Controller gain
K_p	Process gain
L	Time delay
M	Joint sensitivity
M_s	Maximum sensitivity
M_t	Complementary sensitivity
P	Transfer function of process
S	Sensitivity function
s	Laplace variable
T	Time constant
T	Complementary sensitivity function
T_i	Integration time constant
T_d	Derivation time constant
λ	Stoichiometric ratio
τ	Normalized time delay
ω	Frequency

Subscripts

p	Process
i	Integration
d	Derivation

Abbreviations

CO	Carbon monoxide
CO_2	Carbon dioxide
H_2	Hydrogen
HCN	Hydrogen cyanide
K	Potassium
N	Nitrogen
N_2O	Nitrous oxide
NH_3	Ammonia
NO	Nitric oxide
NO_2	Nitrogen dioxide
O_2	Oxygen
Zn	Zinc
OGC	Organic Gaseous Compound
PAC	Programmable Automation Controller
STD	Standard deviation
PID	Proportional-Integral-Derivative (controller)

1 INTRODUCTION

During the last few decades biomass and especially wood combustion has come increasingly attractive throughout the size range in energy production (heat and power). One of the main reasons for the interest towards biomass and wood combustion is the increasing price of fossil fuels. Also the pursuit for less CO_2 releasing energy production is a motif to increase the use of wood fuels. This is due to the fact that wood can be considered a renewable fuel because the CO_2 produced in combustion is compensated by the consumption of CO_2 when new trees are grown, granted the trees are grown in a sustainable manner. However, if a considerable amount of hydrocarbons is produced in the combustion the efficiency of the combustion decreases and the polluting effect of biomass combustion increases thus vitiating the status of biomass as an environmentally friendly and CO_2 neutral energy source.

In order to meet the combustion conditions that assure minimal hydrocarbon emissions, the combustion process has to be carried out properly. This calls for continuous combustion control which is common in large scale boilers. As the efficiency and emission requirements are coming stricter in small-scale appliances as well, combustion control has come into focus also in this field. Usually combustion in small-scale appliances has been carried out with high amount of excess air which minimizes the amount of hydrocarbons but in turn it decreases the efficiency of the boiler and increases the amount of NO_x emissions (e.g. Eskilsson et. al., 2004). This is why optimization of the air feed is a crucial part of combustion control. Another task for control is to maintain the power of the boiler on a level equivalent to the desired heat load. This type of control is referred to as load control. A very common way to conduct load control has been thermostat control leading to intermittent operation which is both inefficient and produces significantly increased amount of emissions. (e.g. Tissari, 2008) However, new sophisticated small-scale combustion units can modulate their power output thus operating continuously.

Applying continuous combustion control demands utilization of control equipment such as PLCs, sensors and frequency converters. Despite this equipment is widely in the market and the control technology as such is well known, the implementation of these appliances in small-scale boilers has not been very common due to the increase in the total investment costs.

The work done during this thesis was a part of a nationally funded research project "Development of environmentally friendly decentralized energetics" in Czech Republic. The aim of this project was to develop methods and techniques to improve efficiency of small-scale boilers and to decrease emissions produced when they are operated. One way to reach this goal is utilization of combustion control in a way that does not increase the investment costs too much.

The aim of this thesis was to improve the combustion conditions in a small-scale pellet boiler by utilizing combustion control. Thus, the main aim was on combustion control instead of load control. The utilization of the combustion control was made based on theory of combustion process as well as on experimental process data. The control variables for the combustion control were oxygen concentration and temperature in upper end of the combustion chamber. Oxygen concentration in the flue gas provides information about the state of the combustion and gives estimates of produced emissions. The temperature measurement indicates the stability of both, the combustion and fuel feed. The designed feedback control schemes were implemented and tested on a pellet boiler situated in one of the laboratories of CTU. The quality of the control was evaluated mainly based on the emissions produced and the stability of the combustion conditions.

2 SMALL-SCALE PELLET COMBUSTION

Although pellet combustion appliances have been in market since the 1980s the major breakthrough has happened only during the past ten years. Pellet heating appliances suitable for central heating purposes were developed during the 1990s. Nowadays a notable share of domestic heat is produced with these appliances and the interest is only increasing due to increasing oil prices and tightening CO_2 standards. Other factors making pellet heating an attractive option for the consumer are convenience of the fuel and the possibility to install a pellet burner to an existing boiler, formerly equipped with an oil burner. (Van Loo & Koppejan, 2008)

2.1 Wood pellets

Wood pellets are compressed from industrial wooden waste such as saw dust and cutter chips. Also bark and woodchips are suitable for pelletizing. Their shape is usually cylindrical but sometimes they can also be square (Alakangas, 2000). Pellets are 8–12 mm in diameter although it is quite typical in Central Europe that pellets are 6 mm in diameter. The length of a pellet is from 10 to 30 mm. The moisture content of a pellet is 7–9 % and the ash content of a pellet is low, approximately 0.2–0.8 weight percent (Obernberger & Thek, 2004). Weight of a bulk cubic meter of pellets is 600–750 kg/m^3 and solid density is about 1100–1500 kg/m^3. The calorific intensity of pellets is 14 – 17.5 MJ/kg. (Alakangas, 2000; Alakangas et. al. 2007)

The diameter and the length of pellets are important variables in pellet combustion because most of the pellet boilers have fuel feeding systems that are based on volume e.g. screw conveyors. Large fluctuations in length cause inconsistent combustion process since a different amount of pellets (thus a different heat power) is fed during a revolution of screw conveyor. Also pellet density plays a major role in pellet combustion since high heating value requires high density. Fluctuations in all of the previous cause problems especially in boilers using fixed air and fuel feeds. One important quality parameter for pellets is abrasion which means the mechanical stability of a pellet. (Fiedler, 2004) High abrasion leads to a high amount of fines which causes feeding problems such as vaulting of fuel in the storage. Small amounts of biological binding agents can be used to decrease the abrasion. (Obernberger & Thek, 2004)

In Finland, the pellet production was c. 330,000 t in 2007 and this amount was produced by 24 pellet plants. Finland exports a large portion of the pellets produced since the domestic consumption is only about 117,000 tonnes. Of the domestic consumption, 61,000 tonnes were used in small-scale applications (<25 kW) the number of which in 2008 reached c. 15,000 units. The number withholds pellet boilers, stoves and buckets that are designed especially to combust pellet in normal fireplaces. The number of households is relatively low comparing to the number in Sweden and it is mainly due to a lack of subsidies for upgrading the heating system. Recently the state has started to subsidize this kind of heat appliance upgrading which should increase the interest towards pellet heating units. By the year 2012 the number of small-scale pellet heating units is anticipated to be c. 50,000 and due to that the domestic consumption is expected to increase to c. 300,000 tonnes. Nowadays most of the production, 71 %, is sold in bulk which is transported by normal trucks or by trucks equipped with pneumatic pellet transfer systems. The bulk production is stored in large silos in the plant and in special silos or storage rooms in the customer’s end. The problem with bulk deliveries is the amount of fines. A higher amount of fines is due to crumbling of pellets caused by mechanical stress during the loading, transport and unloading. (Selkimäki et al., 2010)

In Czech Republic, pellet production was 27,000 tonnes in the year 2008, the production capacity being 78,000 tonnes. There are 7 manufacturers producing pellets as their main activity and a few other manufacturers producing pellets as a marginal activity. When comparing to Finland the production is even more export oriented since only 10% of the produced pellets is used domestically. In domestic use the pellets are delivered mainly in small or big bags. In order to enable exporting, the produced pellets are usually of high quality, some producers meeting the pellet standards of Austria (ÖNORM M 7135) and Germany (DINplus). (Pelletatlas) The number of gasifying boilers operating on wood, pellets and wood briquettes in output range 15–50 kW is 40,000 (Heneman & Červinka, 2007). Increasing the number of pellet heating units is limited by high investment costs in residential use. A lack of pellet distribution channels and delivery equipment also restrains the market growth in Czech Republic. However, the state has started to subsidize investments of small-scale pellet heating units. (Pelletatlas) High quality pellets 6 mm in diameter are dominant in Central Europe. There is less abrasion occurring with these pellets comparing to the pellets used in Finland thus avoiding vaulting caused by fines. (Alakangas et al., 2007)

2.2 Combustion process

The combustion process is a complex process in which a fuel particle goes through different stages of combustion due to which the particle finally decomposes releasing heat. Remains of the particle are called ash. Combustion can be divided into continuous and batch combustion processes. Air feed can be conducted by natural draught or by forced draught.

2.2.1 Stages of combustion

The combustion process of a solid fuel particle is divided into different stages which are initial warming, drying, pyrolysis and char combustion or gasification. In case of a large particle all these stages can take place simultaneously. (Saastamoinen, 2002) Also ignition and combustion of volatile gases from pyrolysis can be considered as separate stages of the combustion. Initial warming, drying, pyrolysis and ignition are stages which consume heat (endothermic) and combustion of volatile gases and char are reactions which produce heat (exothermic). (Saastamoinen, 2002; Koistinen et. al, 1986)

In the initial warming stage the temperature on the surface of the particle rises to the temperature where the second stage, drying, begins (Saastamoinen, 2002). Heat energy for the warming is obtained from heat radiation deriving from the burning of pyrolysis gases or gas phase combustion (provided that those occur near the bed surface), radiation from the burning or glowing surfaces of the fuel particles and also from convective heat transfer between particles. (Horttanainen, 2001)

In the drying stage, the water that is combined in the fuel particle is vaporized. (Saastamoinen, 2002) Drying stage is said to begin when the surface of the particle reaches evaporation temperature although in practice the drying begins immediately when the temperature is raised from the initial temperature (Horttanainen, 2001). Factors that affect drying are the amount of heat energy, the initial moisture content of the fuel bed and the shape and size of the particle. The drying is faster if the amount of heat energy is high, if the initial moisture of the fuel is low and if the size of the particle is small. (Impola et. al., 1996) The primary air flow into the fuel bed affects the evaporation rate. Increasing the primary air flow will increase the evaporation rate due to higher heat input to the evaporation zone but after a certain critical point a maximum heat input is reached. After this point both the heat input and consequently the evaporation rate turn into a decline. Higher heat input is ensuing a higher flame front temperature and enhanced radiation heat transfer to the evaporation zone. (Yang et al., 2004)

After all the water or at least the water from the surface of the particle has evaporated, the temperature starts to rise until it reaches a certain level where the pyrolysis begins. In this stage, the solid particle decomposes into volatile matter and/or tar like substance due to thermal decomposition. The portion of fuel which is pyrolised depends on the fuel properties, final temperature and heating rate. Since pellets are a wood based fuel, the portion of mass that is pyrolised is about the same as with wood fuels in general, c. 80 percent. The matter that is left after pyrolysis is called char. (Saastamoinen, 2002) As the heat rate increases it accelerates pyrolysis generating more pyrolysis gases. Due to this, the porosity of the particle increases which accelerates burning and gasification. The particle size has some effect on the pyrolysis, the bigger the particle is the smaller the total surface area is. Also the warming is slower and thus there is less pyrolysis occurring. (Horttanainen, 2001) As was the case with evaporation rate, the pyrolysis rate increases as the primary air flow is increased until a certain critical point is reached. Since pyrolysis is strongly dependent on the temperature, the temperature rise causes the acceleration in pyrolysis. (Yang et al., 2004)

Ignition occurs as the combustible gases react with oxygen. The concentration ratio between these two has to be suitable in order to reach ignition i.e. the fuel/oxygen ratio has to be over the lean limit. Factors affecting ignition are sufficiently high temperature which increases the velocity of the molecules thus increasing the number of collisions between the two gases, low moisture content and sufficient mixing of oxygen with the volatiles at the ignition position. Mixing of gases when burning pellets (packed bed) is usually quite good due to a large surface area of the particles when compared with the flowing channels of gases. Ignition usually occurs in the gas phase when combusting wood particles in packed bed. (Horttanainen, 2001)

Combustion of pyrolysis gases provides the heat energy to the previous stages which consume heat but if there is air-staging in the process the heat from the combustion of pyrolysis gases may occur so far from the fuel bed that the heat rate to the fuel bed can be quite low. Volatile gases ignite outside the fuel particle once the lean limit of gases is reached. A situation where the ignition is prevented can occur when the proportion of volatile gases is too high, i.e. there not enough oxygen in gas mixture to enable ignition. When such a situation occurs the gas mixture is beyond the rich limit. Factors that affect this volatile gas combustion are temperature and mixing of oxygen with gases. Temperature affects the volatilization rate in the fuel particles and thus the amount of volatile gases coming to the combustion zone. Proper mixing of the gases is essential to reach the lean limit. If these factors are disturbed the gas combustion slows down. (Horttanainen, 2001; Ruusunen, 2001) However, in continuous combustion ignition is not an issue.

The final stage of combustion is char combustion and gasification. This stage differs greatly from pyrolysis stage. Unlike the pyrolysis which took place due to heat transfer from the ambient to the fuel particle, char combustion and gasification are caused by diffusion of reacting molecules to the surface and into the inner parts of char where they cause heterogenic reactions with the char. High temperature of a fuel particle accelerates the reaction. In char combustion the atmosphere is usually air or combination of air and flue gas and in char gasification the atmosphere is a mixture of gasification gases and gasification products. (Saastamoinen, 2002) In general, the gasification occurs in oxygen lean environment. Char ignites once the temperature of the char is high enough and when there is oxygen available on the surface of the particle. Char combustion is limited by the rate of pyrolysis because it prevents oxygen to reach the surface of char if the volatilization and/or drying is still occurring inside the particle. Such an effect is especially significant in the case of large fuel particles. Char combustion is also notably slower than the combustion of pyrolysis gases which in the case of low air flow can cause a situation where the pyrolysis gases consume all oxygen. (Horttanainen, 2001) Thus the primary air flow has a clear effect on the char combustion rate. By increasing the primary air the char combustion rate increases due to increased availability of O_2. Also accelerated devolatilisation rate and increased flame front temperature increase the char combustion rate. (Yang et al., 2004) In the case of wood based fuel the portion of char from the dry content is only 10–30 % but when combusted, about 25–50 % of the total heat production is generated during char combustion. (Flagan & Seinfeld, 1988; Koistinen et. al, 1986)

2.2.2 Emissions

When combusting hydrocarbons, the products ideally consist only of H_2O and CO_2, provided that the combustion is complete. In the case of wood combustion there are always several other products present as well. The most important emissions from wood pellet combustion are carbon monoxide, nitric oxides, unburnt hydrocarbons, particle emissions and dust. Because wood and thereby pellets contain only little sulphur, the sulphur emissions are not a major issue in wood pellet combustion since all sulfur emissions derive from the fuel. In general, it can be said that batch-wise operating combustion appliances produce more emissions than continuously operated ones (Johansson et. al, 2004).

2.2.2.1 Carbon monoxide (CO) and unburnt fuel OGC

Both carbon monoxide (CO) and unburnt fuel emissions OGC (Organic Gaseous Compounds) are generated due to incomplete combustion. Combustion may be incomplete because of too low combustion temperature, which may be due to, for example, too high feeding of secondary air which causes the flame to cool down. Another reason for incomplete combustion is insufficient mixing of air and volatile gases due to either lack of (secondary) air or poorly placed secondary air nozzles. Also too short residence time of the gases in the combustion chamber is one reason for high CO and OGC emissions. Short residence time may be caused by too excessive total air feed. (Johansson et. al, 2004) Higher fuel ash content has also been found to cause slight increments in CO emissions. However this can be seen as a consequence of higher air/fuel ratio which lowers the temperature in the combustion chamber causing incomplete combustion. (Sippula et.al, 2007)

2.2.2.2 Nitrogen oxides (NO_x)

In combustion, there is generated both nitric oxide (NO) and nitrogen dioxide (NO_2). 95 percent of the nitrogen oxide emissions in the flue gas are NO and the rest, about 5 percent, are NO_2. Later on in the atmosphere most of the NO becomes oxidized forming NO_2. That is why both of these nitrogen oxides have quite similar environmental effect, causing e.g. acid fallout. Combustion also generates nitrous oxide (N_2O) which is also a so-called greenhouse gas. The amount of N_2O in the flue gas is relatively small comparing to the previous two but still something to keep an eye on since the lifecycle of it is about 150 years which is a long time comparing to many other emission components. (Kilpinen, 2002) However, N_2O formation in biomass combustion is typically low (Van Loo & Koppejan, 2008).

Nitrogen oxides of the flue gas are generated from the nitrogen in the fuel and from the molecular nitrogen N_2 in the combustion air. The portion of nitrogen in the combustion air is c. 79 molecular percent. (Kilpinen, 2002) The portion of nitrogen in wood pellets is quite low, only 0.33 percent (Obernberger & Thek, 2004). Even though this amount is much smaller than the amount of nitrogen in the combustion air, nitrogen in the fuel is much more reactive and this is why fuels with higher nitrogen content have much bigger NO emissions than nitrogen free fuels. In combustion, some of the nitrogen in the fuel is released in the form of hydrogen cyanide (HCN) and ammonia (NH_3). These compounds oxidize to NO if there is oxygen present. This formation of NO is called fuel-NO mechanism. Fuel-NO mechanism is only slightly dependent on the temperature and fuel-NO is generated easily even in low temperatures. (Kilpinen, 2002) Nitrogen oxide emissions in small-scale biomass combustion are usually generated through fuel-NO mechanism (Klason & Bai, 2007).

Fuel-NO mechanism is sensitive to stoichiometry between fuel and combustion air. If there are reducing zones (air-lean zones) in the area where HCN and NH_3 are released, these compounds will react back to molecular nitrogen N_2. The reducing zones can be achieved by air staging in which only a portion of the combustion air is fed next to the fuel (primary air) thus creating zones where most of HCN and NH_3 react back to molecular nitrogen N_2 and the rest of the combustion air (secondary air) is fed later on in the furnace where there is only little HCN and NH_3 left. (Kilpinen, 2002) This is the reason why air-staging is carried out even in small-scale pellet boilers.

There are also three other nitric oxide formation mechanisms. In thermal-NO mechanism, NO is generated from the molecular nitrogen of the combustion air in high temperature. It has only little significance in temperatures below 1400˚C. Thermal-NO can be cut down by reducing surplus oxygen and temperatures in combustion chamber. (Kilpinen, 2002) Temperatures as high as 1400 ˚C are rarely reached in small-scale pellet combustion (Van Loo & Koppejan, 2008).

Next mechanism is fast-NO which takes place in the combustion zone of the flame. In fast-NO, the N_2 from the combustion air reacts with hydrocarbon radicals and after several intermediate reactions forms NO. When compared to thermal-NO, the proportion of fast-NO is highest in cool and air lean combustion conditions and with short residence times. Unlike thermal-NO, fast-NO is only slightly dependent on temperature. (Kilpinen, 2002)

The last NO formation mechanism is through N_2O which, depending on the conditions in combustion chamber, reacts back to N_2 or NO. Also in this case the source of nitrogen is the N_2 from the combustion air. Usually the reaction back to N_2 is dominant but if the air/fuel ratio and combustion temperature increase the NO formation increases also. (Kilpinen, 2002)

2.2.2.3 Particle emissions

Small-scale biomass combustion is a significant producer of fine particles. This is due to lack of flue gas cleaning equipment. The particle emissions from modern pellet boilers consist mainly of PM 1.0 particles (smaller than 1.0 µm in diameter) (Bäfver, 2008). Depending on the operating conditions, the PM 1.0 emissions have been found to vary from 5–25 mg/MJ. (Sippula, 2010)

Wood fuels contain inorganic material which in combustion forms ash. Fine fly ash derives mainly from vaporization of ash-forming elements in the wood fuel. (Sippula et al., 2007) There are mineral compounds in biomass fuels which are mainly bound to the organic structure of the fuel. Due to that they are easily released in the pyrolysis. Combustion temperature through all the combustion phases has a major effect on the fine ash formation. The higher the combustion temperature the higher is the production of fine ash particles. (Davidsson et al., 2002) Potassium (K) is the main fly ash-forming component in addition to zinc (Zn). Other factors affecting fine fly ash formation are combustion conditions (oxygen lean or rich) and fuel composition as well as moisture content and structure of the fuel. (Johansson et al., 2003b; Wiinikka & Gebart, 2004)

Some of the ash-forming elements do not volatilize but instead form bigger ash particles that either remain in the bottom of the grate thus forming the bottom ash or alternatively form the coarse fly ash. (Sippula, 2010) In addition to low volatile ash compounds, the coarse fly ash consists of unburnt char. (Flagan & Seinfeld, 1988)

In wood combustion the first fine particle emissions are soot. Soot particles are formed in the flame region from hydrocarbons. Soot particles form in fuel rich zones and due to insufficient mixing of air and volatile gases. There always exists fuel rich zones in the flame region in small-scale wood combustion appliances despite the overall stoichiometric ratio is over 1. (Tissari, 2008) Soot travelling in the flue gas can stain the heat exchanger surfaces and thus worsen the efficiency of the whole boiler (Fiedler, 2004).

In the flue gas from wood combustion there are also particles that consist of organic material. These organic particles are due to incomplete combustion. These fine organic particles largely condensate on other particles in the flue gas. (Tissari et al., 2008)

2.2.3 Air staging in pellet boilers

In order to optimize the combustion process the combustion chamber is divided into primary and secondary zones. Both of the zones have their own air supply i.e. primary and secondary air. Initial warming, drying, pyrolysis and char combustion take place in the primary zone. They also take place simultaneously since new fuel is added to the fuel bed all the time. These reactions are carried through with air ratio below the stoichiometric value. In the secondary zone, the volatile pyrolysis and gasification gases are combusted with excess air. (Fiedler, 2004) The air division is usually made by structural means so it is not possible to change the ratio later if the boiler is equipped only with one air fan. (Korpela, 2005)

Primary air is usually fed to the chamber under the grate. Drying, pyrolysis and final combustion, in which the char burns, take place on the grate. If there is not enough primary air the burning of the char slows down causing a decrease in temperature and an increase in the size of the fuel bed. If there is too much primary air, NO_x emissions increase (Kilpinen, 2002). The main reason for the air staging is to reduce NO_x emissions by creating reducing zones into the primary zone.

Secondary air is fed to the chamber from chamber walls or roof depending on the burner structure. The purpose of secondary air is to provide air for combustion of the gases that have pyrolysed and/or gasified from the fuel. It is essential that the secondary air that is fed to the chamber mixes well with the volatiles. This can be done by providing sufficient speed for the secondary air and by appropriate placing and dimensioning of nozzles. (Kilpinen, 2002) Too high feeding of secondary air cools the flame, which causes more CO emissions and shortens the residence time of the gas in combustion chamber, which in turn causes an efficiency reduction since all the pyrolysis gases are not burned in the boiler. Additionally, due to the shorter residence time of the flue gas in the boiler the ability of the boiler to receive heat is reduced since there is less time for the heat transfer from the flue gas to water. This is even more significant factor reducing the efficiency of the boiler than the growth of unburnt pyrolysis gases. If there is a shortage of secondary air, all the gases are not burned causing CO and OGC emissions and decrease in efficiency. (Johansson et al., 2003a; Johansson et al., 2004)

When considering minimization of both CO and NO_X, an optimum operating area for total air feed can be found, which can be seen in Fig. 2.1 (Eskilsson et al., 2004).

In the Fig. 2.1, the total stoichiometry equals the stoichiometric ratio which is the ratio between the feeded amount of combustion air and the stoichiometric (theoretically needed) amount of combustion air i.e. $\lambda = \dot{V}_{air} / \dot{V}_{Stoic.Air}$.

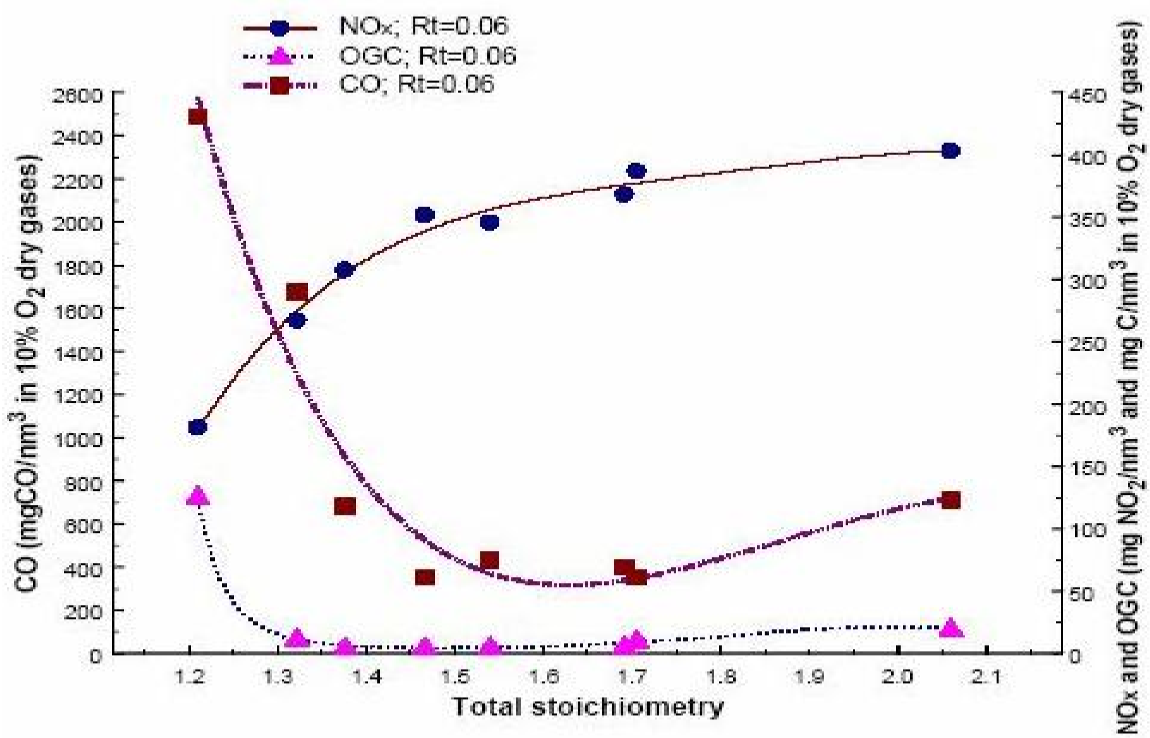


Figure 2.1. *Emissions in flue gas (CO, NO_X, OGC) with different stoichiometries (lambda values)* (Eskilsson et al., 2004/ *edited by* Korpela, 2005)

Usually the emphasis on development of pellet boilers has been on minimizing the amount of unburnt fuel in the flue gas, which is done by feeding more air into the system. Due to this, the amount of excess air is often too high from emission point of view. (Eskilsson et al., 2004; Šulc et al., 2009)

2.3 Pellet heating systems

There are two kinds of pellet heating units available which are central heating units and pellet stoves. Central heating boilers are divided into integrated units which include both the boiler and the burner, and two-unit boilers in which these appliances are separate units. An appliance that can also be counted into the category of central heating units is a pellet stove with a water jacket. Thermal output of pellet heating systems in domestic use ranges from 10 – 40 kW but usually the output is less than 25 kW. (Fiedler, 2004; Van Loo & Koppejan, 2008)

2.3.1 Central heating boilers

Central heating pellet boilers can be used to heat single- or multifamily houses. Pellet boilers are quite similar to oil boilers excluding the fuel itself, the fuel feeding system, vertical heat exchanger surfaces and larger combustion chamber. Heat exchanger surfaces are made vertical in order to prevent soot, fly ash and slag to deposit on the surfaces thus disturbing heat transfer which in turn lowers the efficiency of the whole unit. Comparing to oil boilers, biofuel boilers also require larger combustion chamber because if there is too little space, the flame reaches cool convection surface decreasing the temperature of the flame. This temperature decrease results in worsened gas combustion decreasing efficiency, increasing the amount of emissions and fouling the exchanger surfaces. Pellets are fed from the hopper to the combustion chamber by a conveyor. In the combustion chamber the ignition of the fuel is conducted by the means of an electric device or by maintaining a pilot flame. Combustion taking place after ignition generates hot flue gases. The heat of the flue gases is transferred to the boiler water by conduction through the heat exchanger. The heated water is transported to the heat distribution system by a circulation pump. Combustion air is fed by an electric fan which in many cases provides also secondary air into the system. The maximum power of the boiler and sufficient heat transfer over the whole power range define the size of the combustion chamber. Some boilers can automatically regulate the heat output in the range from 30 to 100%. (Fiedler, 2004; Van Loo & Koppejan, 2008)

Two-unit boilers consist of a separate boiler and a burner. Pellet burners can be installed into existing boilers if the requirements mentioned earlier are met. In many cases the burner is made by a different manufacturer than the boiler itself, which may lead to decreased efficiency due to compatibility problems between the two units. For example, pellet burners cause higher flue gas flow than oil burners which may lead to a situation where the residence time of the hot flue gas in the boiler is too short, resulting in too hot flue gas exiting the boiler which lowers the efficiency of the boiler. Too short residence time of the flue gas also causes emissions of hydrocarbons. Most of the pellet boilers in Sweden and Finland are two-unit boilers. (Fiedler, 2004; Van Loo & Koppejan, 2008)

Integrated boilers are the most common type in Austria and Germany. Integrated boilers have a fixed burner attached to the boiler. Comparing to two-unit boiler the advantage of integrated boiler is the better compatibility between the burner and the heat exchanging surfaces of the boiler since they are specifically designed to operate together. (Fiedler, 2004; Van Loo & Koppejan, 2008) Manufacturers of integrated boilers often promise a high efficiency (90% >) for the boilers. When comparing the two-unit boilers used in Finland to the integrated boilers used in Central Europe, the integrated boilers have the benefit of using very high quality pellet 6 mm in diameter. Since these boilers are especially designed for these pellets, they would not work as well with the lower quality pellets that are in the market in Finland. (Alakangas et al., 2007)

2.3.1.1 Pellet burners

There are burners available in the market that are solely designed to combust pellets and also burners, which are designed to combust woodchips but with different settings can combust pellets (Korpela, 2005). Pellet burners are divided into three types according to how the pellets are fed into the burner. These types are horizontally fed burners, bottom fed burners and top fed burners. (Van Loo & Koppejan, 2008) The different types are illustrated in Fig. 2.2.

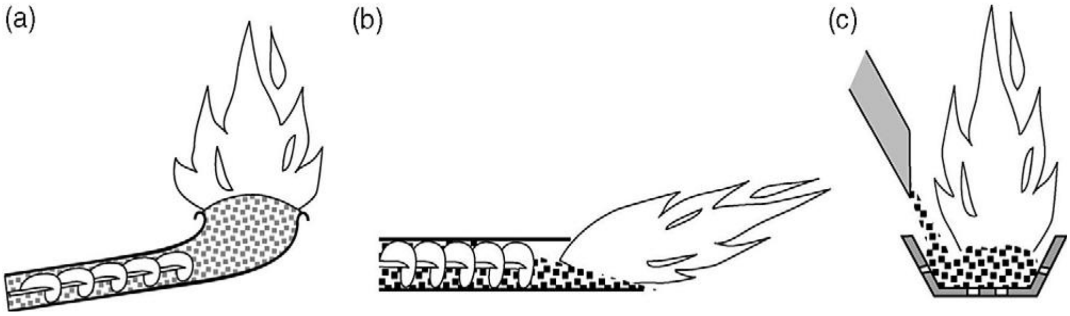


Figure 2.2. *Different burner types: Underfed, horizontally fed and top fed burner. (Fiedler, 2004)*

Top fed burners are very common in both boilers and stoves. Top feeding technique is considered rather safe because it minimizes the risk of back burn since the pellet store is always separated from the furnace. The so-called afterglow time after the burner has been switched off is relatively short compared to the other techniques. Also dosing of pellets is accurate. Disadvantages of top fed burner are that falling pellets stir the fuel bed releasing increased amount of dust and unburned particles. Also these falling pellets disrupt the burning process resulting in an unsteady combustion behavior. In underfed burners fuel is pushed onto the grate by a screw conveyor. Conveyor forces the fuel onto the combustion disk thus pushing the ash over the edges of the disk and that is why no ash removal equipment is necessary. Primary air is fed from the pellet supply or through holes at edge of the combustion disk. Secondary air is supplied to the combustion disk or by tubes above the disk. The advantage of this feeding system is that the combustion is quite steady. Disadvantages are that it has very long after glow and that there is a risk of a back burn. This is why in these burners pellets are moved on several phases thus creating pellet free zones which can prevent back burn (e.g. Aritherm BeQuem). Horizontally fed burners are very similar to bottom fed burners, only difference being the shape of the combustion bed. Additional ash removal equipment is necessary for this kind of burners. (Fiedler, 2004) In horizontally fed burners a primary combustion zone can be clearly defined since the gases are released during devolatilization in the combustion chamber. (Van Loo & Koppejan, 2008) In horizontally fed burners the primary air is fed under the grate and the secondary air is usually fed from nozzles on the walls or the roof of the combustion chamber.

2.3.2 Pellet stoves

The basic principles are mainly the same with stoves and boilers. Pellet stoves are used to provide heat for single rooms or small apartments. The heat is transferred to the surrounding space by convection and radiation. (Fiedler, 2004) Output of the stove can be regulated according to room temperature (Pellettienergia). In some stoves there is a fan circulating air through the boiler in order to improve heat transfer to ambient air. Combustion air is sucked into the boiler due to underpressure in the combustion chamber which is caused by a draught fan pushing flue gas out of the stove. Pellet stoves have usually top-fed burners. (Fiedler, 2004)

There are also pellet stoves that are equipped with water jackets. These stoves can be connected to central heating network and/or hot-water tank. The maximum heat power output is c. 10 kW and output can be regulated according to room temperature. However, the regulation is often conducted by thermostat control resulting in “on/off”-control. (Fiedler, 2004; Tissari, 2008)

2.4 Automation in pellet heating units

Automatic operation in pellet heating units is achievable by combining load control and combustion control. (Obernberger & Thek, 2002)

Load control is conducted by measuring the temperature of circulating water in the boiler. Load control can also be conducted by measuring the ambient room temperature. (Fiedler, 2004) Often the pellet boilers are run by thermostat control which leads to cyclic and intermittent operation (Johansson et al., 2003b). In some more sophisticated boiler systems, also the temperature outside the building is measured and this is used to predict the impending load (SHT). In sophisticated systems it is possible to have power level modulating between 30–100%. In these systems the boilers are equipped with draught fans instead of air fans. (Fiedler, 2004)

The combustion control in sophisticated systems is based on utilization of a lambda sensor and/or temperature measurement from the combustion chamber. These kinds of boilers are common in Central Europe. In systems using modulating power levels, frequency converters are used to provide input signals for feeding screw, air fans and draught fans (SHT, Windhager). By utilizing oxygen/lambda sensor for combustion control, it is possible to define oxygen value below of which the process should not operate on a given power level. Thus it is possible to conduct air feed so that the oxygen remains at such a level that emissions are minimized and the efficiency is high due to avoidance of excess air feed. Yet, these limit values for oxygen are equipment and power level dependent so for every power level the limit value should be redefined. (Eskilsson et al., 2004; Oravainen & Linna, 2004) The air feed to the boiler can be conducted by controlling a draught fan based on the under pressure (draught) measurement from the boiler and the primary and secondary air division is done by controllable flaps (Windhager, SHT). Another possibility is that there are two separate air fans conducting the primary and secondary air feed. In these cases the primary air feed is set according to the fuel feed (from load control) and the secondary air feed is used to trim the oxygen level to a desired value according to the oxygen/lambda measurement. It is possible that there is a separate draught control in boilers equipped with separate fans also. (Obernberger & Thek, 2002) Combustion control with the temperature measurement from the combustion chamber can be utilized to make the pellet feed as even as possible because temperature measurement is a good indicator of the heat power obtained, thus indicating the stability of fuel feed. By utilizing a soft-sensor approach, the temperature measurement can be used as an oxygen measurement since in pellet combustion the oxygen level usually gives quite good negative correlation to the temperature in the combustion chamber. (Korpela et al., 2008) Temperature measurement can also be utilized in order to control the recirculation of flue gas in the boiler (Obernberger & Thek, 2002). Some boiler manufacturers utilize adaptive control in their boilers (SHT).

Pellet manufacturers also offer automatic pellet transport appliances which move pellets from bigger storage room to the fuel hopper of the boiler. These transport appliances are usually screw conveyors but also pneumatic conveyors are available. In these pneumatic conveyors pellets are moved from the storage room to the intermediate silo by air suction. Nowadays, the state of the art boilers include automatic cleaning of the heat exchanger surfaces in order to keep the heat transfer as efficient as possible. Also the ash is removed from the combustion chamber automatically. (Windhager, SHT) If necessary, the automatic maintenance procedures also include grate sweeping which evens the fuel bed and pushes ash to the ash bin from the grate. Grate sweeping can be conducted by moveable rods which are operated at given intervals.

3 LABORATORY BOILER SYSTEM

The experimental work during this thesis project was conducted in the heavy laboratories of Czech Technical University in Prague. The boiler that was used was a commercial boiler made by Verner Inc. which is a Czech boiler company. The boiler was also used for testing of biomass pellets made from alternative sources. However, the testing of these pellets was not a part of this thesis and thus this work is not included in the thesis. The measurement equipment included sensors and the custom made switchboard electronics designed in CTU's Department of Instrumentation and Control engineering. All the used equipment is thoroughly presented in the following chapters.

3.1 Boiler and burner

The boiler that was used is Verner A25 with a rated thermal output of 25 kW. The boiler is an integrated boiler. The boiler is designed to combust biomass and wood pellets. The rated efficiency of the boiler is 92.7 percent with wood pellets as fuel. Standard fuel feed at rated output is 5.6 kg/h and the boiler is designed for pellets of 6–8 mm in diameter. (Verner datasheet) In all the experiments 6 mm pellets were used. The burner is top-fed and the fuel is fed by a screw conveyor. The conveyor feeds the fuel from the storage hopper to the grate through a hole in the back wall of the combustion chamber. The original control unit feeds the fuel in preset periods. These periods include the screw rolling period and the idle state. The boiler is equipped with one electric air fan which is originally impulse-controlled and which can be set to four different air flow rates. However, in the experimental setup the air fan has been changed to be driven by a frequency converter. The fan feeds combustion air into the combustion chamber. Before the combustion chamber the air is split into primary and secondary air. Primary air is fed to the chamber under the steel grate and the secondary air is fed through nozzles on the sidewalls of the burner. The ratio between primary and secondary air is fixed but it can be changed by manipulating a disk valve that can choke the primary air flow (see Fig. 5.1). Operating characteristic of the valve is strongly nonlinear due to the type of the valve. In the combustion chamber there is a moveable grate sweeper to even the fuel bed and to move ash to the ash bin which is located at the end side of the combustion chamber. Original control unit conducts the grate sweeping procedure periodically, once every 10 minutes.

In Fig. 3.1, made by M.Sc. Viktor Plaček, there is depicted the functional scheme of the boiler with the additional sensors and data acquisition equipment used in the measurements performed during this project.

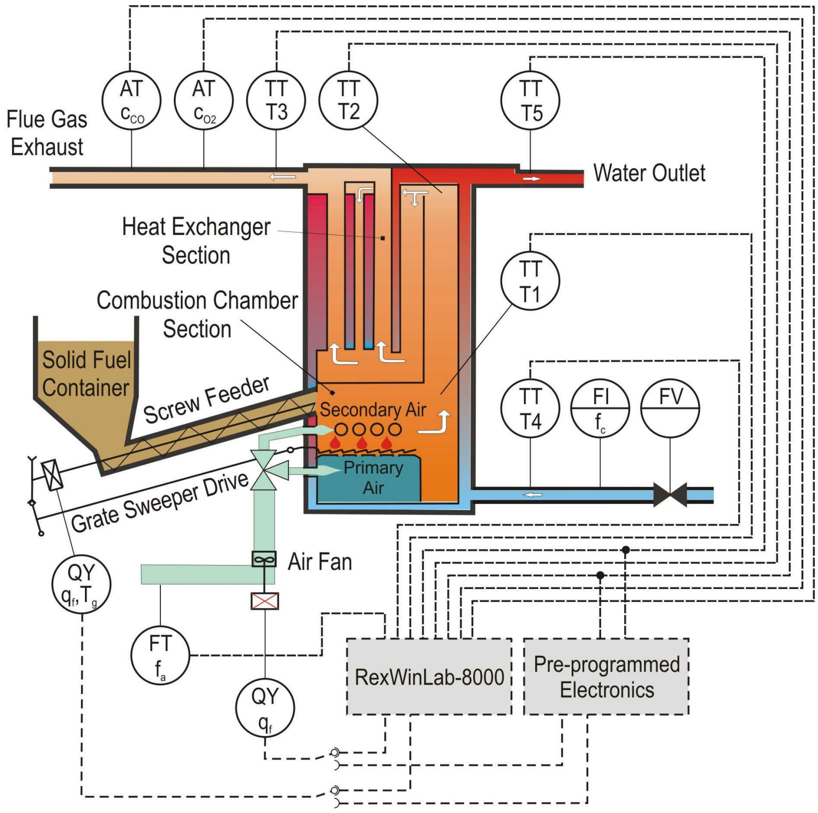


Figure 3.1. *Functional boiler setup (Plaček, 2010).*

Flue gases from the boiler are led to a chimney. There is no draught fan in the flue gas duct i.e. the boiler operates in natural draught conditions. The variables that are measured in the boiler in the factory setup are the temperature of outgoing water and temperature of flue gas (Verner datasheet).

3.2 Measurement system

The locations of measurement sensors are also illustrated in Fig. 3.1. In the experimental setup there were four thermocouples installed. Temperatures that were measured were combustion chamber temperature (T1), temperature after the first heat exchanger (upper end of the combustion chamber) (T2), temperature of the flue gas (T3) and temperature of the outgoing water (T5). Temperature of the inlet water (T4) was measured by a conventional thermometer placed in the tube of the ingoing water. In addition to the thermocouples, there was a gas analyzer measuring the concentrations of CO and O_2 in the flue gas. The gas analyzer was sample taking type and the analyzer part consisted of electro-chemical cells. In the last experiments also an air flow meter was installed to the duct preceding the combustion air fan. Flow rate of water was read periodically from a meter installed in a tube feeding the water into the water circuit.

Temperature measurement of the combustion chamber (T1) was located c. 20 cm from the end of the grate. The thermocouple was still in the flame zone and due to that, the variation in the signal was quite large. Since the thermocouple could see the flame, the radiation heat transfer also affects the measurement. The next thermocouple (T2) was located downstream the flue gas duct, after the first heat exchanger. At this point the gas flow can be assumed to be sufficiently mixed, i.e. no significant channeling occurs, which means that the temperature value obtained represents the situation well enough. Variation in this signal was significantly smaller than in the case of the previous thermocouple due to absence of the flame front and radiation. The thermocouple measuring the temperature of the flue gas (T3) was located just before the gas analyzer probe, c. 25 cm after the flue gas comes out from the boiler. Also at this point the gas flow can be said to be sufficiently mixed and thus the measured temperature is representative. The temperature of the out coming water (T5) was located quite close to the boiler wall, c. 10 cm after coming out from the boiler. The thermocouple was located at the bottom of the tube but since the spatial temperature distribution in the tube of the out coming water is minimal, the location of the thermocouple has only minor effect on the reliability of the measurement.

The sample probe of the gas analyzer was located in the flue gas duct c. 30 cm after exiting the boiler. The gas analyzer consists of the sample probe, sampling pump, sample pre-processing units (condenser, flue gas drier etc.) and an electro-chemical cell(s) which carries out the actual analysis procedure. The analyzer can measure several gas components and every gas component has its own cell in the analyzer. Concerning oxygen, when compared to a lambda probe based on electro-ceramic technology, the dynamics and dead time of the analyzer are heavier which is due to the more complex structure. The operation of an electro-chemical cell is based on oxidation-reduction reaction. In the oxygen cell, oxygen is reduced on cathode to hydroxyl ions which are carried via an electrode to the anode on which they oxidize the metal-anode. According to Faraday’s law, the current that is generated is proportional to the amount of oxygen reduced on the cathode. Pre-processing units are necessary to avoid the breaking of the analyzer cells and to enable long and continuous measurement sessions from wet flue gas. (Docquier & Candel, 2002; Torvela, 1993)

The inputs to the process were fuel feed, air feed and grate sweeping. Also the position of the air staging valve was an input to the process but since it was not online controllable it is not discussed any further in this part. The air staging valve is discussed more closely in Chapter 5.1. The fuel feed to the process was controlled by varying the fuel feed period which consisted of a constant rolling time of the screw feeder and variable idle state time. Though it was possible to control the actual rolling time of the screw feeder, the desired rolling time variations would have been several tenths of a second and since there is a time delay of some tenths of seconds from the PC to the actuator and back, a more accurate control could be achieved by varying the idle state. Air feed could be controlled by varying the frequency of the fan motor via the frequency converter. The grate sweeping could be controlled online by setting the intervals according to which the grate sweeping would be carried out. The amplitude of the grate sweeping movement could only be changed mechanically.

Status data of the grating, fuel feeding instances, opening of the hopper lid, frequency of fan motor and fire starting resistor were also acquired. Also the feeding commands coming from the original control system were acquired. All the variables gathered are listed in the Table 3.1.

Table 3.1. List of all the variables measured and acquired.

Variable	**Type**	**Data type**	**Unit**
Grating	Input	On/Off	-
Fuel feeding	Input	On/Off	-
Time	Output	Continuous	s
Pellet hopper limit switch	Output	On/Off	-
Grating limit switch	Output	On/Off	-
Temperature of water (T5)	Output	Continuous	°C
Temperature after the first exchanger (T2)	Output	Continuous	°C
Temperature in combustion chamber (T1)	Output	Continuous	°C
O_2 concentration	Output	Continuous	%
CO concentration	Output	Continuous	ppm, 10% O_2
Feeding (by original control unit)	Output	On/Off	-
Frequency of fan motor	Input	Continuous	Hz
Temperature of flue gas (T3)	Output	Continuous	°C
Air flow rate	Output	Continuous (only in the last two experiments)	m/s

The experimental setup included also a switchboard containing protection against short-circuit and overload situations, as well as prevention of forbidden combination of inputs. The switchboard also included power sources, central earth and emergency stop functions. The switchboard contained a RexWinLab-8000 control and data acquisition unit which is based on PAC (Programmable Automation Controller) Wincon 8000 series (Šulc & Vrána, 2009). All the sensors were connected to the I/O-card of the PAC. The PAC and a computer were connected by a network cable and the PAC sends all the measured data to Simulink running on the computer. The PAC and Simulink were made compatible with each other by an RDC communication block of the REX control system (see Chapter 3.3). Finally the data was saved to the computer in Simulink environment. The sample rate of the system was 5 Hz.

3.3 REX control system

REX control system software is a tool for developing control algorithms and for monitoring the actions of developed algorithms. The software package is divided into Rexdraw, Rexview and Rexcore programs. Rexdraw is used to design and compile control algorithms made from the REX block set blocks. After the compilation the algorithm schemes are loaded into Rexcore and run there. The operation of an algorithm running in Rexcore is monitored by Rexview. REX also includes Rexlib which is a block set containing basic function blocks for control algorithm development. These blocks are compatible with Matlab's Simulink. (Šulc et al, 2009; Šulc & Vrána, 2009; Rex controls, 2002)

Since Rexlib blocks are compatible with Simulink, the algorithms can be developed and simulated in Simulink. There is also one special block in Rexlib called RDC, which provides a communication interface between Simulink and Rexcore/PAC. One RDC block can send at most 16 signals to another device but it is possible to use several RDC blocks to enable more signals to be sent. (Šulc et al, 2009)

By using the RDC block all the variables that are measured from the process are sent to the RDC block in Simulink and thus they can be saved onto a computer. Also all the inputs to the actuators are sent via the RDC block.

4 CONTROL SYSTEM DESIGN

In this chapter all the control methods utilized in automatic control of the boiler are presented. Proper variables for the feedback control are evaluated and the process identification according to the chosen variables is conducted. Controller tuning is done based on the identified process models. A few different feedback control schemes and one cascade control scheme are presented later in this chapter. Also the possibility of compensating the disturbing effect of grate sweeping is evaluated.

4.1 Process identification

Process dynamics can be obtained by using transient response in which a change is introduced to the input of the process and the process output is measured. These input changes can be e.g. steps, ramps or impulses and depending on the change type we obtain step response, impulse response etc. Before the input variable is changed, the process must be in a steady state. In order to get a sufficient signal-to-noise ratio the size of the change into input signal should be large enough, although too large change might push the process to nonlinear operating area. (Åström & Hägglund, 2006) Step changes to the input were used in this work.

Dynamics between input and output signal can be described by a first order transfer function G(s) with a delay (FOTD) which contains the following parameters: process gain K_p, process time constant T and process time delay L.

$$G(s) = \frac{K_p e^{-Ls}}{Ts + 1} \tag{4.1}$$

First order transfer functions can be fitted to the step responses acquired during a step response experiment. The procedure of determining the FOTD parameters visually is illustrated in Fig. 4.1.

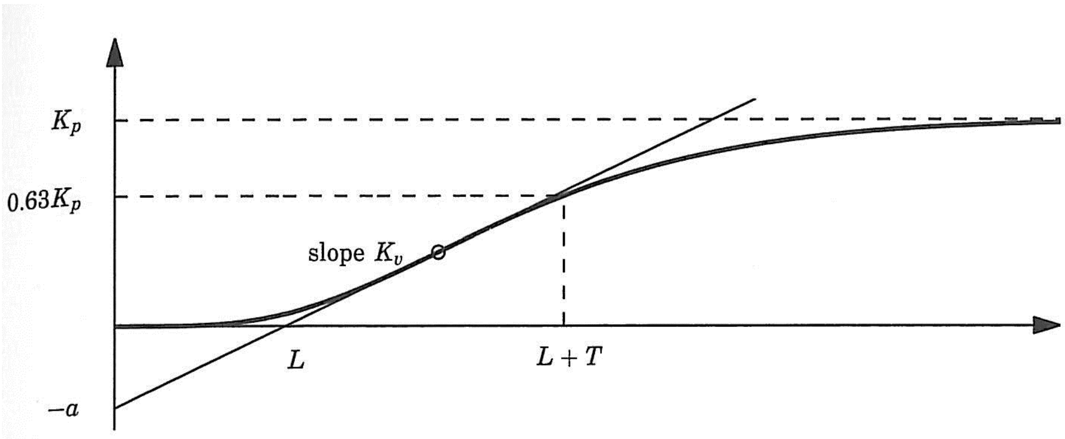


Figure 4.1. *Unit step response of a process and the procedure to determine the parameters K_p, T and L of a FOTD model. (Åström & Hägglund, 2006)*

K_p is calculated by dividing the change of the output with the change of the input $K_p = \Delta y/\Delta u$. Time delay can be obtained at the point where the steepest tangent of the step response intersects the initial steady state value. Time constant T is obtained as the time where the value of the output has risen to 63% of the final value. The process delay is subtracted from this time.

$$T = T(0.63\Delta y) - L \tag{4.2}$$

An important variable describing how easy it is to control a process is the normalized time delay τ defined in (Eq. 4.3).

$$\tau = \frac{L}{T + L} \tag{4.3}$$

Values of a normalized time delay ranges from $0 < \tau < 1$. Processes with small τ are called lag dominated (i.e. time constant dominated), processes with τ close to 1 are called delay dominated and processes with τ around 0.5 are called balanced processes. (Åström & Hägglund, 2006)

Model structures, such as FOTD, can be fitted to the step response data by using modeling tools which use some optimization procedure. (Åström & Hägglund, 2006) Matlab's Identification toolbox is one such modeling tool and it will be used during this project to obtain the process models. When using Identification toolbox, the input and output data were preprocessed before starting the analysis. Clearly disturbed data was removed before inserting the data to the software because it will cause errors in the modeling. Means of both data sets have to be removed and the data is split into evaluation and validation data.

4.2 Controller tuning

Controllers that were used when regulating the process variables during the experimental runs were PID controllers. The transfer function of a PID controller is presented in (Eq. 4.4)

$$C(s) = K\left(1 + \frac{1}{T_i s} + T_d s\right) \tag{4.4}$$

where K is the controller gain, T_i is the integration time constant and T_d is the derivation time constant. Eq. 4.5 defines how the output of the controller u is formed

$$u(s) = K\left(1 + \frac{1}{T_i s} + T_d s\right) \cdot e \tag{4.5}$$

where e is the error variable which is the difference between the setpoint and the process output.

Tuning of the controllers was done by conventional Ziegler-Nichols method, more recent AMIGO method (Åström & Hägglund, 2006) and the method proposed by Åström and Murray (2009). Also lambda tuning method was utilized in one experiment. In order to use any of these methods step responses from the process are required.

4.2.1 Ziegler-Nichols step response tuning method

In Ziegler-Nichols step response method, the step response of an open-loop system defines the parameters a and L by which the parameters of a PID controller are calculated. Since there are only two parameters that characterize the process, the process model is very simple. These parameters are obtained from the step response by drawing a tangent to the point where the slope of the step response has its maximum (derivative) and the points where this tangent crosses the coordinate axes give the parameters a and L as illustrated in Fig. 4.2.

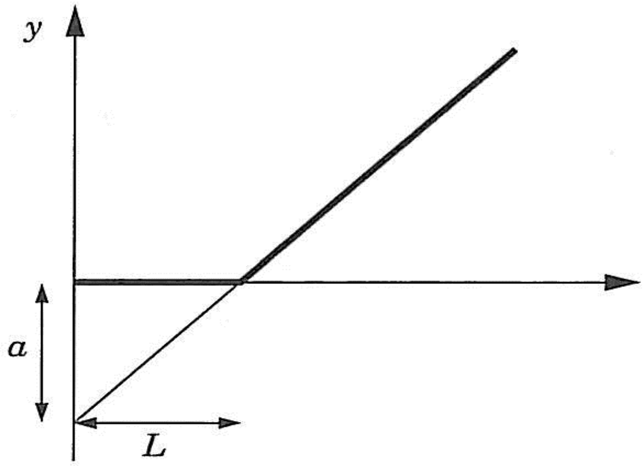


Figure 4.2. *Characterization of a step response in Ziegler-Nichols step response method. (Åström & Hägglund, 2006)*

Modeling that is conducted this way is said to be modeling by an integrator and a time delay. Ziegler and Nichols give the PID-parameters as functions of these two parameters. Tuning rules are shown in Table 4.1. (Åström & Hägglund, 2006; Ziegler & Nichols, 1942)

Table 4.1. Ziegler-Nichols step response tuning parameters. (Ziegler & Nichols, 1942)

Type	K	T_i	T_d
P	1/a	-	-
PI	0.9/a	3L	-
PID	1.2/a	2L	0.5L

Ziegler-Nichols method often gives quite heavy controller gains because it aims to maximize the integral gain which may lead to stability problems. Additionally, this method is based on linear control theory which can cause problems with nonlinear processes. These facts in mind, it is quite unlikely that this tuning method will produce suitable controller parameters for the nonlinear combustion process in question. However, this tuning method is closely related to the AMIGO method (see Chapter 4.2.2) and that is why it presented here.

4.2.2 AMIGO step response tuning method

AMIGO (Approximate M-constrained Integral Gain Optimization) tuning method (Åström & Hägglund, 2006) is based on maximizing the integral gain just as Ziegler-Nichols' method. By doing so, load disturbances should be minimized but since too high integral gain can cause oscillatory behavior, poor robustness or instability, there is a robustness constraint added into AMIGO. This robustness constraint is defined as a function of the sensitivity function *S* and the complementary sensitivity function *T*.

Sensitivity function *S*, defined in Equation 4.6, reflects the properties of the feedback system, such as disturbance attenuation and robustness to process variations. On frequencies ω disturbances are either amplified ($|S(i\omega)|>1$) or attenuated ($|S(i\omega)|<1$) by feedback control. Maximum sensitivity M_s tells the worst-case amplification of the disturbances. M_s is a function of sensitivity function *S*, defined in Equation 4.7.

$$S = \frac{1}{1+P(s)C(s)} = \frac{1}{1+G_l(s)} \tag{4.6}$$

$$M_s = \max|S(i\omega)| = \max\left|\frac{1}{1+P(i\omega)C(i\omega)}\right| = \max\left|\frac{1}{1+G_l(i\omega)}\right| \tag{4.7}$$

P is the transfer function of the process and *C* is the transfer function of the controller. Complementary sensitivity function *T* defines the largest variation the process can have while still maintaining its stability (Equation 4.8). The stability of the system is maintained if the condition in Equation 4.9 is fulfilled. This condition implies that as long as *T* is small, large relative perturbations to the process are allowed.

$$T(s) = \frac{P(s)C(s)}{1+P(s)C(s)} = \frac{G_l(s)}{1+G_l(s)} \tag{4.8}$$

$$\left|\frac{\Delta P(i\omega)}{P(i\omega)}\right| < \frac{1}{|T(i\omega)|} \tag{4.9}$$

A conservative estimate of the relative error permissible to the process transfer function is $1/M_t$ where M_t

$$M_t = \max |T(i\omega)| = \max \left| \frac{P(i\omega)C(i\omega)}{1+P(i\omega)C(i\omega)} \right| = \max \left| \frac{G_l(i\omega)}{1+G_l(i\omega)} \right|, \qquad (4.10)$$

is the largest magnitude of |*T*|. For a system with error feedback M_t indicates also the largest gain of the transfer function from the setpoint to the output.

Joint sensitivity *M* is defined as a variable equal to the maximum sensitivity M_s and the complementary sensitivity M_t.

$$M = M_s = M_t \qquad (4.11)$$

The robustness constraint in AMIGO is defined by the joint sensitivity and the joint sensitivity that the rules provide is smaller than 1.4.

When the AMIGO tuning method was developed the integral gain of a PID controller was evaluated by an optimisation algorithm, considering the mentioned robustness constraint, to a large number of processes that had essentially monotone step response. This test batch withheld several different processes from lag dominated to delay dominated processes. The results of the optimisation algorithm were fitted to process data in order to obtain formulas which give controller parameters as functions of process parameters.

Another characteristic of AMIGO that differs from Ziegler-Nichols method is the fact that it requires more process information. When using AMIGO method the step response is approximated by a first order transfer function (FOTD) characterized by three parameters static gain K_p, delay of the process *L* and time constant *T*. (see Eq 4.1) So instead of Ziegler-Nichols' two parameters *a* and *L*, AMIGO requires three parameters. The tuning rules given by AMIGO step response method for PI and PID controllers are presented in equations 4.12–4.16.

PI

$$K = \frac{0.15}{K_p} + \left(0.35 - \frac{LT}{(L+T)^2}\right) \cdot \frac{T}{K_p L} \qquad (4.12)$$

$$Ti = 0.35L + \frac{13LT^2}{T^2 + 12LT + 7L^2} \qquad (4.13)$$

PID

$$K = \frac{1}{K_p} \cdot \left(0.2 + \frac{0.45T}{L}\right) \qquad (4.14)$$

$$T_i = \frac{0.4L + 0.8T}{L + 0.1T} \cdot L \qquad (4.15)$$

$$T_d = \frac{0.5LT}{0.3L + T} \cdot L \qquad (4.16)$$

This tuning method can give high controller gains for processes that are lag dominated. However, since there is a robustness constraint based on joint sensitivity in this tuning method, the nonlinearities in the controlled processes will not cause so severe problems when tuning the controllers. Thus, this tuning method is more suitable for the process in question than for example the previous Ziegler-Nichols method.

4.2.3 Åström & Murray's tuning method

Tuning method proposed by Åström and Murray (2009) requires also the same three parameters of the first order transfer function as AMIGO and thus it also requires a step response experiment. The method is based on the tuning method development presented by Åström and Hägglund in 2006. Generally it gives lower controller gains than the Ziegler-Nichols method. The rules proposed are as follows. (Åström & Murray, 2009)

PI
$$K = \frac{0.15L + 0.35T}{K_p L} \tag{4.17}$$

$$ki = 0.46L + \frac{0.46L + 0.02T}{K_p L^2} \Rightarrow Ti = ki / K \tag{4.18}$$

In comparison to AMIGO this method gives slightly higher controller gains which might occasionally lead to stability problems. However, this method still gives significantly more modest controller gains than Ziegler-Nichols and thus it may provide suitable controller parameters for this process also.

4.2.4 Lambda tuning method

In lambda tuning method the process is modeled by the FOTD model (see Equation 4.1). This method is a special case of the pole placement method. When using a PI controller the controller parameters are chosen so that the integral time constant T_i is equal to time constant T of the process. By doing so the process pole is cancelled and the performance of the closed-loop system is thus characterized by the closed-loop pole T_{cl}. The simple tuning rules given by the lambda method are as follows. (Åström & Hägglund, 2006)

$$K = \frac{1}{K_p} \frac{T}{L + T_{cl}} \tag{4.19}$$

$$T_i = T \tag{4.20}$$

Since the T_{cl} is the only design parameter the choice of the value is of extreme importance. In order to obtain a robust controller a common rule of thumb is to choose $T_{cl} = 3T$ and for an aggressive controller $T_{cl} = T$. (Åström & Hägglund, 2006) Since the dynamics of the process in question are quite heavy it is reasonable to tune the controller according to the robust rule to avoid problems with stability.

4.3 Feedback control of oxygen in flue gas

Oxygen or lambda probe situated in the flue gas duct provides information about the amount of oxygen in flue gas. Since lambda probes have become more common in today's state-of-the-art small-scale boilers it is possible to use feedback control of the oxygen excess in the flue gas, thus also controlling the composition of the flue gas. (Eskilsson et al. 2004) However, in this project instead of lambda probe the oxygen was measured by a gas analyzer (see Chapter 3.2). When compared to the dynamics of a lambda probe, the gas analyzer used has heavier dynamics and longer measurement delay which is due to different operating principle. Since gas analyzers are more versatile instruments than lambda probes they are also more expensive and due to that they are not an economically reasonable choice for small-scale boilers.

In this chapter feedback control enabling control of the oxygen in the flue gas is introduced. As it could be seen in Fig. 2.1 when aiming for minimization of emissions there is trade-off between NO_x and unburned gases (CO and OGC). Additionally, the amount of air fed to the system affects the efficiency that is gained from the boiler. Too high air feed lowers the efficiency as the residence time of the flue gases decreases and thus hotter flue gases exit the boiler. Also the short residence time results in higher emissions as the pyrolised gases have less time to react with oxygen in the boiler. Too high air feed also starts to increase NO_x emissions. Lowering the air feed causes an increase to the efficiency until a point is reached where the losses due to unburned gases exceeds the efficiency improvement gained by lowering the air feed. Thus the excess oxygen in the flue gas should be kept between some optimum limits as illustrated in Fig. 4.3. (Eskilsson et. al., 2004; Šulc et al, 2007).

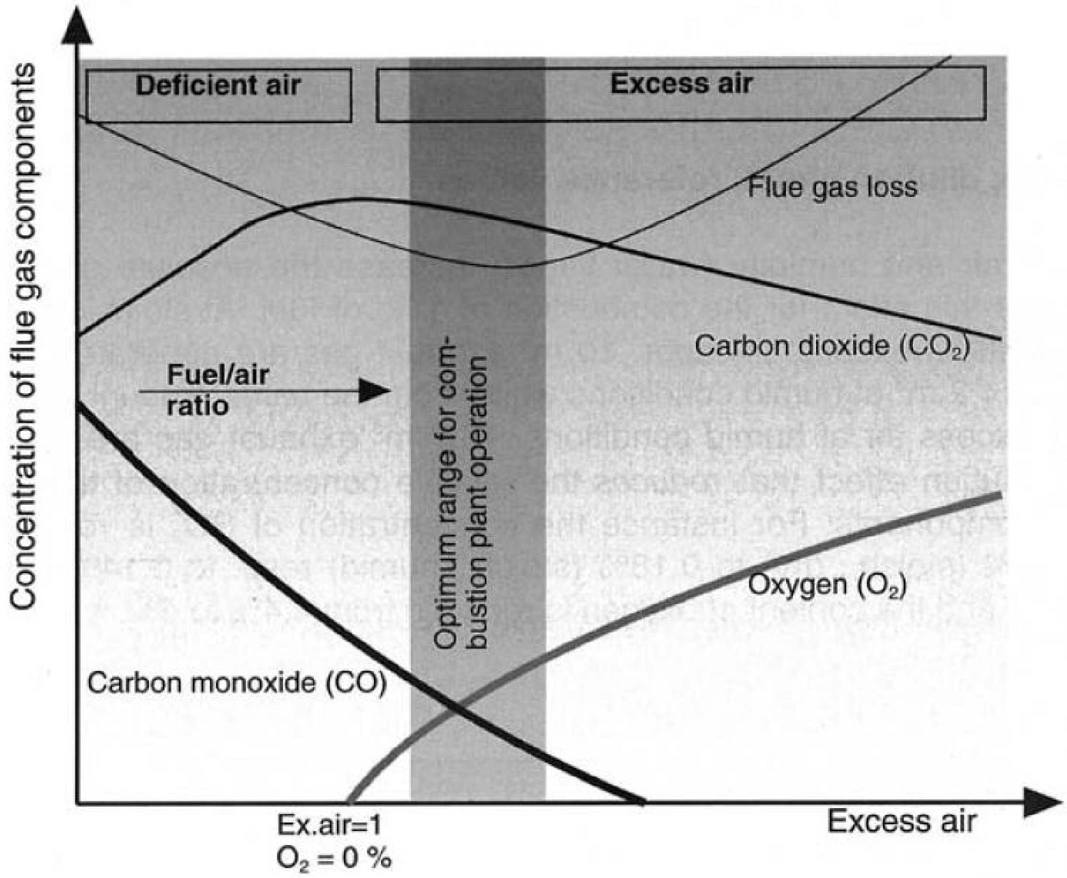


Figure 4.3. *CO, CO2 and flue gas losses as a function of excess combustion air. (Phoenix Instrumentation)*

Lower limit is the value of oxygen after of which a further reduction will cause too severe increase of CO emissions. Higher limit can be said to be the one where the CO starts to rise due to excess feed of combustion air causing the flame to cool down. From this point of view as low air feed as possible should be used. These optimum limits are case-specific so they have to be investigated individually for every boiler model. The limits depend also on the operating conditions such as power region etc. in the boiler. Basically, air feed should be maintained at such state that it always provides enough air to get the oxygen value in the flue gas to the lower limit but not exceeding this value. (Eskilsson et al., 2004) Naturally, there has to be some safety margin between the lower limit and the desired oxygen level given to the controller due to deviations in the process which can momentarily sway the process over the lower limit value thus causing higher emissions. This fluctuation in the process can be called short term steadiness of the boiler. The worse the short term steadiness the larger safety margin is needed thus lowering the efficiency of the boiler due to running the boiler with more excessive air feed. Thus the short term steadiness can be said to be directly linked to efficiency of the boiler.

4.3.1 Feedback control of oxygen by air feed

In the process in question there is only one fan providing both primary and secondary air. The division of these air feeds is done mechanically so the possibilities to control combustion are limited. This is because in case of fixed primary air/secondary air ratio it is not possible to use a typical secondary air control strategy quite common in larger boilers (Kovács & Mononen, 2007). This strategy is based on the principle that when the oxygen level goes down, the secondary air is increased in order to provide combustion air to the secondary combustion zone in order to burn all the pyrolised gases. The problem with this strategy and single fan case is that when air feed is increased to get increased amount of secondary air, also the amount of primary air is increased. This increased amount of primary air accelerates char combustion causing more pyrolysis and thus more unburnt gases. This leads again to the situation where more air should be fed to combustion chamber to burn all the gases and the same circle will start all over again. By doing the same thing in the opposite way and decreasing the secondary air in situation where the oxygen goes down would lead to incomplete combustion and more emissions because although the primary air is decreased causing char combustion and pyrolysis to slow down, there is not enough air to burn all the gases in the chamber. This kind of active disturbance compensation is thus out of the question in our case. (Korpela et al. 2009a, Korpela et al. 2009b)

Despite this fundamental obstacle, it is still possible to control the undesired slow drifting of the process by the air feed. This kind of control requires that the control is quite calm and slow so that it will not cause any further disturbances to the process when the controller starts to act on the drifting. The feedback control scheme that was used is illustrated in Fig. 4.4.

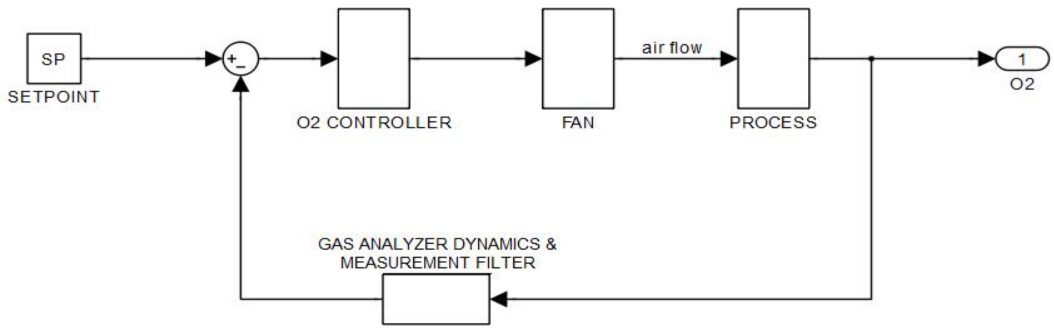


Figure 4.4. *Principle scheme of feedback control of oxygen by air feed.*

As it can be seen, the oxygen controller gives input to the combustion air fan according to the oxygen measurement from the process. The oxygen measurement has c. 23 s time delay due to the dynamics of the analyzer. The time delay was estimated in the correlation studies that were conducted. A suitable measurement filtering is presented in Chapter 5.2.

4.3.2 Feedback control of oxygen by fuel feed

Due to the obstacle mentioned in Chapter 4.3.1, another possibility to control oxygen excess is to control it by the fuel feed thus settling the fuel feed to a constant air feed. By conducting oxygen excess control this way, the desired heat output is actually defined by setting the amount of combustion air that is fed. (Korpela et al. 2009a; Korpela et al. 2009b)

The dynamics from the fuel feed to oxygen are slower than the ones with air feed, thus the performance that can be expected from this control structure may be slower. Due to slower performance, regulating out fast disturbances might be impossible by the means of fuel feed but slow drifting, which is quite common in processes in question, is possible to compensate. The feedback control scheme of the oxygen excess by the fuel feed can be seen in Fig. 4.5.

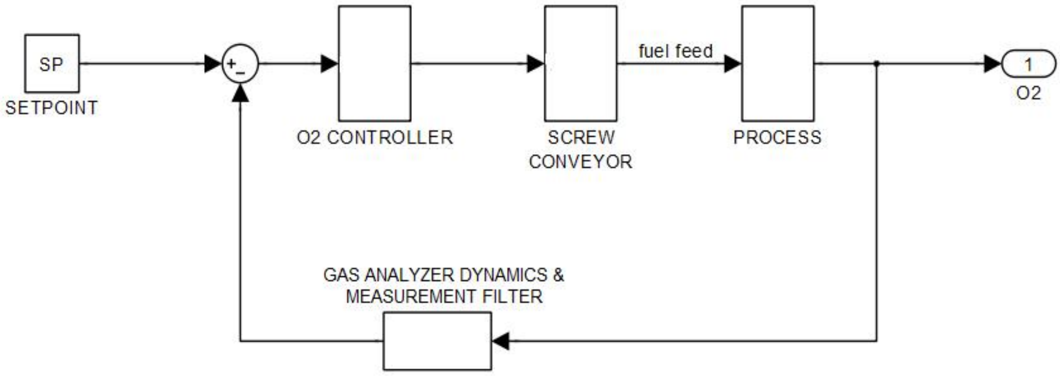


Figure 4.5. *Principle scheme of the feedback control of oxygen by fuel feed.*

The output of the controller is the fuel feed ratio f which is later transformed to the whole fuel feed period T_{whole} (see Equations 4.21 and 4.22).

$$f = T_{on} / (T_{on}+T_{off}) \tag{4.21}$$

$$T_{whole} = T_{on}+T_{off} = T_{on} / f \tag{4.22}$$

T_{on} is a constant 3 s period during of which the screw conveyor is feeding fuel and respectively T_{off} is the varying period when the conveyor is in idle state. The nominal power level corresponds to T_{whole} = 20 s. The output of the controller is constrained so that power level ranges from 50 % to 100 % of the nominal 25 kW output and thus the range of T_{whole} is from 40 to 20 seconds respectively. The reason for limiting the output of the controller was firstly to assure that the boiler would not be driven over its nominal power range and secondly that the fuel feeding algorithm in use did not allow T_{whole} periods smaller than 20 s.

Measurement filtering is again needed and it is presented in Chapter 5.2. The same 23 s time delay of the analyzer affects the control system.

Due to time limitations of the project the air feed to the system was not controlled thus making the situation so that the air fan was only run with a constant input. This means that the actual air flow to the system fluctuates as a function of draught, flow resistance and underpressure in the boiler. This makes it harder to control the excess oxygen in the boiler (Kovács & Mononen, 2007). Stabilization of the air flow with feedback control is proposed by Korpela et al. (2009b) and it would most probably result in better control results.

4.4 Feedback control of combustion temperature

Since pellets have uniform quality and moisture content, the heating power obtained from, for example, one pellet is relatively constant within the same set of pellets. (see Chapter 2.1) That is why quality changes do not cause significant problems in controlling pellet boilers. However, one typical disturbance in pellet boilers is that the degree of filling varies in the fuel supplying conveyor, for example feeding screw. This is due to momentarily uneven unloading of the pellets from the storage. Structure, volume and dimensions of the fuel storage have a major effect on the size of the disturbance. In addition, properties of the fuel feeding conveyor and possible equalizing equipment in the fuel storage have an impact on these disturbances. The disturbance causes a decrease in the heat power since less fuel is fed to combustion chamber than is desired. Due to smaller addition of fuel, the fuel pile is thus decreased and if the disturbance is severe enough, the combustion starts to decline causing a drop into the generated heat output. This drop can be detected by temperature measurement. (Korpela et al., 2009a; Korpela et al., 2008)

Since there is no feasible way of conducting fuel flow measurement in order to utilize feedforward measures for disturbance compensation, another way to compensate the disturbances from the fuel feed, is to utilize feedback control based on temperature measurement from the combustion chamber or near the combustion chamber. By utilizing this feedback control, stable combustion power could be obtained. (Korpela et al., 2008) Stable combustion power means also more stable behavior in the burner in general and thus, other important variables in the process act in more stable manner as well, making them easier to control. In Fig. 4.6 there is illustrated the feedback control of fuel feed based on the temperature measurement from the upper end of the combustion chamber.

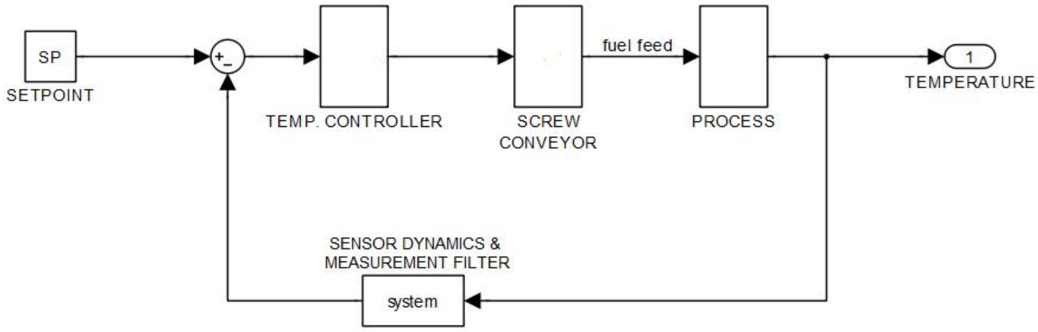


Figure 4.6. *Principle scheme of the feedback control of temperature by fuel feed.*

Air feed to the process is kept constant (i.e. air fan input is constant) when this control loop is used. Temperature in the upper end of the combustion chamber is dependent on many variables so finding a reasonable setpoint for this feedback control might be tricky. One way of solving this problem is to utilize cascade control of the process as presented by e.g. Korpela et al. (2008) and which is presented in Chapter 4.5. Temperature measurement is very fast so practically no time delays from the measurement are affecting the control system. Still some filtering of the measurement signal was needed and suitable filter is presented in Chapter 5.2.

As earlier, the output of the controller is the fuel feed ratio f which is then transformed to the whole fuel feed period T_{whole} (see Equations 4.21 and 4.22).

4.5 Cascade control of oxygen

The oxygen concentration in the flue gas can be indirectly controlled with the previous temperature control scheme. When the air feed to the boiler is kept constant, setting a high temperature setpoint will lead to a lower oxygen concentration in the flue gas and vice versa. So as long as a suitable temperature setpoint can be found, the process is always run with a proper oxygen excess. However, since there are several factors affecting the temperature, a feasible way to find a suitable temperature setpoint is to utilize a cascade control structure illustrated in Fig. 4.7. (Korpela et. al, 2009a)

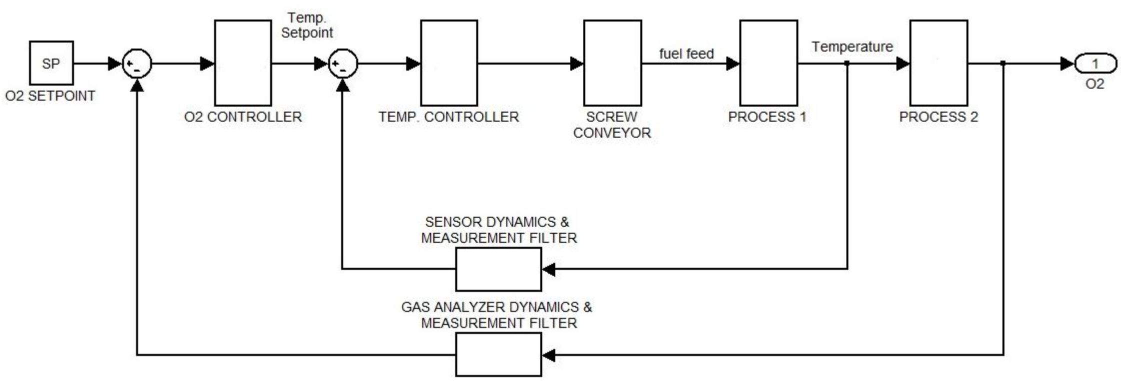


Figure 4.7. *A cascade control structure for oxygen control.*

The idea of the structure is that the primary controller, controlling the oxygen concentration in the flue gas, gives a suitable setpoint to the secondary controller controlling the temperature. The temperature setpoint is decreased if there is a lack of oxygen in the process and vice versa. Thus the process can always be driven with a proper amount of oxygen. If the power level needs to be changed the air feed is increased or decreased and the cascade control structure will adjust the fuel feed to the air feed. (Korpela et. al, 2009a) Since this control structure controls also the temperature, the sudden fluctuations in the fuel feed are also compensated if they occur as was discussed in the previous control scheme. By utilizing this control structure, the problems caused by the nonlinearities of the process (operation point, fouling etc.) can be avoided thanks to the cascade structure of the control system.

Cascade control is especially useful if there are heavy dynamics in the process which is often the case with combustion. The dynamics of the inner loop (includes the secondary controller) should be faster than the ones of the outer loop. According to a common rule of thumb the average residence times of the inner and outer loop should have a ratio of at least five. The tuning of the controllers is conducted so that the secondary controller is tuned first using for example the methods presented in Chapter 4.2 and after that the primary controller is tuned according to a rule of thumb which proposes that the integration time constant of the primary controller (PID) should be 5 times longer than in the secondary controller. (Åström & Hägglund, 2006)

4.6 Grate sweeping compensation

Grate sweeping which is done in order to move fuel and ash towards the combustion chamber and ash bin causes clear disturbances in the combustion process. (Korpela & Björkqvist, 2008; Šulc et al., 2009) Grate sweeping causes disturbances because the sweeping movement stirs the bed so that the primary air flows through the bed in different routes than before the movement. Also the resistance of primary air flow varies due to new flow channels. If the grate sweeping causes an increment to primary air flow, due to a decrease in the flow resistance, it can cause accelerated burning of char on the grate which leads to a higher combustion temperature which in turn increases the amount of pyrolized gases. Another factor leading to an increase in pyrolysis speed is the fact that fresh fuel is revealed from the fuel bed as the sweeping movement is done. (Korpela et al., 2008; Korpela & Björkqvist, 2008)

If the sweeping is not compensated in any way, the events described cause a peak in emission levels since there usually is not enough secondary air to burn all the gases from the accelerated pyrolysis (see Chapter 2.2.3). Also there will be a slight decrease in the efficiency of the boiler since there will be more CO and OGC emissions in the flue gas, although the decrement is often quite small. The size of the peak is dependent on the size of the fuel pile. (Korpela & Björkqvist 2008) In our case, the peaks of CO level occur tens of seconds after the grate sweeping. (see Fig. 4.7)

Since there is quite a long delay it would be possible to do compensative measures (i.e. utilize feedforward control) such as to decrease the primary air and simultaneously increase the secondary air after each grate sweeping event. (Korpela & Björkqvist 2008) But since the boiler in question is equipped with only one fan it is not possible to do these compensations by fan control which would be possible in the case of two separate fans. In this case the ratio of the air feeds is controlled by adjusting a valve that is dividing the primary and secondary air.

Thus in our case one possibility of increasing the secondary air would be to manipulate the air ratio valve but even if the air ratio valve had been on-line controlled, compensation by manipulating the ratio would have been difficult to arrange. This is because, firstly, the nonlinearity of the valve would have made the control implementation cumbersome. Second reason is that the air ratio valve choked primary air feed instead of increasing the secondary air feed the same amount. As it can be seen in Chapter 5.1, the total air feed got choked due to growth in the resistance of the flow to the boiler as the primary air hole was choked. So even though the proportion of the secondary air of the whole air feed grew, the amount of the growth is impossible to measure since there was no flow measurement of the secondary air feed. In the worst case the compensative actions, which would have choked primary air after grate sweeping, could have caused even more severe disturbance as the oxygen level would have actually dropped more due to a decrease in the total air feed.

Since the air ratio bolt was not online controllable, it made the situation so that instead of using sophisticated control measures to compensate the grating disturbance, the amplitude and period of the grate sweeping was altered from the original in order to decrease the effect of the disturbance. In the original setup the grate sweeping was done once in 10 minutes and the amplitude of the movement was c. 30 mm. In earlier studies, conducted at CTU by Viktor Plaček, this cycle was altered so that the amplitude of the movement was decreased to 11 mm and the movement period changed to 2 minutes. Due to the smaller amplitude of the movement, the disturbance caused by the grate sweeping decreased significantly and due to the higher frequency of the grate sweeping the fuel bed was still maintained sufficiently even and low.

The results obtained with this mechanical altering of the grate sweeping are illustrated in Fig. 4.7, which was made for the IFAC article by M.Sc. Viktor Plaček.

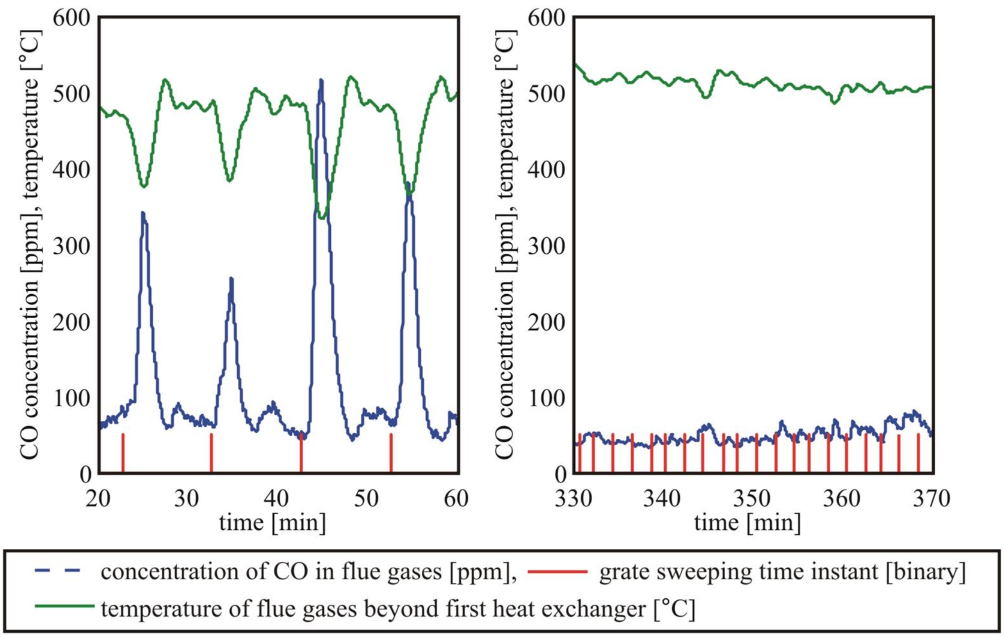


Figure 4.7. *Comparison of grate sweeping effects on the combustion process with the original and the altered grate sweeping sequence and amplitude.*

As can be seen in Fig. 4.7, the improvement is significant. The CO peaks have almost disappeared and the temperature after the first heat exchanger is remarkably more stable than before the alteration. This improved grate sweeping sequence was used in the rest of the trial runs carried out in this work.

5 DESIGN AND IMPLEMENTATION OF COMBUSTION CONTROL

There were a set of experiments done with the boiler in order, first of all, to get knowledge about the boiler in question, then to carry out step response tests to obtain the dynamics of the boiler. These dynamics were used in creating process models for control design purposes and tuning the PID controllers. In the experiments after these, the designed control systems were tested.

5.1 The position of the air staging valve

The purpose of the experiment was to examine the effect the valve dividing the primary and the secondary air has on the combustion and also to find a good position for it for the next experiments. The size of the fuel bed can be controlled by manipulating the primary air feed and in this case the ratio of the primary and secondary air. A suitable amount of primary air was searched by finding a suitable position for the air ratio valve. This position would be a position where the fuel bed covers almost the whole grate (no pellets falling off the grate) and is not blocking any of the secondary air wholes (i.e. is not too thick).

The valve was a round disc attached to the tip of a bolt (see Fig 5.1) and the position of it could be changed by rolling the bolt.

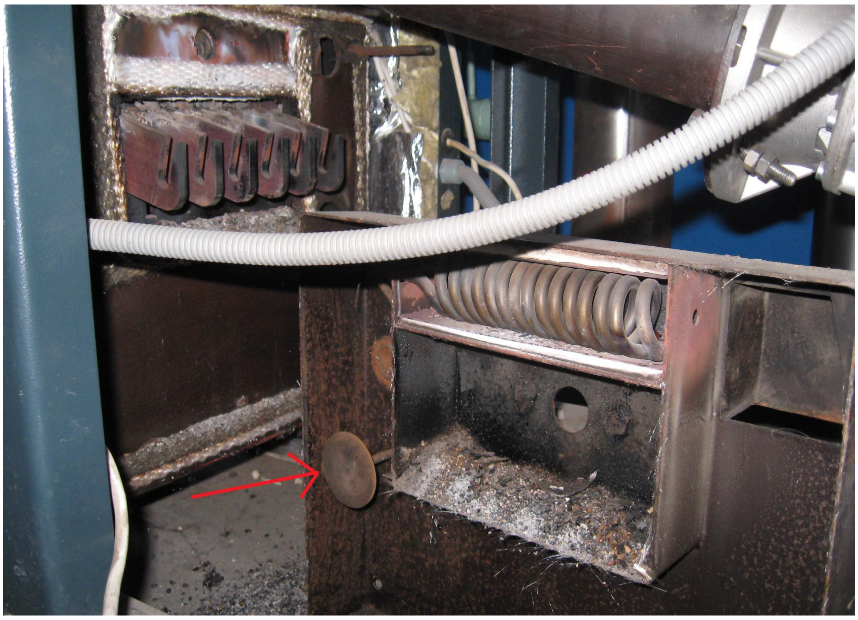


Figure 5.1. *Dismantled back of the boiler. In foreground there is the back wall of the boiler where the air ratio valve (marked with the red arrow), ignition resistor and air fan (on the opposite side) are located. In the upper left part there can be seen the grate's end, primary and secondary air holes.*

The disc-like shape of the valve caused severely nonlinear working characteristic. The changes to the position of the bolt were made from larger to smaller value (i.e. closing the bolt) to see the actual effect to the fuel bed height. This way the large growth of the fuel bed with smaller values of the bolt (less primary air) which would still affect the next change(s) could be avoided. This would occur when doing the changes in the opposite direction.

The primary/secondary air division bolt was tested by making the following changes to its position: 18, 13, 9, 6, 4 and 3 revolutions from the closed position. 18 revolutions was the maximum value that the bolt turned.

As it can be seen in the experiment data illustrated in the Fig. 5.2, in the beginning the excess O_2 stabilized to c. 11.4 %. When the air bolt was closed to 13 revolutions, the O_2 started to go down slowly and in the end of the step the value was c. 10.8%.

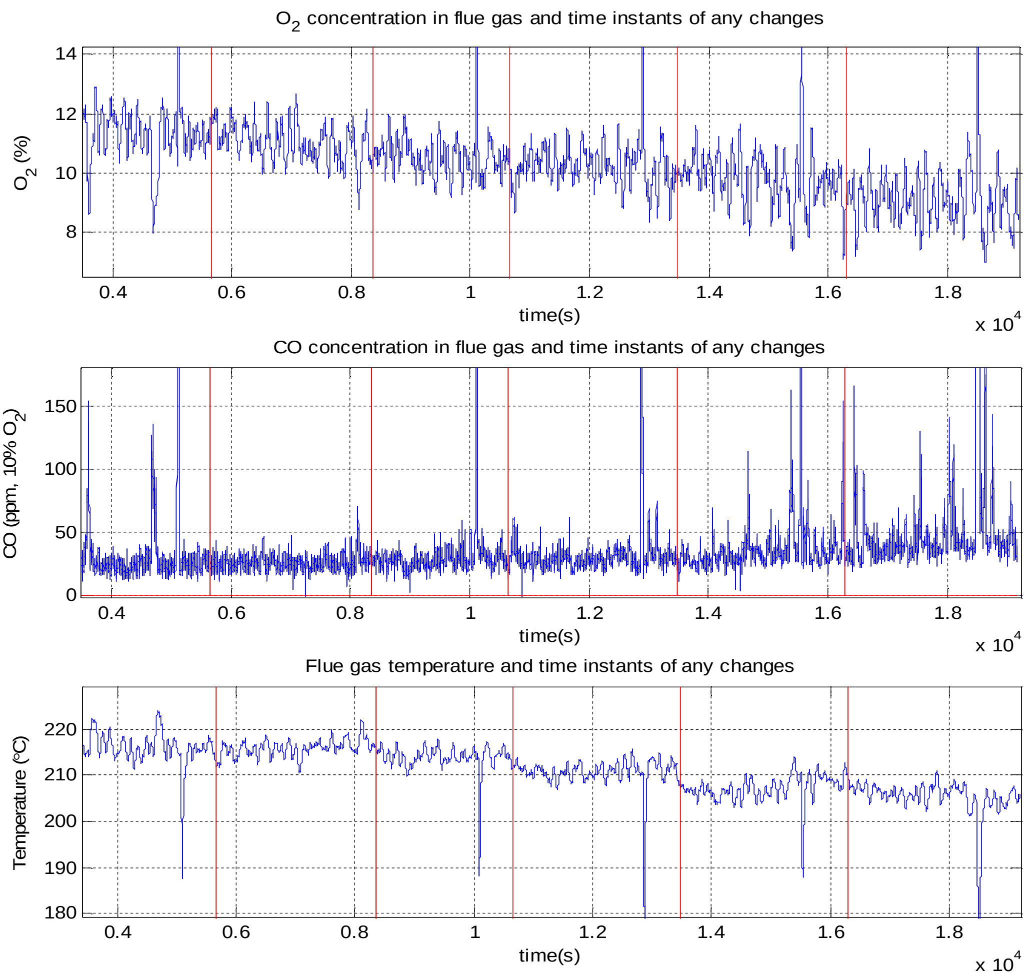


Figure 5.2. *Up: Behavior of oxygen excess (O_2); Middle: Carbon monoxide; Down: Temperature of the flue gas. The times of pri/sec bolt change are indicated by red vertical lines. Values of the bolt were 18, 13, 9, 6, 4 and 3 revolutions from the closed position.*

Similar trends occurred during the steps with 9 revolutions and 6 revolutions but after that the O_2 started to fall more clearly and at the end of the experiment the excess oxygen value was c. 9%. This kind of trend in the excess oxygen values can be explained by firstly the decreased amount of air the radial fan is producing as the primary air hole is closed. Clearer declines towards the end can also be explained by the fact that in the end the amount of primary air was so small that the fuel bed started to grow significantly and this greater amount of fuel on the grate caused larger resistance to the primary air flow in the boiler thus reducing further the air feed into the boiler. The sharp peaks and drops in the Fig. 5.2 are due to opening the combustion chamber door for taking photos of the fuel bed.

During the first 4 steps (positions 18, 13, 9 and 6) the carbon monoxide concentration in the flue gas was on a modest level and no larger peaks emerged, as it can be seen in the middle of Fig. 5.2. This is due to sufficient amount of combustion air in the combustion chamber. As the amount of combustion air started to decline towards the end more clearly, CO concentration also started to rise due to lack of oxygen. Also severe peaks, reaching over 100 ppm, started to appear. The behavior of CO correlates well with the behavior of O_2 during the experiment. The growth in the fuel bed may also have some impact on the CO behavior.

Changes in the amount of combustion air caused by the manipulation of the air ratio valve, can be seen clearly in the flue gas temperature. The nonlinear effect of closing the valve can be seen in lowest figure in Fig. 5.2. As the first change from 18 to 13 revolutions had almost no effect at all to the flue gas temperature, but the change from 6 to 4 revolutions caused a decline of 4 ˚C to the temperature.

Fuel beds in the combustion chamber after the 18 revolutions and the 9 revolutions step are illustrated in Fig. 5.3.

Figure 5.3. *On the left hand side, fuel bed in the combustion chamber after the 18 revolutions. On the right hand side, fuel bed in the combustion chamber after the 9 revolutions.*

The situation in the combustion chamber looks good just before closing the primary-secondary air bolt to 13 revolutions. As it can be seen on the left hand side in Fig. 5.3, there still is some space left on the grate where there is no pellets. This should indicate higher efficiency since no unburnt pellets are falling off the grate to the ash bin. Also the size of the fuel bed looks quite reasonable. The size of the fuel bed grew during the 13 and 9 revolutions steps but as it can be seen on the right hand side of Fig. 5.3, there is still some space left between the edge of the grate and the edge of the fuel bed. So none or only few pellets should fall off the grate. The size of the bed is not too large yet since all the secondary air holes are still well above the fuel bed.

Situations in the combustion chamber after the 6 revolutions, 4 revolutions and the 3 revolutions step are illustrated in Fig. 5.4.

Figure 5.4. *On the left fuel bed after the 6 revolutions step; in the middle, fuel bed after the 4 revolutions step; on the right fuel bed after the 3 revolutions step.*

After closing the air bolt to a value of 6 revolutions, the fuel bed grew further and as it can be seen on the left in Fig. 5.4, there are pellets quite close to the edge and few unburnt pellets have already fallen off the grate. In the picture these unburnt pellets are still glowing in the ash bin. The size of the fuel bed has grown and it has almost reached the lowest secondary air holes which is not a desirable situation.

On the middle in Fig. 5.4 it can be seen that by closing the air bolt further on, the size of the fuel bed continued to grow and finally the pellets started to fall off the grate in larger numbers. This decreases the efficiency as the amount of unburnt fuel increases. Similar progress continues during the step where the air ratio valve was closed to 3 revolutions, illustrated on the right hand side of Fig. 5.4.

Based on this experiment it was decided that a good position for the air division bolt is nine revolutions from the closed position. This position was chosen because the combustion quality during that particular step was quite good since the CO concentration of flue gas was on a modest level and also no significant peaks emerged during this step. Also all the other variables according to the data were on good levels. According to the pictures taken from the combustion chamber, after this nine revolutions step, the size of the fuel bed started to grow so great that pellets started to fall off the grate and lowest secondary air holes got blocked by the pellets.

During this experiment it could be seen that making conclusions solely based on the measurement data can lead to wrong conclusions because the data does not give any information of the actual size of the fuel bed in the combustion chamber. In this case bolt positions smaller that 9 would have been reasonable choices since the combustion according to the data is still working quite well but after a longer period the fuel bed would have grown so great that the air feed to the boiler would be so blocked that there would not be enough combustion air. Also a lot of unburnt pellets would have fallen out of the grate. In the case of this particular boiler, the limiting factor for reducing primary air is the size of the grate since the first and the most severe consequence of reducing primary air is the growth in the amount of unburnt fuel. It has to be taken into account that if the properties of the fuel change notably, it may be necessary to evaluate the position for the air ratio valve again.

5.2 Process disturbances and filtering

During the experiments focused on the air division it was noticed that there are significant disturbances affecting the combustion process. The disturbance caused by the grate sweeping could be greatly reduced by mechanical means as described in Chapter 4.6, but there still seemed to emerge disturbances related to the grate sweeping as well as other variations. This can be seen in Fig. 5.5 which illustrates the data from an experiment where the grate sweeping was stopped in order to see the effect it has.

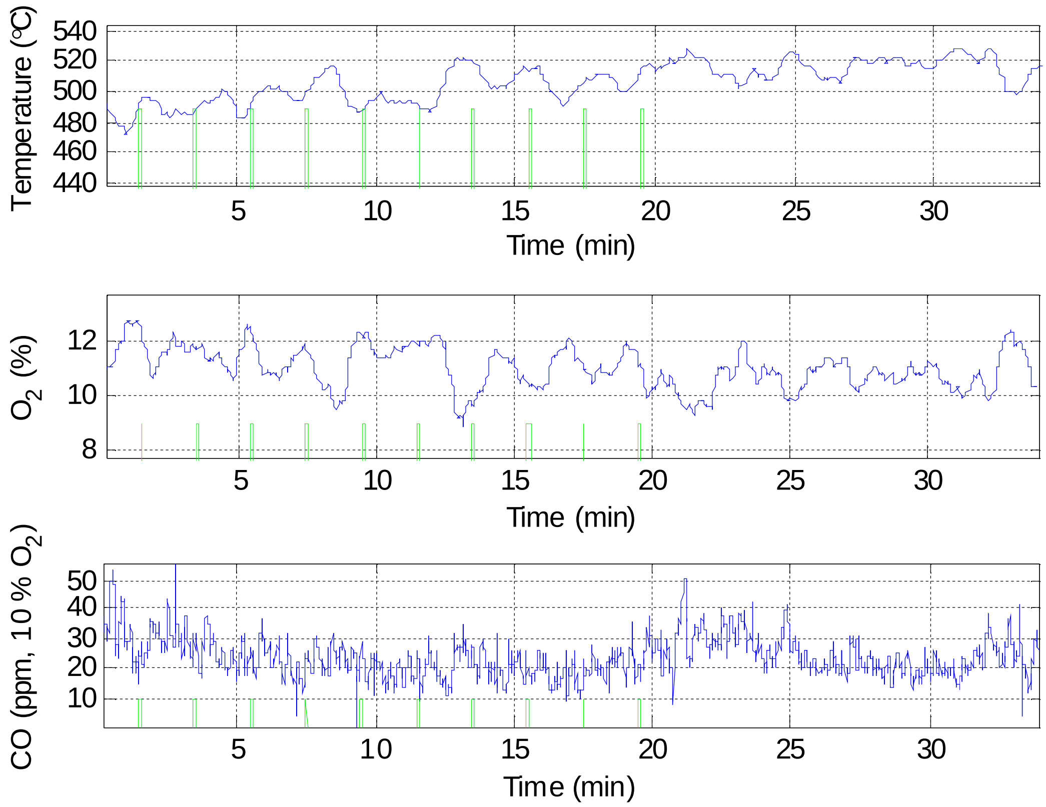


Figure 5.5. *Temperature, O_2 and CO behavior. Green vertical bars indicate the time instants of grate sweeping.*

Even though the grating was stopped, the process variables keep on fluctuating on a slightly lower level. Grate sweeping causes disturbances that seem to occur almost every 120 seconds but not every time as can be seen also in Fig. 5.5. Usually the grate sweeping was followed by a rise in temperature but sometimes it is followed even by a drop. Based on correlation analysis made by M.Sc. Korpela for the IFAC article, it was concluded that grate sweeping is not the main reason for fluctuation in the process.

Smaller periodical deviation in the variables is due to the fuel feed since the disturbance occurs on 20s interval which is equal to the fuel feeding periods during the experiment. This could be seen also in the cross correlation studies between different variables in which similar deviation occurs in interval of c. 20 seconds. Every addition of fuel causes a disturbance to the process because new, cold and moist fuel is added. Warming and vaporization of the water requires heat and thus the addition causes a slight drop to the temperature in the combustion chamber after of which the combustion accelerates as the fresh fuel starts to burn raising the temperature. Nevertheless, the disturbance caused by the fuel feed was insignificant compared to the other disturbances. Thus it did not set any requirements for the filter designing at all.

After the grating was switched off, (c. 1500 s in Fig. 5.5) the oxygen level kept on having peaks. Since the draught is free in the boiler, the pressure difference between the laboratory and the atmospheric pressure can vary, thus causing disturbances. But since the disturbances seemed to last tens of seconds, another explanation could be that mechanical ventilation in the building where the laboratory is situated causes pressure changes causing disturbances to the boiler. The possibility that ventilation is causing the disturbances could not be ruled out due to limitations to shut down ventilation momentarily. However, the correlation analysis revealed that the disturbances correlated poorly with the air flow so it is quite unlikely that the disturbances would have been due to the pressure variation.

Another possible cause of the disturbance is that the process itself generates random disturbances. This kind of disturbances might be caused by random acceleration of combustion at some point on the grate raising the temperature which in turn accelerates pyrolysis further on. After a while the preconditions for such an accelerated combustion disappear and the combustion fades to a normal level. Since the grate is quite large withholding a large number of pellets, the possibility of such random occurrences is higher than in the case of smaller grate. Nevertheless, the fluctuation in the fuel feed could cause more severe disturbances to the combustion if the grate size was decreased due to lesser amount of energy stored on the grate. Thus, there is a clear tradeoff between the combustion performance of a smaller and larger grate.

Due to these heavy and partly unidentified disturbances, filtering of the measurement signals was needed. The same disturbances can be seen in both variables; oxygen concentration and temperature in the upper end of the combustion chamber. So it was reasonable to use the same filtering with both variables. Since it was quite clear that grate sweeping causes disturbances occurring on 1/120s frequency, it was concluded that the filter should be able to filter effectively such low frequency disturbances. The unidentified, long lasting, disturbances also required heavy filtering. When setting the requirements of the filter, the time delay of the gas analyzer and the disturbance from the fuel feed were not nearly as important factors as the heavy disturbance from the grate sweeping and the long lasting unidentified disturbance.

A few different filters were tested. The sample rate of the whole system was 5 Hz which leaded to large buffers in the filters. The first was a running mean filter with time window of 150s (750 samples) carried out by a discrete transfer function. This filter produced a weighted running mean as presented in (Eq. 5.1)

$$y_{filtered}(t) = \frac{749}{750} \cdot y_{filtered}(t-1) + \frac{1}{750} y(t), \qquad (5.1)$$

which is not equal to a genuine running mean filter. The discrete transfer function is presented in Appendix 1. The second tested filter was a running median filter with time window of 120s (600 samples). This filter was carried out by a Matlab script which is also presented in Appendix 1. The filter sorts the samples feeded into it as a sorted array and gives as an input the value in the middle (in the case of 600 samples the middle value is the one in the 300th cell of the array. The next filter was a genuine running mean filter and it was tested with time windows of 120s (600 samples) and 150 s (750 samples). These filters were also carried out by a Matlab script which is presented in Appendix 1. These filters have heavy dynamics that affect the control system in a slowing way.

5.3 Step response experiments and identification

Step response tests were conducted for two inputs, the air feed and the fuel feed in order to provide process data for the identification. The oxygen level in the flue gas and temperature after the first heat exchanger (T2) were chosen as the output variables. This particular temperature measurement was the fastest and the most representative temperature measurement in the combustion chamber. It provides information about the intensity of the combustion and gives an indication of the amount of fuel that is fed. It would be beneficial to use a temperature sensor as close to the combustion chamber as possible. But to avoid the disturbing effect of radiation, the temperature was chosen to be measured from the upper end of the combustion chamber, after the first heat exchanger. It was the thermocouple situated closest to the combustion chamber without seeing the flame (i.e. no radiation).

First order transfer functions with a time delay (FOTD) were fitted to the step response data of the chosen variables. Since it was concluded in Chapter 5.2 that heavy filtering was necessary due to the slow but severe disturbances, the raw step response data of the control variables (output) were filtered before fitting the transfer function into it. Filtering was conducted by running the raw step response data through a filter with desired dynamics and after that the FOTD model was fitted to this new filtered step response. This was done because the controller can only act on the filtered process measurement signal so the dynamics are different than in the case of unfiltered measurement. Since several filters were tested, new process models had to be calculated after changing the filter. All the calculated process models and filters that were used are gathered in Tables 5.5 – 5.7 in Chapter 5.5.

It has to be acknowledged that process behavior on higher frequencies was lost as the process identification was done based on step responses that were heavily low-pass filtered. However, the loss of the higher frequencies did not play a major role in this case due to the generally slow behavior of the process. The slow behavior can be seen in all of the identified time constants and delays throughout Chapter 5.3 as well as in the process models presented in Tables 5.5 – 5.7. Additionally, most of the behavior on higher frequencies is due to disturbances and natural swaying of the process and all this behavior is undesirable from control point of view since active disturbance compensation was out of question. After suitable models were obtained, the controllability of the processes was evaluated by comparing the normalized time delays of the processes (see Eq. 4.3).

Additionally, it has to be acknowledged that the dynamics of a process obtained in step response experiment are valid only in that operating point. Since the process in question is very nonlinear, the dynamics in some different operating point can be very different. Thus the process dynamics throughout the whole power range should be investigated by conducting a series of step experiments to obtain valid process models. The power range should also be covered by both downwards and upwards steps because the dynamics can be different depending also on the direction of the step. However, due to time limitations of the project such extensive step experiment series were not conducted.

5.3.1 Step response to air feed change

Step response test for the air feed was conducted so that the process was given some time to stabilize with all the process inputs being constant, the fuel feeding was kept at a rate corresponding to the nominal heat output of the boiler (25 kW) and the frequency of the fan motor was 35 Hz. Step responses obtained are illustrated in Fig. 5.6.

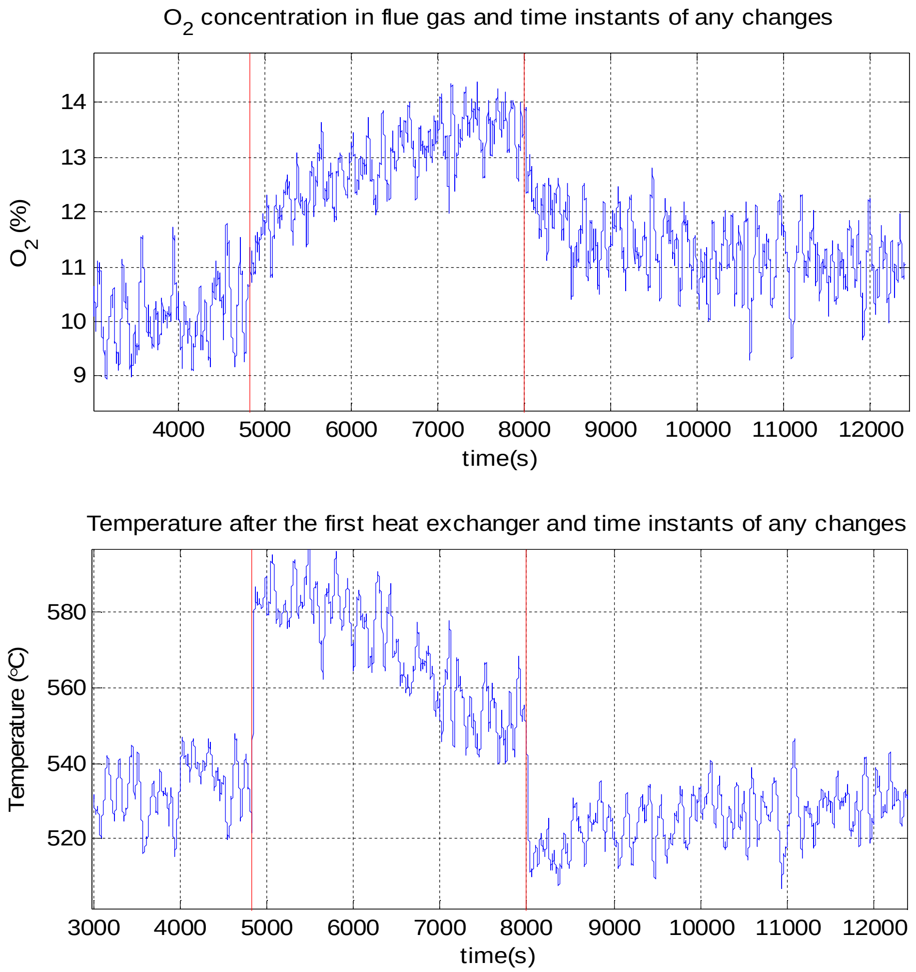


Figure 5.6. *Step responses of oxygen concentration (upper) and temperature (lower) to air feed changes. Vertical lines (red) indicate the time of step changes. First step was up (+8 Hz) and second down. (-8Hz).*

With the mentioned process input values the oxygen concentration settled to c. 10 %. At 4,800 s a step was made to the air feed increasing the frequency of the fan motor to 43 Hz. Oxygen concentration value started to rise quite slowly and by the end of the step it had reached c. 13.5 %. The slow trend may be explained by the fact that higher air feed will reduce the fuel bed size until the point where a new steady state for the bed size is reached. A step downwards into the air feed was made at 8,000 s. This step brought the frequency of the fan motor back to 35 Hz. The effect of the step change into the oxygen concentration was more rapid than in the case of upwards step, indicating different dynamics of the process depending on the direction of the step. By the end of the step oxygen concentration had reached c. 11 %. Based on these steps process models from the air feed to oxygen concentration were calculated.

In the lower part of Fig. 5.6 there is illustrated the step responses of the temperature in the upper end of the combustion chamber. As it can be seen, the response to air feed change is quite rapid. Before the first step change the temperature had settled to c. 530

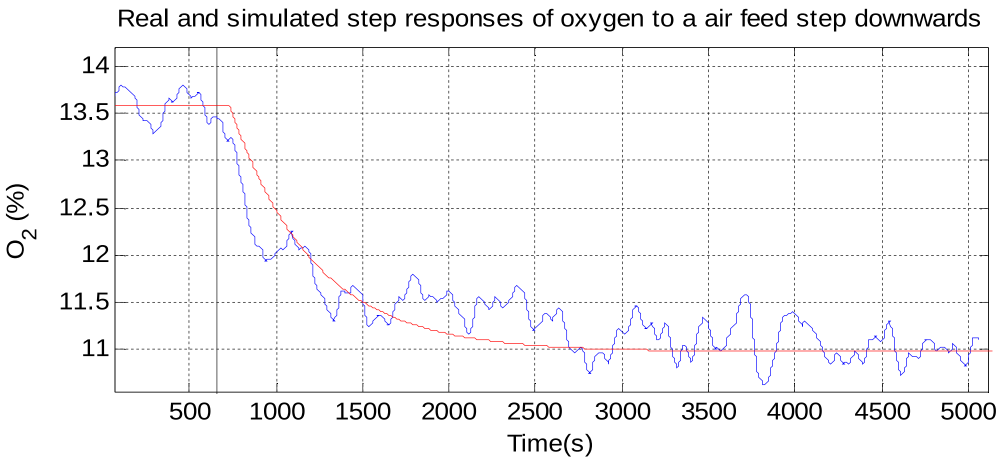
˚C and after the first step upwards was made the temperature surpassed 530 ˚C in 9 seconds. In 48 seconds the temperature had reached a level of c. 585 ˚C but slowly after this the temperature started falling down slowly and by the end of the step temperature had reached c. 550 ˚C. This kind of trend is due to, firstly, when the air feed was increased the temperature rose due to shorter residence time in the combustion chamber that pushes hotter gases further on into the boiler and in the second place, after the initial increment the temperature started to fall due to the cooling effect of the increased air feed in the combustion chamber. The slowly decreasing temperature trend is also due to smaller fuel bed due to higher primary air feed. Similar but opposite trends could be detected during the downwards step. The initial quick response was even faster downwards and the temperature started to fall within 4 seconds from the step change. After the second step the temperature in the upper end of the combustion chamber had reached c. 530 ˚C. Step responses for the temperature obtained during air feed changes have significant overshoot and it is difficult to state whether a steady state was reached at the end of the steps or not. Though, changing the air feed had almost an instant effect on the temperature, controlling this temperature by air feed would, in the end, drive the process either to a point where the oxygen would run out or to a point where the fuel bed would shrink until a point where the temperature would not be high enough to maintain the combustion. In either of these cases, the fire would extinguish. Due to these facts it was not reasonable to utilize feedback control of air feed by the temperature measurement. This is why no process models from the air feed to this temperature were estimated. When examining the step responses of the temperature it is quite clear that changes in the air feed cause fast changes in the temperature. This kind of temperature changes are bound to disturb the combustion process.

First order transfer function with a time delay (FOTD) was fitted into both upwards and downwards step data of the oxygen concentration, so that firstly the data was divided into upwards and downwards step parts.

As an example, there are step data compared with process model step responses in Fig. 5.7. The filter that was used in this particular case was running mean filter with time window of 150s. The filtered downwards step response of oxygen and the step response of the fitted FOTD model are illustrated in upper part of Fig. 5.7. In the lower part of Fig. 5.7 there is illustrated the filtered upwards step response of oxygen data and step response of the fitted FOTD model. The fitted process models are presented in Equations 5.2 and 5.3.

$$P1_{up} = \frac{0.388}{484s+1}e^{-79.8s} \tag{5.2}$$

$$P1_{down} = \frac{0.325}{484s+1}e^{-79.8s} \tag{5.3}$$



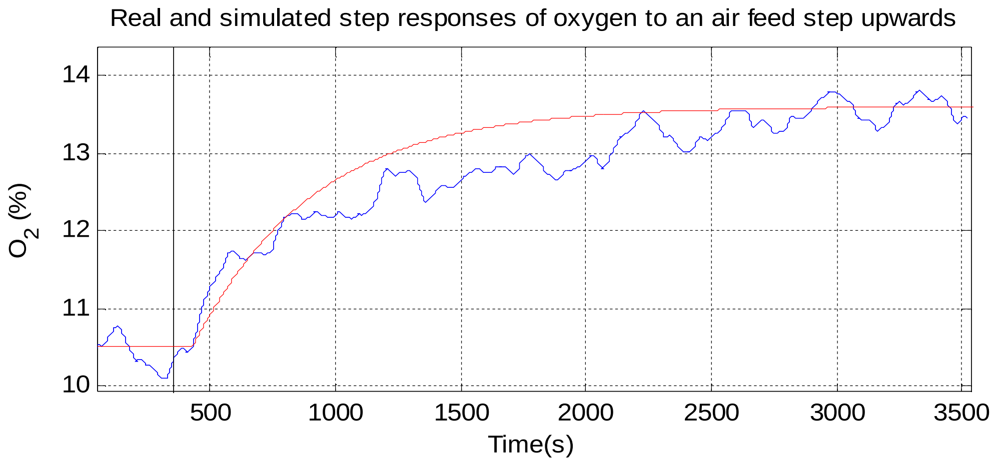


Figure 5.7. *Upper figure: Step response of a -8Hz step that the models $P1_{down}$ (red) gives in comparison to step response data; Lower figure: Step response of a +8Hz step that the model $P1_{up}$ (red) gives in comparison to step response data. Data in both steps is filtered with running mean filter with a time window of 150s.*

When comparing the process models that were obtained, it can be seen that there is a significant difference in the process gain. This might indicate that the process was not stabilized yet when making the first step from 35 to 43 Hz. The dynamics of the downwards step was faster and a FOTD model with as short time constant as 214s gave quite good fit to the data also. Though, the same time constant 484s as with the upwards step, seemed to give quite accurate fit to the downwards step as well. The time delay was the same regardless of the direction of the step.

With this filter, the normalized time delay of these processes is the same due to the same time constant and time delay.

$$\tau = \frac{L}{T+L} = \frac{79.8}{484+79.8} = 0.14 \qquad (5.4)$$

Normalized time delay of 0.14 indicates that the process is lag-dominated and also that it should be easily controlled.

5.3.2 Step response to fuel feed change

Another process input was fuel feed. Similar step response experiments were conducted with the fuel feed also to see the effect it has on the output variables.

The step response experiment was conducted so that in the beginning the boiler was run on constant air feed with 35 Hz as the frequency of the fan motor and constant fuel feed that corresponded to the nominal 25kW power level. After c. 140 minutes from the start, the first step was made in which the fuel feed was decreased to 90% of the nominal power output. The process was given c. 90 minutes to stabilize and after that the fuel feed was increased back to 100% of the nominal power level. The step responses of oxygen concentration and temperature after the first heat exchanger to these fuel input changes are illustrated in Fig. 5.8.

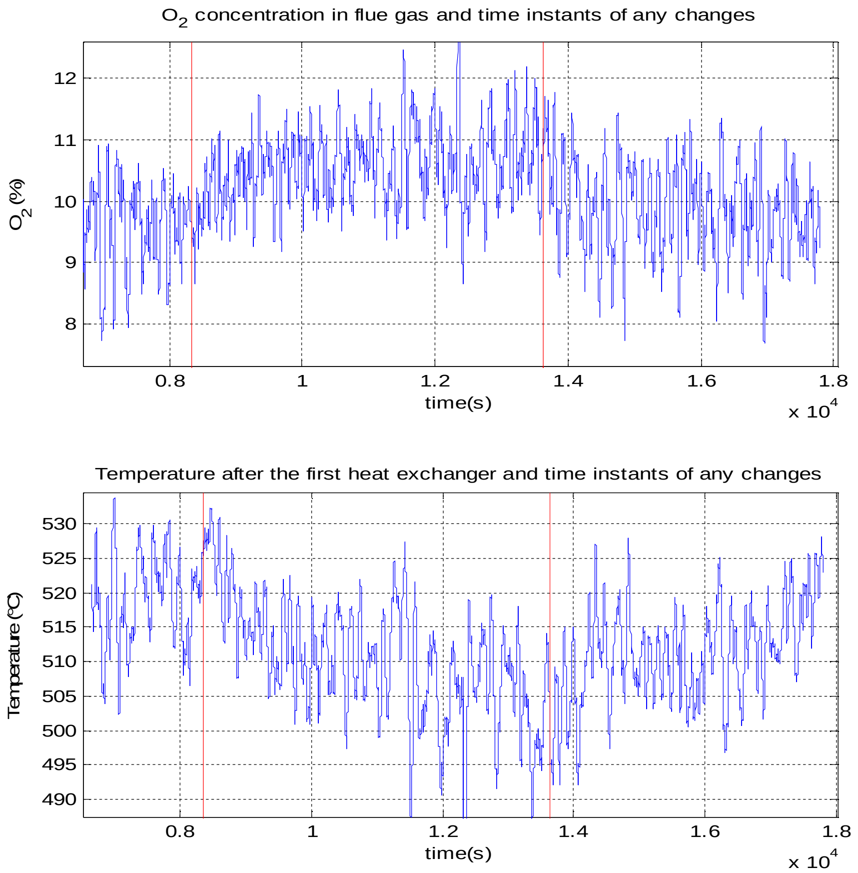


Figure 5.8. *Step responses of oxygen concentration (upper) and the temperature (lower) to fuel feed changes. Vertical lines (red) indicate the time of step changes. First step was down and second one up. Fuel feed values were 3s/20s (100%), 3s/22s (90%) and 3s/20s (100%).*

The temperature is illustrated in the lower part of Fig 5.8. Before the first step was made the temperature had stabilized to c. 520 ˚C and after the step downwards the temperature started to go down very slowly reaching c. 500 ˚C in the end of the step. Temperature seemed to be quite unstable at the end of the step, making it hard to state whether a steady state was achieved or not. Step upwards to the fuel feed caused the temperature to slowly start rising and by the end of the step the temperature had reached 520 ˚C. This latter step does not correspond to the normal shapes of FOTD models too well. Since this temperature was quite sensitive to changes in the air feed, the strange trends in it might be due to some disturbances in the air feed. Unfortunately, there was no air flow measurement in use during this experiment. Another explanation could be random occurrences (e.g. acceleration of combustion) in the combustion chamber, which could cause occasional peaks to temperature.

Oxygen concentration in the flue gas was the other variable in concern and it is illustrated in upper part of Fig. 5.8. Oxygen excess in the flue gas settled to c. 9.7 % before making the first step. Naturally, after decreasing the fuel feed rate the oxygen excess increased. In the end of the first step it had reached c. 10.8%. When fuel feed was increased back to 100% power level, the oxygen level returned to 9.7%. During the 100% power periods, when O_2 concentration was low, the CO emissions started to rise and there emerged higher peaks when compared to the high O_2 concentration period.

First order transfer functions with delay were fitted to the steps of both these variables since it is possible to control both of these variables with fuel feed without causing any undesired situations, as was the case with air feed effect on the temperature.

Again a few different filters were tested so new process models had to be calculated after changing the filter. All the process models that were calculated based on this step response experiment and according filters that were used are gathered in Tables 5.5 – 5.7. As an example in the following Fig. 5.9 the output data was filtered with a running mean filter with time window of 150s. Illustration of the downwards (upper figure) and upwards (lower figure) step responses of temperature given by the FOTD model P1 (Eq. 5.5) compared with the filtered step response data is in Fig. 5.9.

$$P1 = \frac{1490}{916s+1}e^{-313s} \tag{5.5}$$

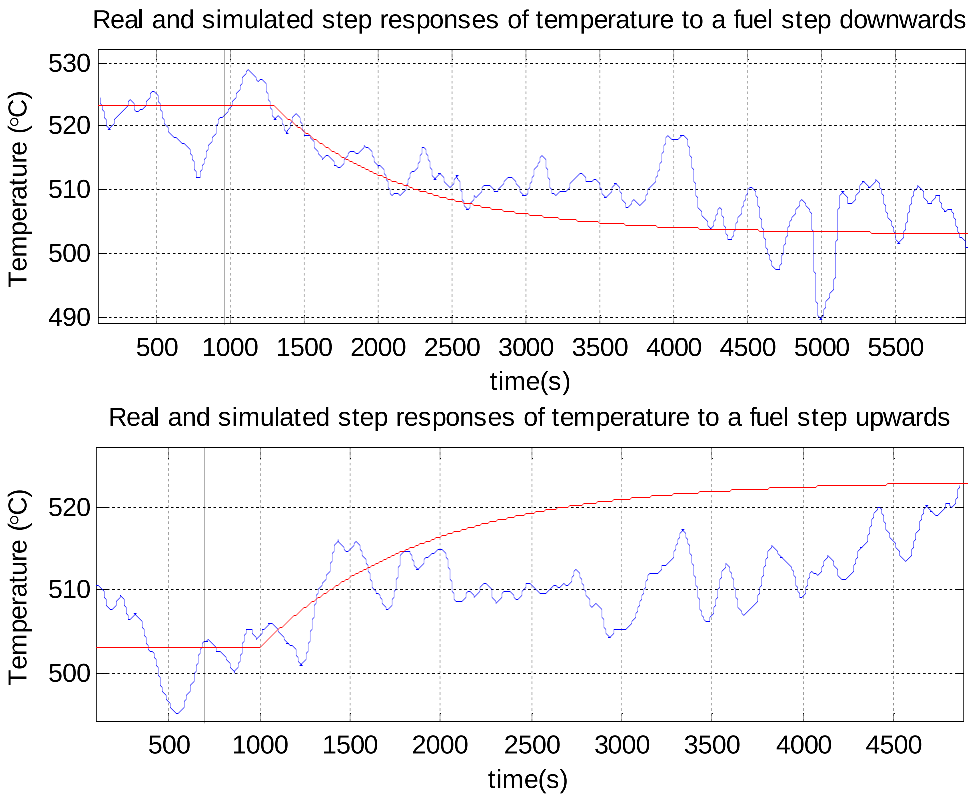


Figure 5.9. *The step response of the temperature to -10% units (upper) and +10% units (lower) steps to fuel feed that the model P1 gives in comparison to step responses. Data is filtered with a running mean filter with a time window of 150s.*

There is a significant time delay from the fuel feed to the temperature and when the step response data is filtered the delay is over 5 minutes. The dynamics of the process are also quite heavy in the terms of time constant of the FOTD model. The low-pass filtering did not remove any essential information of the process since the behavior was naturally slow. The process model that was computed does not give very exact fit to the downwards step data but on the other hand this temperature was very sensitive to disturbances in the air feed which can cause some differing from the normal behavior.

The step upwards did not result in a typical FOTD step response. Due to this the same process model that was obtained from the step downwards does not fit to this data at all. The shape of the response seems to be more like a one from an integrating process, which cannot be the case here. This strange behavior may again be due to some disturbances in the air feed as discussed earlier in this chapter.

It was concluded that the process model from the fuel feed to the temperature after the first heat exchanger has to be made based on the downwards step only, due to the distorted step response obtained from the upwards step. When a similar upwards step was made earlier, the step response corresponded to a normal FOTD-model but the problem with using that particular step response data is that the operation point was totally different and also the fuel feed change was 100% of the nominal heat output

(66 -> 166%). In order to make sure that the step response obtained was really distorted by disturbances, the experiment should have been repeated but due to time limitations this was not conducted.

The normalized time delay of the process is presented in Equation 5.6

$$\tau = \frac{L}{T+L} = \frac{313}{916+313} = 0.255 \tag{5.6}$$

and it indicates a lag dominated process. However, when comparing the normalized time delay 0.255 to the process model of air feed to the oxygen concentration this process is significantly more balanced in terms of time delay and time constant. Longer normalized time constant indicates a process which should be harder to control.

FOTD process models from the fuel feed to the oxygen concentration in the flue gas were also calculated from the same experiment data. The step response given by the FOTD model P1 (Eq. 5.7) in comparison to filtered data is illustrated in Fig 5.10.

$$P1 = \frac{-89.5}{923s+1} e^{-187s} \tag{5.7}$$

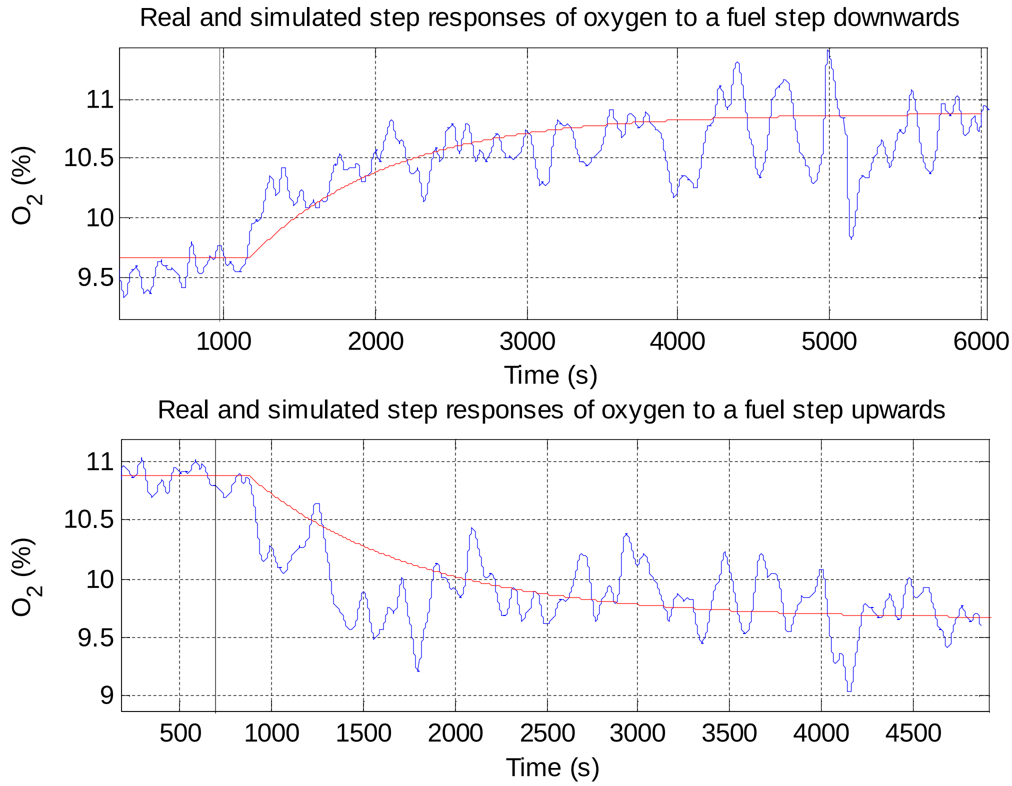


Figure 5.10. *Step responses of oxygen concentration to a -10% units step (upper) and +10% units step (lower) to fuel feed that the model P1 gives in comparison to step response data. Data is filtered with a running mean filter with a time window of 150s.*

Despite the upwards step to the fuel feed caused quite a distorted response to the temperature, the response of O_2 corresponded quite well to a normal step response. A significant time delay can be found but what is interesting is that the time delay is shorter than in the process model from fuel feed to the temperature in the upper end of the combustion chamber. The process time constant was 923 s which indicates that the process is still very slow. Thus the heavy low-pass filtering in this case also is justified. The sign of the process gain is minus due to the fact that increasing fuel feed with constant air feed decreases the oxygen concentration in flue gas and vice versa.

The normalized time delay of this process is presented in Equation 5.8

$$\tau = \frac{T_d}{T_{p1} + T_d} = \frac{187}{923 + 187} = 0.168 \qquad (5.8)$$

and indicates that the controllability of the process should be better than the controllability of the process from the fuel feed to temperature due to smaller normalized time delay value, which means that the process is less time delay dominated. When comparing the normalized time delay of this process to the one from air feed to oxygen concentration it can be seen that these two processes are quite close to each other regarding controllability.

5.3.3 Setpoint limits

As was discussed in Chapter 4.3, the lower limit for the oxygen level is a case-specific for every boiler model, heat output level and air division. For example, on lower heat output level a higher excess oxygen level is required in comparison to a higher heat output level. (e.g. Korpela et al., 2009a) Limits where the CO emissions start to incline must be searched in order to be able to run the process so that it produces minimal amount of emissions and at the same time obtain as high efficiency as possible. During step response experiment for the air feed, the lower limit for the oxygen level was searched by sweeping over a range of oxygen levels and comparing them with CO emissions. In Fig. 5.11 the results from that experiment are illustrated by comparing the O_2 and CO values to each other.

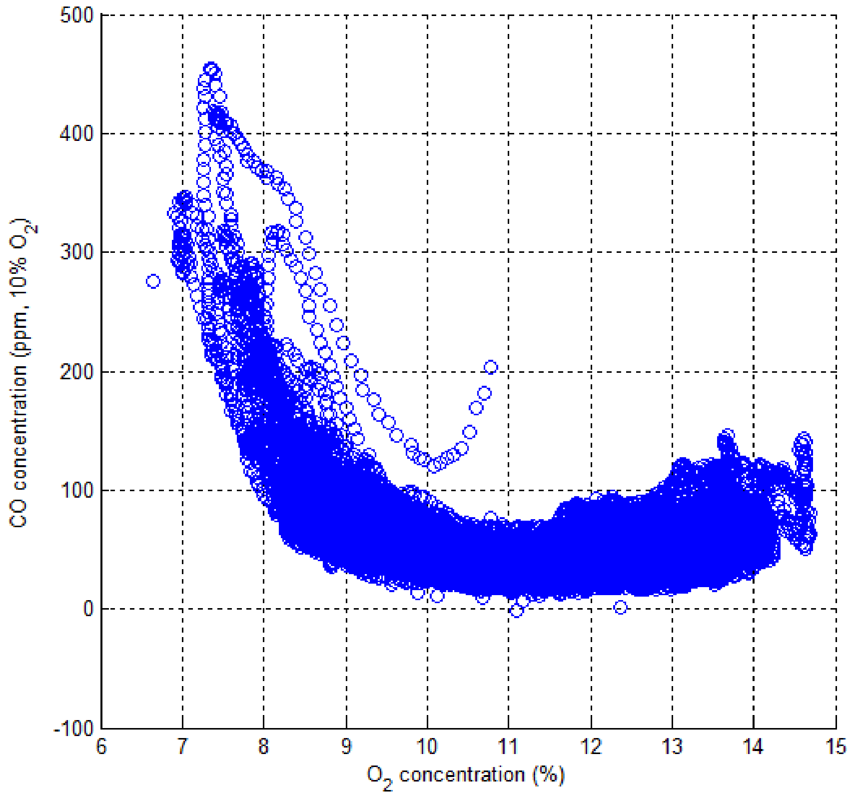


Figure 5.11. *Oxygen concentration (O_2) in comparison with carbon monoxide (CO). No time delay compensation was necessary with these variables.*

The dependency is very similar to the one depicted in Fig. 2.1. CO emissions start to incline quite rapidly after a certain limit is reached. In this case the CO emissions start to rise more rapidly after the oxygen level undershoots 10 %. The standard deviation in the oxygen value during normal run (oxygen excess c. 10.5 %-units, power level 25 kW, frequency of fan motor 35 Hz) is 0.56–0.58 percentage units which has to be taken into account when setting the setpoint. The lower limit is reached with quite high oxygen concentration comparing to many pellet boilers which can operate with oxygen excess as low as 7–8% (Korpela et al., 2009b). This limits the efficiency that is possible to obtain from the boiler. It is likely that this characteristic is due to the dimensions of the combustion chamber which are quite large in comparison to the nominal output of the boiler. However, it has to be stated that with O_2 concentration range around 10–11 % the CO emissions are remarkably low.

The point where the minimal emissions are produced seems to be at c. 11% and from this point on the CO emissions start to rise due to too high air feed. Also the deviation starts to grow when moving further on from the minimum point.

Setpoint limits for the temperature in the upper end of the combustion chamber are somewhat abstract because the dependence of the temperature on the other variables is very complex. The temperature after the first heat exchanger in comparison to CO emissions is illustrated in Fig. 5.12.

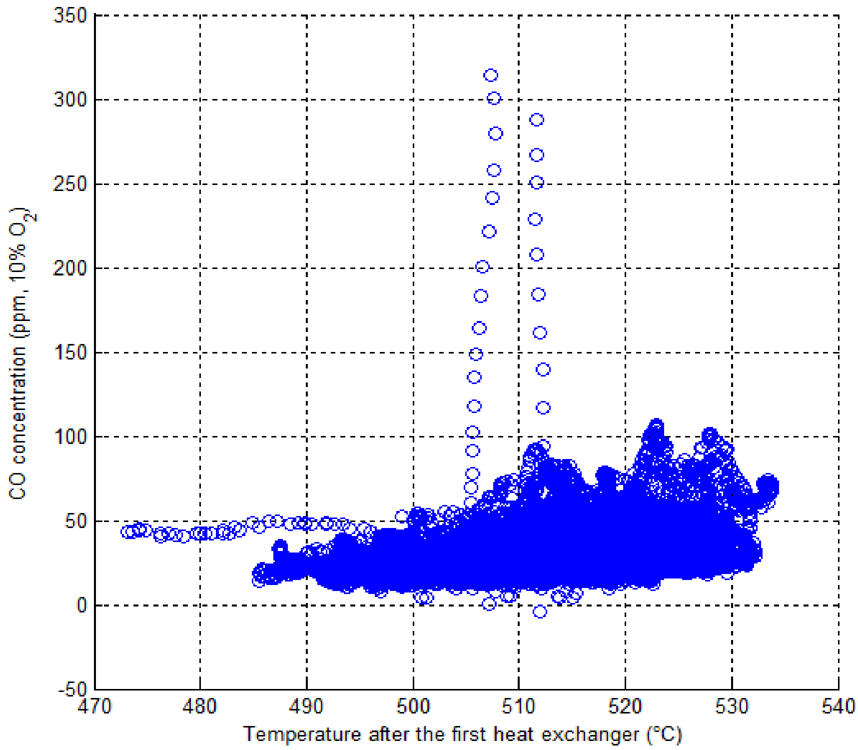


Figure 5.12. *Temperature after the first heat exchanger in comparison with carbon monoxide (CO). CO value is delay compensated by 26 s due to the time delay of the analyzer.*

The data was obtained from the fuel step response experiment. The CO concentration seems to lower as the temperature decreases but actually this behavior is due to increasing amount of oxygen in the process due to smaller amount of fuel fed (90% power level). When running the boiler on the nominal power level (25 kW) and with fan motor frequency of 35 Hz, the temperature in the upper end of the combustion chamber is 525–530 °C. This temperature is gained only in this working point because the temperature at this location rises significantly when air feed is increased and vice versa, as discussed in Chapter 5.3.1. Due to this working point dependency a proper setpoint finding can be very difficult and the sensibility of some setpoint must be evaluated with criticism. A setpoint given by a controller controlling oxygen in cascade mode, however, can provide a sensible setpoint for a temperature controller (Korpela et al., 2008).

The delay compensation was done based on the correlation analysis studies conducted by Timo Korpela which showed a time delay of 26 s between the CO and the temperature measurement.

5.4 Controller tuning

Controller tuning was carried out by utilizing the formulas presented in Chapter 4.1 and process models as discussed in Chapter 5.3. As an example, in the following chapters there are presented all the controller parameters given by the tuning rules to the example process models calculated in Chapter 5.3. Since a process model was recalculated for every filter, all the process models and the controller parameters that were used in the experiments are illustrated in Tables 5.5 – 5.7 in Chapter 5.5.

As discussed in Chapter 5.3, due to the nonlinearity of the process the identified process models are valid only in one operating point. Thus when using a controller tuned by the tuning formulas which use process models that are valid only in a restricted area, the controller can act either too aggressively or too calmly when controlling the process out of the validity area. Additionally, all the tuning formulas presented in Chapter 4.1 are developed based on linear control theory. This fact in mind, utilization of these tuning formulas for controlling a heavily nonlinear process can be regarded as somewhat dubious. However, some of these formulas provide good rough estimates for the controller parameters to begin with.

When comparing the controller parameters with the calculated process models in general, the AMIGO PI method provided always the smallest controller gain and Ziegler-Nichols (ZN) methods in turn the largest. The controller gain from the Åström & Murray (Å&M) method fell always in between the above-mentioned two. Gain from ZN methods were regularly too high as was the case quite often with the Å&M and AMIGO PID methods.

5.4.1 Oxygen control by air feed

Process models from the air feed to oxygen concentration were identified in Chapter 5.3.1 and they were as follows.

$$P1_{up} = \frac{0.388}{484s+1} e^{-79.8s} \tag{5.9}$$

$$P1_{down} = \frac{0.325}{484s+1} e^{-79.8s} \tag{5.10}$$

The filter that was used with these process models was a genuine running mean filter with time window of 150 seconds. By placing the process model parameters of the $P1_{up}$ to the tuning formulas, PI and PID controller parameters were obtained and they are presented in Table 5.1. The $P1_{up}$ process model was chosen to be used with the tuning formulas because there was uncertainty concerning the length of the time constant with $P1_{down}$. However, afterwards as the obtained process models were evaluated with criticism, it was concluded that the process was not stabilized when the upwards step change was made and thus it would have been better to use $P1_{down}$ with the tuning formulas.

Table 5.1. PID parameter values obtained with different tuning methods for the oxygen controller by air feed.

ZN	PI	K = 14.1 Hz / %-point T_i = 239 s T_d = 0 s
	PID	K = 18.8 Hz / %-point T_i = 160 s T_d = 3180 s
AMIGO	PI	K = 3.96 Hz / %-point T_i = 355 s T_d = 0 s
	PID	K = 7.56 Hz / %-point T_i = 260 s T_d = 38 s
Åström & Murray	PI	K = 5.87 Hz / %-point T_i = 312 s T_d = 0 s

5.4.2 Oxygen control by fuel feed

Process model from the fuel feed to oxygen concentration was identified in Chapter 5.3.2 and it was as follows.

$$P1 = \frac{-89.5}{923s+1} e^{-187s} \tag{5.11}$$

The filter that was used with this process model was a genuine running mean filter with time window of 150 seconds. By placing the process model parameters to the tuning formulas, PI and PID controller parameters were obtained and they are presented in Table 5.2.

Table 5.2. PID parameter values obtained with different tuning methods for the oxygen controller by fuel feed.

ZN	PI	K = -0.0496 feed ratio / %-point T_i = 561 s T_d = 0 s
	PID	K = -0.0662 feed ratio / %-point T_i = 374 s T_d = 17500 s
AMIGO	PI	K = -0.0133 feed ratio / %-point T_i = 719 s T_d = 0 s
	PID	K = -0.0271 feed ratio / %-point T_i = 544 s T_d =88.1 s
Åström & Murray	PI	K = -0.0210 feed ratio / %-point T_i = 628 s T_d = 0 s

5.4.3 Temperature control by fuel feed

Process model from the fuel feed to the temperature in the upper end of the combustion chamber was identified in Chapter 5.3.2 and it is as follows.

$$P1 = \frac{1490}{916s+1}e^{-313s} \tag{5.12}$$

The filter that was used with this process model was a genuine running mean filter with time window of 150 seconds. These process model parameters were placed to the tuning formulas. PI and PID controller parameters that were obtained are presented in Table 5.3.

Table 5.3. PID parameter values obtained with different methods for the temperature controller by fuel feed.

ZN	PI	K = 0.0018 feed ratio / ˚C T_i = 939 s T_d = 0 s
	PID	K = 0.0024 feed ratio / ˚C T_i = 626s T_d = 49000 s
AMIGO	PI	K = 4.2 e-004 feed ratio / ˚C T_i = 797s T_d = 0
	PID	K = 0.001 feed ratio / ˚C T_i = 664 s T_d = 142 s
Åström & Murray	PI	K = 7.9 e-004 feed ratio / ˚C T_i = 709 s T_d = 0 s

5.4.4 Cascade control of oxygen

The secondary controller in the cascade scheme was a controller controlling temperature by manipulating fuel feed. In the previous chapter the tuning parameters for such controller were presented (see Table 5.3) and the same parameters were used for the secondary controller in this cascade scheme also. Since the controller gains of the primary controller should be smaller than the ones of the secondary controller and since the controller gains of the secondary controller were already quite small, it was concluded that in the experiments the primary controller would only be an I-controller. The integration time constant of the I-controller was chosen to be 5 times as large as the integration time constant of the secondary controller. When the temperature setpoint is raised the oxygen concentration goes down and vice versa. Due to this fact the sign of the integral gain in the primary controller was negative.

5.5 Control experiments

In the following chapters the experiments with different input-output pairs from Chapters 4.3 and 4.4 are discussed. There is always one figure withholding plotted continuous data from one example run with the particular input-output pair. Also the open loop control experiments conducted are discussed in Chapter 5.5.1. The feedback control results are compared with the open loop control results. In all the following figures and tables carbon monoxide measurements are reduced to oxygen concentration of 10 % according to EN 303-5 (Suomen standardoimisliitto).

5.5.1 Open loop control

The statistics from open loop control experiments are illustrated in Table 5.4. It can be seen that the same process inputs can result in quite different operating areas in terms of O_2 and temperature which verifies that there is a tendency of drifting in the process. This kind of behaviour is undesirable and calls for feedback control. In the oxygen a difference of one percentage unit could be seen, which will cause significantly differing combustion conditions. Also a difference of c. 10 °C in the temperature could be noticed. This change itself is insignificant but it indicates changes in process behaviour. Yet, the open loop behaviour of the combustion process was rather stable after the optimization of the grate sweeping sequence which can be seen in the deviations in Table 5.4.

Table 5.4. Results from the open loop control experiments.

Experiment	O_2 mean ± std (%)	CO mean ± std (ppm)	Tex mean± std (°C)	Power (% of nom.)	Input of the air fan (Hz)
1	10.56±0.56	28.3±5.7	518.6±8.3	100	35
2	10.88±0.58	33.5±5.2	519.6±8.1	100	35
3	10.12±0.59	30.8±9.5	533.4±7.4	100	35
4	11.08±0.56	27.0±6.8	528.0±6.5	100	35

In the open loop experiments, the boiler was run on the nominal power level (25 kW). On this power level, the combustion got more restless in the terms of CO production when the O_2 concentration in the flue gas reached a mean concentration of 10.1%. The first indication of this can actually be seen in the standard deviation of CO rather than in the mean value of CO. The same restlessness cannot yet be seen in the deviations of the O_2 or the temperature.

5.5.2 Control of oxygen by air feed

The results obtained by controlling oxygen by air feed are listed in Table 5.5.

Table 5.5. O_2 control by air feed. In the filter column: 1 = running mean filter with 150s time window (discrete transfer function), 2 = running median filter with 120s time window, 3 = running mean filter with 120s time window, 4 = running mean filter with 150s time window. O_2 setpoint = 10.5% in every experiment. u stands for fan input in Hz.

Fil-ter	FOTD model: K_p/T/L	Tuning method	PID parameters: K/T_i/T_d	contr. mean ±std	CO mean ± std	u mean ± std	Tex mean ± std	Power (% of nom.)
1	0.338/150/41	AMIGO PI	2.4 / 125 / 0	10.50 ±0.67	31.4± 11.7	34.5± 0.7	535 ±9.4	100
1	0.338/150/41	AMIGO PID	5.5 / 100 / 18	10.47 ±0.56	32.7± 12.6	33.4± 1.1	517 ±12.5	100
2	0.325/216/87	Å&M PI	3.14 / 174 / 0	10.50 ±0.61	31.5± 6.8	35.3± 1.4	533 ±15.3	100
3	0.325/466/67	AMIGO PI	5.6 / 327 / 0	10.43 ±0.60	32.5± 9.3	34.9± 1.9	500 ±21.2	91
3	0.325/240/67	AMIGO PI	2.44 / 201 / 0	10.50 ±0.67	27.8± 8.7	35.2± 0.8	509 ±10.1	91
4	0.388/484/79.8	AMIGO PI	3.97 / 356 /0	10.43 ±0.7	49.9± 30.1	31.4± 1.7	441 ±14.4	76.3
-	-	75% AMIGO PI	2.98 / 356 / 0	10.49 ±0.68	56.3± 16.3	32.1± 1.1	456 ±9.8	76.3
-	-	50% AMIGO PI	1.98 / 356 / 0	10.54 ±0.61	50.0± 12.8	32.3± 0.7	458 ±7.4	76.3
-	-	Lambda	2.73 / 484 / 0	10.44 ±0.57	47.8± 13.5	32.2± 0.8	457 ±7.5	76.3
4	0.388/484/79.8	50% AMIGO PI	1.98 / 356 / 0	10.55 ±0.66	38.3± 7.8	34.3± 0.5	485 ±6.7	87

When evaluating the results it can be seen that the setpoint was always achieved and maintained. Yet, when comparing the standard deviations of all variables in the oxygen control experiments with 100% power to the ones from the open loop control experiments, such low deviations could not be obtained with this feedback control. The oxygen behaviour with this feedback control strategy was thus more restless which in turn causes larger deviation into other variables as well. This is to some extent due to the long time delay of the oxygen measurement and the heavy filtering of the measurement signal. Another reason for the larger deviation was that the primary and secondary air was produced with the same fan. Thus, the undesired effects from adding more primary air when only increased amount of secondary air was wanted worsens the results severely. In general, better results were obtained when using small controller gains to take the unnecessary aggressiveness from the controller.

FOTD models in Table 5.5 are so accurately presented only to emphasize the connection to the new controller parameters. It has to be stated that the significance of the last number in the models is quite small to the real behaviour of a process. The statistics of oxygen concentration in Table 5.5 are also quite exact to reveal the small differences between the control tests. However, since the differences are so small, the evaluation of the results based on the oxygen concentration is quite tricky. The nature of the control can be seen more clearly in the standard deviations of CO, fan input and temperature.

The effect of the operating point can clearly be seen when comparing the results of feedback control with 100 % and 76 % power levels. Feedback control made the process more restless in the terms of CO on a lower power level. Even though the oxygen excess was on a reasonable level during the lower power level experiment, the CO emissions rose due to inadequate mixing of combustion air to volatile gases in the relatively large combustion chamber. The mixing is worse because the volume flow to the chamber is smaller and thus the speed of the air is slower. Additionally, due to lower fuel feed the combustion chamber temperature was lower which can momentarily be so low that the combustion gets disturbed by it.

As discussed in chapters 5.3 and 5.4 the nonlinearity causes difficulties into the process identification and into the tuning process. Since the obtained process model was obtained in power region between 90 – 100%, the process model as such cannot be said to be valid in the 76% power region and due to that, the controller parameters in that area might be incorrect. Additionally, since the control experiments were conducted in different power regions the straightforward comparison of the different controller parameters is quite tricky. Thus only the experiments done on the same power level can be said to be comparable with each other. This control strategy was the first one to be tested on the boiler. There emerged some difficulties with finding a suitable measurement filter and that is the main reason why this control strategy was tested quite many times.

Since the disturbances in the process were quite fast and the dynamics of the process were quite heavy, it was impossible to control the disturbances despite the air feed was the fastest way to affect the process. However, by utilizing this strategy the drifting of the process could be avoided.

The 76 % power level experiment is discussed in more details in the following. Data from the experiment is illustrated in Fig. 5.13.

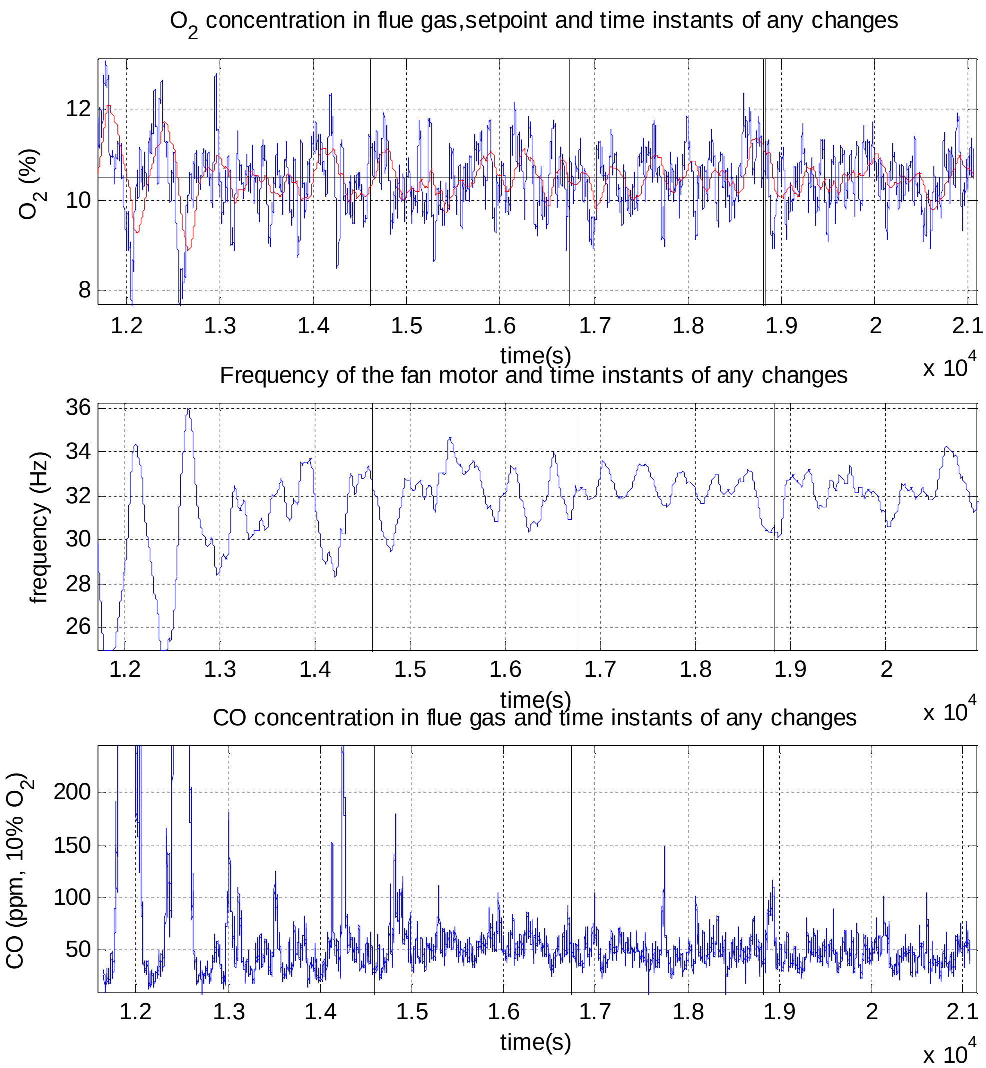


Figure 5.13. *Upper: Oxygen in the flue gas (unfiltered blue and filtered red), setpoint for oxygen (black) and times of regulator parameter changes (black); Middle: Frequency of the fan motor; Low: CO concentration in flue gas. PI-parameters: 1) K =* 3.97, *Ti = 356, 2) K = 2.98, Ti = 356, 3) K = 1.99, Ti = 356, 4) K =* 2.73, *Ti =484. K is proportional gain [Hz / % unit], Ti is integration time constant [s].*

Although the normalized time delay of process, $\tau = 0.14$ indicates good controllability, the disturbances affecting the process and the fixed primary/secondary air ratio made it quite difficult for the controller to maintain the setpoint. Also, the small value of τ causes the tuning formulas to give quite heavy controller gains. In the experiment, the fuel feeding period was 26.2 s (76% of the nominal power). Grate sweeping was done according to the upgraded sequence, once in every 120 seconds.

The first set of PI controller parameters was obtained from the AMIGO PI tuning formula which gave the smallest controller gain. Despite the controller gain was the smallest; there emerged quite severe fluctuation when the controller was switched on. It took c. 1500 s for the controller to get the large fluctuations in oxygen concentration stabilized. This kind of behavior is an indicator of too high proportional gain in the controller. When the values stabilized, the filtered oxygen concentration remained within c. 0.5 percentage units and the unfiltered within c. 1 percentage units from the setpoint. During these PI parameters the standard deviation of the real, unfiltered oxygen concentration was 0.7 percentage units. At c. 14,600 s the proportional gain of the controller was decreased to 75 % of the original value and due to this, the behavior of oxygen got a bit more stable lowering the standard deviation to 0.68 percentage units. A further decrease to 50% of the original value into proportional gain value caused again slightly calmer behavior in O_2 value. During this period the peaks were slightly smaller and the filtered value remained within less than +/- 0.5 percentage units from the setpoint and the standard deviation of the oxygen concentration was 0.61 percentage units. During the last set of PI parameters, which were obtained by using the lambda tuning method, the O_2 behavior seemed to get slightly more restless than during the previous PID-parameters. However, the standard deviation of O_2 got somewhat smaller to 0.57. Also when compared to the second set of controller parameters which had approximately the same controller gain, the process behavior got more stable in the terms of the standard deviations of the other process variables. This might be due to the bigger integration time constant value which should calm down the control. The effect of the different controller parameters on the oxygen behavior during this experiment was quite small, excluding the first controller parameter set. Since the setpoint was always maintained, the standard deviation is the only variable changing and the changes in it were very small. Thus the evaluation of the results has to be made based on the other process variables.

As it can be seen in the middle part of Fig. 5.13, after the controller was switched ON, there emerged heavy fluctuation in the controller output which was due to too heavy proportional gain. After getting the fluctuation stabilized, the output of the controller varied 4–5 Hz in amplitude. After decreasing the proportional gain in the next set of PI parameters the fluctuation was reduced to 3–4 Hz and a further reduction to fluctuation in the output was obtained with the third set of PI parameters where the fluctuation was about 1–2 Hz. During the last set of PI parameters the fluctuation was within 1 Hz but there were few occasional peaks. This variable is more revealing in the terms of the nature of the control and since the varying air feed has a disturbing effect on the process as small variation as possible is desirable. Thus, the less aggressive, smaller controller gains worked better. Again it has to be stated that the differences between the last three controller parameters sets were quite small.

The purpose of the oxygen control was to ensure clean combustion i.e. as low emissions as possible, and a good indicator about the quality of the control is thus carbon monoxide concentration in the flue gas which is illustrated in the lower part of Fig. 5.13. During the first set of PI parameters there emerged huge peaks in the CO concentration due to the large variation in air feed which in turn caused large variation into the oxygen concentration in the combustion chamber and also into the temperature in the combustion chamber due to the cooling effect of the increased air feed. After the air feed was stabilized by the controller, CO value remained around 40–50 ppm which still is quite a low value. During the first set of PI parameters the standard deviation was high due to the very high peaks of CO emissions in the beginning. When the second set of PI parameters was in use the mean value of CO was 56 ppm and the standard deviation was 16 ppm. Mean value was decreased to 50 ppm and standard deviation to 12.3 ppm during the third set of parameters. The last set of PI-parameters caused only a slight change in the CO emissions, mean value of CO was 47.8 ppm and standard deviation was 13.5 ppm. During the regulation periods there are occasional peaks in the CO value due to fast drops in the O_2 value which are probably due to fast changes in air flow conditions discussed in Chapter 5.2. In general, the CO emissions during this experiment were higher than in the experiments with higher power levels. This is due to different operating area (smaller fuel feed), which leads to smaller air feed in order to reach the same 10.5 % oxygen setpoint. In other words, the O_2/CO curve in Fig. 5.11 is different at different power levels. Again the less aggressive behavior with the smaller controller gains can be seen more clearly in this variable than in the actual control variable and since the less aggressive control resulted in smaller emissions, the smaller controller gains worked better. However, the changes in the last three sets were quite insignificant.

5.5.3 Control of oxygen by fuel feed

The response of the oxygen measurement to changes in fuel feed was quite slow and calm. Thus, feedback control with this input leads to slow and calm behaviour of the controlled system which suits well to drifting prevention, although there is no possibility of fast disturbance control with this control strategy either. During the experiment, the setpoint was maintained well and the deviation of oxygen level was actually smaller than in the case of the open loop system as can be seen when comparing the results in Table 5.4 and Table 5.6. The calm control can also be seen in the small deviation in the output of the controller. Since the air feed during this strategy is kept constant (if constant power is desired), the behaviour of the combustion is very stable and it approaches the behaviour of open loop system which can be seen in the standard deviations of CO and temperature. Low CO emissions can be explained by the fact that the air feed to the system was quite high assuring good mixing of oxygen and volatile gases in addition to the relatively high temperature in the combustion chamber.

Table 5.6. Results from an experiment, where the fuel feed was controlled by the oxygen measurement. Setpoint for the O_2 was 10.5 %. Filter 4 = running mean filter with time window of 150 s.

Fil-ter	FOTD model: K_p / T / L	Tuning method	PID parameters: K/T_i/T_d	contr. mean ± std	CO mean ± std	u mean ± std	Tex mean ±std	Fan (Hz)	power (% of nom.)
4	-89.5 /923/187	AMIGO PI	-0.00133 /719/0	10.49 ±0.53	32.5± 6.8	21.7± 0.67	513.5 ±6.00	36	92

When comparing this control strategy to the oxygen control by air feed on the same power level, the standard deviation in the oxygen level was smaller and the standard deviation of the temperature was also smaller with this control strategy. Additionally the CO deviation was slightly smaller. Due to the slow effect of this input to the oxygen level, the time delay of the gas analyzer did not affect to the control results as much. Therefore, this control worked better than the control with air feed but since the difference was still quite small and since this control strategy was tested only once, a very solid comparison between the two control strategies can not be made.

In the upper part of Fig. 5.14 there is illustrated a part of oxygen concentration data from an experiment where the fuel feed was manipulated by the oxygen measurement. Also the output of the controller (fuel feed period) and CO during the same period is illustrated in the Fig. 5.14.

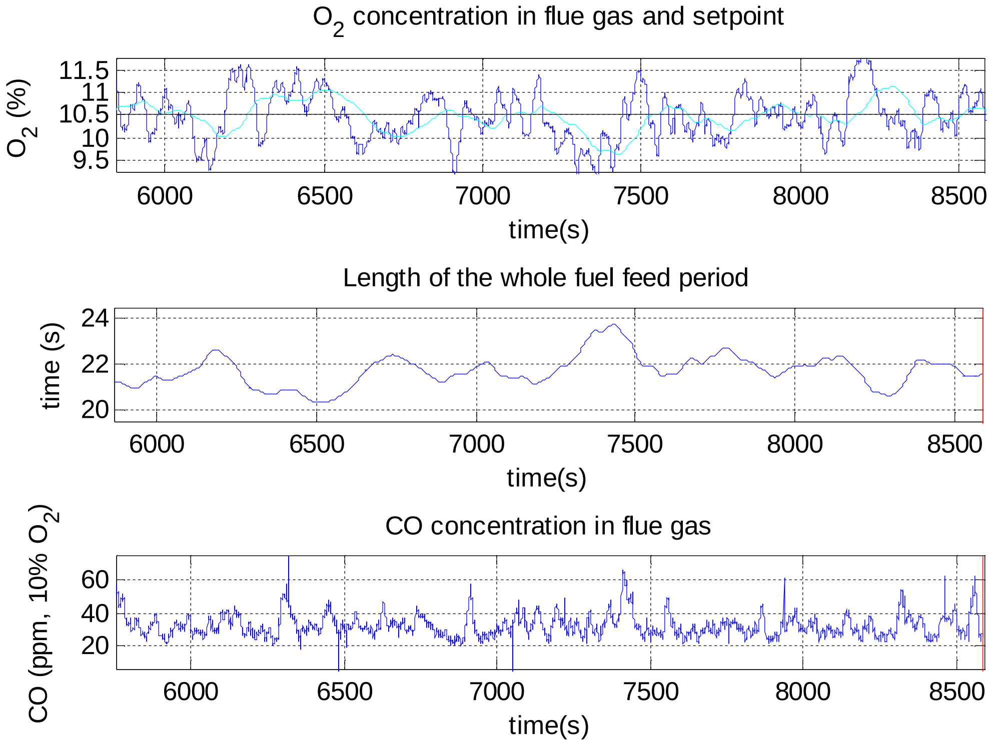


Figure 5.14. *Upper: Oxygen (unfiltered blue and filtered cyan) in flue gas and setpoint for oxygen (black); Middle: Whole fuel feed period length (controller output); Lower: Carbon monoxide in the flue gas.*

When examining the experimental run visually, the oxygen in the flue gas remained within c. 0.6 percentage units from the setpoint in the filtered value and about 1.0 percentage units in the unfiltered value.

The calm control can also be seen in the small deviation in the output of the controller which is illustrated in the middle part of Fig. 5.14. The whole length of the fuel feeding period is obtained from fuel feeding ratio *f* which is the output of the controller. The few heavier peaks in oxygen measurement caused a few peaks into the output of the controller also but excluding those, the output varied between 21–22.5 seconds which in power output means variation of 1.6 kW between 22.2–23.8 kW.

Carbon monoxide concentration remained on a very modest level during the feedback control of the fuel feed by the oxygen concentration measurement. The mean value of CO concentration was 32.5 ppm and the standard deviation was 6.8 ppm. The highest peaks did not reach any higher than c. 65 ppm. These low CO emissions can be explained by the calm behavior of the process during this period due to stable air feed (constant input of the air fan).

5.5.4 Control of temperature by fuel feed

Control of the temperature by the fuel feed resulted in quite stable combustion behaviour which is due to the slow dynamics from the fuel feed. During the experiments, the temperature setpoint was always maintained and the deviations in the temperature were quite close to the ones from the open loop system, as can be seen in Table 5.7.

Table 5.7. Results from the temperature control experiments by fuel feed. 2 = running median filter with 120s time window, 3 = running mean filter with 120s time window, 4 = running mean filter with 150s time window. u stands for the total fuel feed period.

Filter	**FOTD model: K_p/T/L**	**PID parame-ters: K/T_i/T_d**	**Tuning method**	**Set point (°C)**	**contr. mean ±STD**	**u (s) mean ±STD**	**CO (ppm) mean ±STD**	**O_2 (%) mean ±STD**	**Fan (Hz)**	**power (% of nom.)**
4	1490 / 916 / 313	0.000416/ 797 / 0	AMIGO PI	465	464.9 ±6.3	26.2± 0.4	16.8± 6.8	12.3± 0.6	35	76.3
2	1490 / 949 / 303	0.000837 / 722.5 / 0	Å&M	480	482.2 ±9.7	25.8± 1.4	28.2± 9.5	12.5± 0.7	35	77.4
3	1490 / 916 / 305	0.000429 / 703 / 0	AMIGO PI	500	497.8 ±6.7	24.0± 0.5	27.2± 6.8	11.2± 0.6	35	83.3
4	1490 / 916 / 313	0.000416 / 797 / 0	AMIGO PI	510	511.5 ±7.9	22.5± 0.4	33.8± 5.2	11.3± 0.5	38	88.9

The process models and controller parameters are quite accurately presented only to emphasize the small differences between them. Deviations of oxygen and CO were also about the same size as in the open loop system. The CO emissions were quite low during the experiments due to relatively high oxygen level. The variation in the output of the controller, excluding the second experiment in Table 5.7, was quite small indicating stable combustion behaviour.

Since the input of the fan was kept constant during the experiments, the oxygen level floats as a function of the added fuel. In order to maintain a reasonable oxygen level, a corresponding setpoint for the temperature must be found. By increasing the temperature setpoint, the oxygen level goes down and vice versa. This control strategy only calls for very simple instrumentation since there is only one thermocouple needed. Since the oxygen level correlates with the temperature, it is therefore possible to control the oxygen level indirectly by setting the setpoint properly. The problem of finding correct temperature setpoints for every power level can be quite tricky since the oxygen level is a function of air feed and fuel feed and the dependences in between are often nonlinear.

When comparing the last two experiments in Table 5.7, there is a power raise conducted by raising the temperature setpoint and the input for the air fan. The oxygen level in both of the experiments remained the same.

The identified process models were very similar and that resulted in very similar controller parameters. Only in the second experiment there was a bit higher controller gain in use and it produced slightly more aggressive behavior which can be seen in the standard deviations of temperature and input of the actuator. Since the controller parameters were so close to each other no significant differences emerged in the process variables and thus the any of the controller parameter sets, the second set excluded, could be used.

In general, the power levels were lower than in the open loop control experiments and thus the comparison of the statistics is somewhat difficult. The lower power levels were due to the fact that since the output of the fuel controller was constrained to a power region between 50–100% of the nominal, higher setpoints around 520–530 °C would have caused the controller to reach the higher limit and thus the actual control would have saturated. It was therefore necessary to set the temperature setpoint safely below a setpoint which would have caused the controller to reach the higher limit in order to achieve a successful experiment trial. The power levels were also out of the 90 – 100% region where the process identification was conducted. Thus the controller parameters might be incorrect on these lower power levels.

In Fig. 5.15 there is illustrated a continuous part of data in which temperature was regulated by the fuel feed.

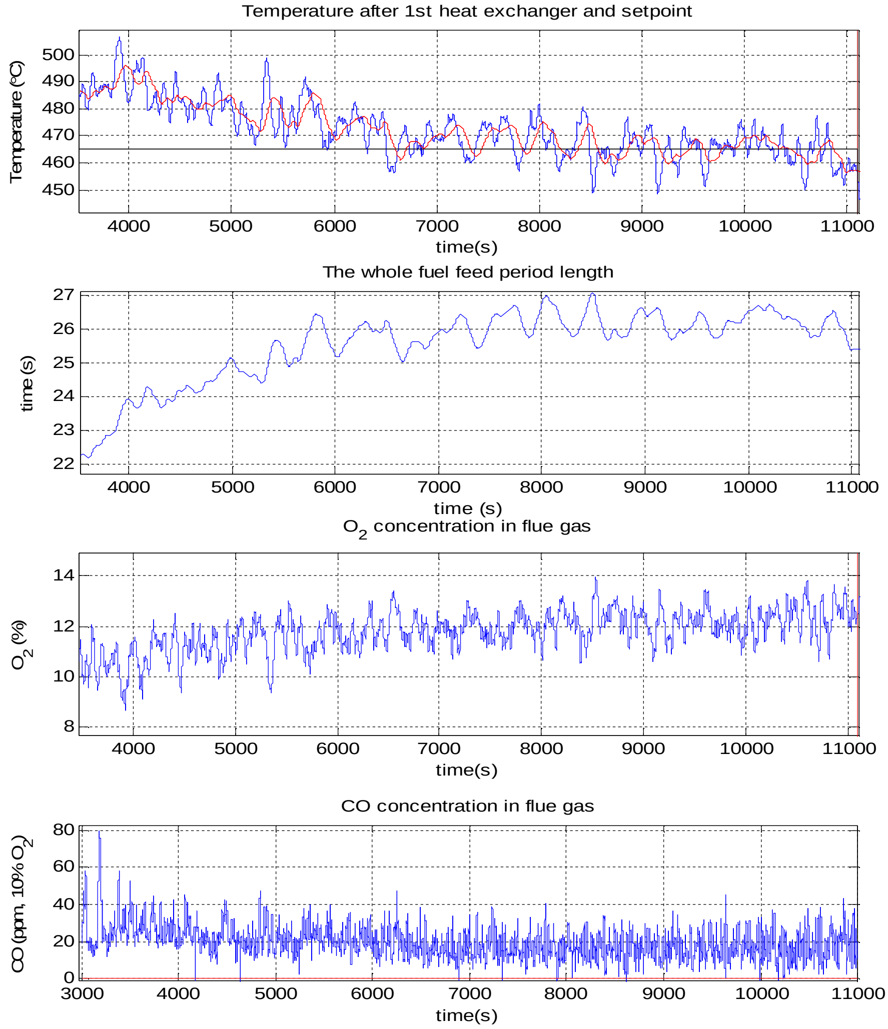


Figure 5.15. *Up: Temperature after the first heat exchanger (unfiltered = blue, filtered = red) and temperature setpoint (black); Second from the top: Fuel feed period (output of the controller); Second from the bottom:* O_2 *concentration in flue gas; Bottom: CO concentration in flue gas.*

This data part is from the experiment which is on the first row in Table 5.7. It can be seen that temperature control is quite slow since it takes over an hour for the controller to get the temperature to the setpoint.

However, this slow behavior is natural to the whole process and since the integration time constant of the controller is as large as 797 s it also slows down the control. However, after the controller gets the temperature to the area of the setpoint, the temperature fluctuates around the setpoint within +/-10˚C. Occasionally there emerged some larger peaks or slopes but those cannot be regulated out due to heavy dynamics of the process. Since this temperature is quite sensitive to changes in the air flow conditions, these fast peaks and slopes might be due to air feed disturbances and not due to problems with the fuel feed. During the last 3000 seconds of the regulation period the standard deviation was 6.3 ˚C. As it can be seen in the second figure from the top in Fig. 5.15 that at c. 8000 seconds the controller had reached a value, around of which the output from this point on started to fluctuate. The fluctuation of the fuel feed period in the end, when the setpoint was reached, was about one second which in fuel power means fluctuation of c. 1.3 kW.

5.5.5 Cascade control of oxygen

Since this control structure is very closely related to the previous control structure, the behavior that was obtained was somewhat similar but also slower due to the very long integration time constant of the primary controller. The process behavior was quite stable due to the slow dynamics of the process from the fuel feed. This can be seen in the small standard deviations in Table 5.8.

Table 5.8. Results from the O_2 control by a cascade structure. 4 = running mean filter with 150s time window.

Filter	FOTD model: K_p/T/L	PID parameters: K/T_i/T_d	Tuning method	Set-point	O_2 Mean ±STD	CO mean ±STD	u mean ±STD	Tex mean ±STD	Fan (Hz)	power (% of nom.)
4	1489 / 916 / 313	Sec. 0.00042 / 797 / 0 Pri. 0 / 3990 / 0	AMIGO PI	11.5	11.7± 0.6	38.0± 8.7	22.7± 0.5	511.4 ±8.9	38	88.1

However, in Fig. 5.16 as well as in Table 5.8 it can be seen that the O_2 control did not reach the setpoint as well as the previous control structures but it actually drifted away from it.

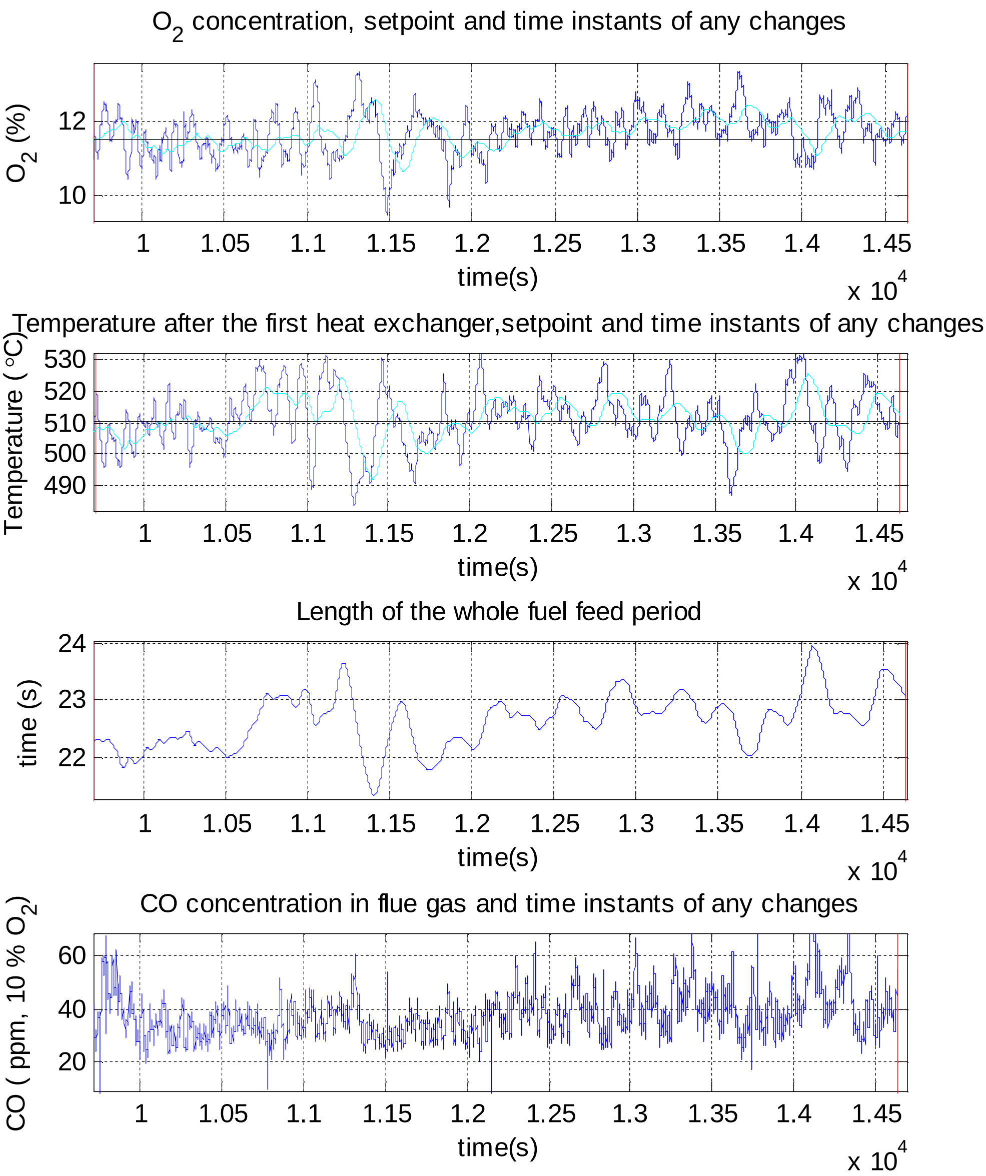


Figure 5.16. *Continuous data period of the cascade control experiment. Top: O_2 concentration (unfiltered = blue, filtered = cyan) and the setpoint (black); Second from the top: Temperature (unfiltered = blue, filtered = cyan) and the setpoint(black); Second from the bottom: Whole fuel feed period; Bottom: CO concentration.*

Yet, the cascade control gave signs of reacting to the growing control error in the end of the trial. This is largely due to the long integration time constant of the primary controller which raised the temperature setpoint during the examined period only very slightly and this did not lower the O_2 concentration sufficiently. In the second figure from the top in Fig. 5.16, it can be seen that the temperature remained around the given temperature setpoint but had quite severe peaks every now and then which can be also seen in the large standard deviations of the temperature in Table 5.8. This was not due to control actions but rather due to disturbances or normal process fluctuation. The slow performance of the whole control system can be clearly seen in the third figure from the top in Fig. 5.16, in which the secondary controller is still decreasing fuel feed due to the fact that the temperature is slightly over the setpoint in a situation where the O_2 concentration has been over the setpoint for a long time which actually calls for more fuel to be added to the boiler to get the O_2 down. The CO emissions during this period were on a modest level. However, this is due to the high O_2 concentration setpoint and the standard deviations of CO were higher than in majority of the open loop experiments.

Naturally this control structure is meant only for drifting control since the time constants of the integrative parts in the controllers are so long. However, since the air feed depends on the backpressure in the boiler and the draught, it would be extremely beneficial with this control structure that the air feed would be constant i.e. the air feed would be controlled. For example, an increase in the draught would cause an increase to the air feed to the boiler. This would increase both the temperature and the O_2 concentration. In the case of this cascade control structure, the secondary controller would actually decrease the fuel feed (if the setpoint was reached before the draught change) and this would result in even higher oxygen concentration. Although the O_2 concentration was controlled, the long integration time constant of the primary controller causes so slow raising of the temperature setpoint that the O_2 concentration can be on a wrong level for very long times which is very harmful if the draught changes drop the O_2 level below the limit where the CO emissions start to grow. For the O_2 control, the previous two structures gave better results.

It has to be stated that the duration of experimenting this control structure was insufficient due to time limitations of the project. A significantly longer experiment period would have been needed to evaluate properly the performance that is achievable with this control structure. Also the case may be that, had there been a small K-term in the primary controller the control error would have been controlled out faster. However, there is always the risk of losing stability of the control system if the gains of the primary and the secondary controllers are not correct.

6 RESULTS AND DISCUSSION

In this thesis, three different feedback control schemes with different input-output pairs and a cascade control structure were designed, implemented and tested for combustion control of a commercial pellet boiler in CTU's laboratory.

The O_2 feedback control by controlling the air feed always reached the setpoint and was able to maintain it. Standard deviation in the O_2 value is regularly larger than in the open loop control which is not desirable. This is due to the disturbing effect of changing air feed on the process. Since the O_2 control was established in order to obtain as low emissions as possible, the comparison of CO emissions with feedback control of air feed and open loop control reveals that it is possible to reach as low CO emissions with feedback control as with open loop control. Although compensation of fast disturbances was not possible, the drifting of the process can be avoided by utilizing this strategy.

Especially for this control strategy, the tuning rules used seemed to give excessive controller gains. This is partly due to the fact that this process model was the most lag dominated process of the ones discussed in this thesis which caused the heavy gains from the formulas. Additionally, it may indicate that there was some modeling error in the process models obtained which is very possible since the process is nonlinear.

The process, actuators and measurement equipment set severe limitations for the performance obtainable by this control strategy. That is why fast disturbance compensation was out of question and the best controller parameters were the ones with particularly low controller gains to ensure less aggressive behavior because aggressive control only disturbed the process. If the primary and secondary air feeds had been carried out by two separate fans, the results, very likely, would have been better. This is because the undesired addition of primary air when only addition into secondary air feed is wanted, that occurs in this single fan case, could have been avoided. Another way to conduct this separation of air feeds would be to utilize a draught fan and online controllable flaps onto the openings of the primary and secondary ducts. By separating the air feeds, compensation of the grate sweeping would also have been possible. Another limiting factor was the relatively long time delay of the gas analyzer by the means of which the O_2 measurement was conducted. If O_2 would have been measured by a lambda sensor which has significantly faster dynamics, the performance of the system most probably would have been improved. Also the prices of lambda probes are so low that they would make a more reasonable choice for oxygen control in this kind of boiler.

The O_2 feedback control by controlling the fuel feed worked quite well. The combustion behavior when this strategy was in use was very stable. The standard deviation of O_2 was even slightly smaller than in the open loop control case and significantly smaller than with the O_2 control by air feed. CO emissions were also on a very modest level and the deviation in the CO concentration was as small as the ones with the open loop control. Reasons for this stable behavior are the constant air feed, stable combustion temperature (due to the first reason) and the fact that the long measurement delay of the gas analyzer did not affect the system performance so much since the process dynamics were heavier also. Due to the heavy dynamics the fast fluctuations were impossible to control but the drifting of the process was avoided. However, this strategy was only tested once so further testing with it should be conducted. Since the boiler operates in free-draught conditions, in order to obtain genuinely constant air feed, feedback control of the air feed should be utilized by installing air velocity measurement. This would also assure desired power with this strategy since the fuel feed is fitted to the air feed. The desired volumetric flow to the boiler could be calculated based on the desired stoichiometry, desired power and the approximate composition of the fuel.

What is indicated in the Tables 5.5 and 5.6 is that despite the setpoint has been the same in the oxygen control experiments, the operating point plays a clear role in emission production. When run on partial power, the CO emissions are higher due to lower combustion chamber temperature and poorer mixing of air and combustion gases due to lower air flows. Due to these facts the used O_2 setpoints should be power level specific in order to assure clean combustion.

The temperature control by the fuel feed resulted in, yet again, slow but rather stable combustion behavior. Setpoint temperatures were maintained and the deviations in the temperature were as low as with the open loop control. Deviations in the other variables as well were about the same size as in open loop experiments. Despite the slow behavior of the system drifting of the process, which is a clear issue, can be avoided by utilizing this strategy. CO emissions were low due to relatively high air feed in comparison to the setpoint. Since the oxygen correlates with the temperature it would be possible to control the oxygen by setting a suitable setpoint but finding one can be a cumbersome task.

In order to improve the performance of the temperature control by fuel feed, one possibility would be to locate the thermocouple closer to the combustion chamber so that the occurrences in the combustion process would be detected faster. Also finding a location in which the temperature measurement would not be so severely affected by the changes in the air feed would be extremely beneficial.

The cascade control of O_2 resulted in stable but significantly slower system behavior than the previous control structures. This is due to the very long integration time constant of the primary controller. The CO emissions when the cascade control structure was utilized were quite modest but this is due to the high setpoint that was set for the primary (O_2) controller.

This control structure is meant only for level control i.e. avoiding the drifting of the process. The faster secondary controller which controlled temperature was also so slow that only drifting control was possible.

Despite the poorer performance obtained during this one experiment the ability to control both, the O_2 concentration and the temperature, is a major benefit when compared to the previous feedback schemes which can only control one variable. Thus further testing with this control structure should be conducted in the future. Utilizing feedback control of air feed is crucial to enable this control structure to work properly. This is also one reason why the results obtained were poorer than in the previous structures.

All in all, even though the designed feedback control schemes operate quite well, excluding the cascade control structure, it seems to be quite hard to obtain a more stable process behavior by the means of feedback control than can be obtained with the open loop control with this instrumentation. Since the effect from the inputs to the outputs in our case is always quite slow, it is hard or even impossible to control short term fluctuations of the process. Thus in short term, the operation of this boiler is dependent only on the natural behavior of the process in that particular operating point. This explains partly why it was not possible to improve the combustion behavior of the boiler in short term.

7 CONCLUSION

The experimental part of this work was conducted on a commercial pellet boiler in the laboratory of Czech Technical University in Prague (CTU). The project was cooperation between CTU and Tampere University of Technology (TUT).

In the beginning of the thesis, a brief overview of the pellet itself, pellet markets and the popularity of small-scale heating appliances in Finland as well as in Czech Republic was made to motivate the reader why improving combustion control in small-scale appliances is important. The combustion process and emission production in combustion were examined thoroughly in order to build basis for process knowledge and to enable justifiable decision making when evaluating the different control strategies. The equipment that was used included the commercial small-scale boiler, laboratory measurement equipment (i.e. sensors etc.) and REX control system.

The fourth chapter included all the control methods that were used i.e. identification and different controller tuning methods for PID controllers. Also a survey was made to evaluate the possibilities of utilizing feedback control with different inputs to different outputs. The inputs were fuel feed and air feed whereas the outputs were chosen to be oxygen concentration in flue gas and temperature after the first heat exchanger. The three feedback control schemes that were further examined were: feedback control of oxygen by the air feed, feedback control of oxygen by the fuel feed and feedback control of temperature by the fuel feed. In addition to these, a cascade control structure was also designed and tested. The possibility of compensating the grate sweeping disturbances by control was examined and it was concluded that with these actuators, the compensation is impossible.

The feedback control schemes were designed and implemented based on the principles from fourth chapter and the results from the control experiments were evaluated in the fifth chapter. All the feedback control strategies worked and some of them achieved as stable combustion behavior as with the open loop control. However, due to the heavy dynamics from the inputs to the process variables, it was not possible to obtain smaller deviations into the combustion variables than with the open loop control i.e. to utilize active disturbance control. However, since drifting, which is a clear issue in combustion processes, was prevented the improvement to the combustion conditions that was obtained with the feedback control is significant. It was concluded that the lack of two separate air feeds (primary and secondary) and lack of air velocity measurement limited severely the obtainable performance improvement with the different control structures. Additionally, in order to improve the performance of the oxygen control it would be essential to utilize lambda probe to avoid the long measurement delay of the gas analyzer.

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APPENDIX 1

The discrete transfer function of the running average filter with 750 s time window:

$$y_{filter} = \frac{\frac{1}{750}}{1 - \frac{749}{750} z^{-1}}$$

Moving median filter-code:

```
function med = mov_median(sample)
  global i;
  global buff

  buff(i) = sample;
  sbuff = sort(buff);
  i = i+1;
  if (i > 600)
      i = 1;
  end
  med = sbuff(300);
end
```

Moving mean filter-code:

```
function mean_m = mov_mean(sample,buff_size);
  global i2;
  global buff2;

  buff2(i2) = sample;
  i2 = i2+1;
  if (i2 > buff_size);
      i2 = 1;
  end
  mean_m = sum(buff2)/buff_size;
end
```